

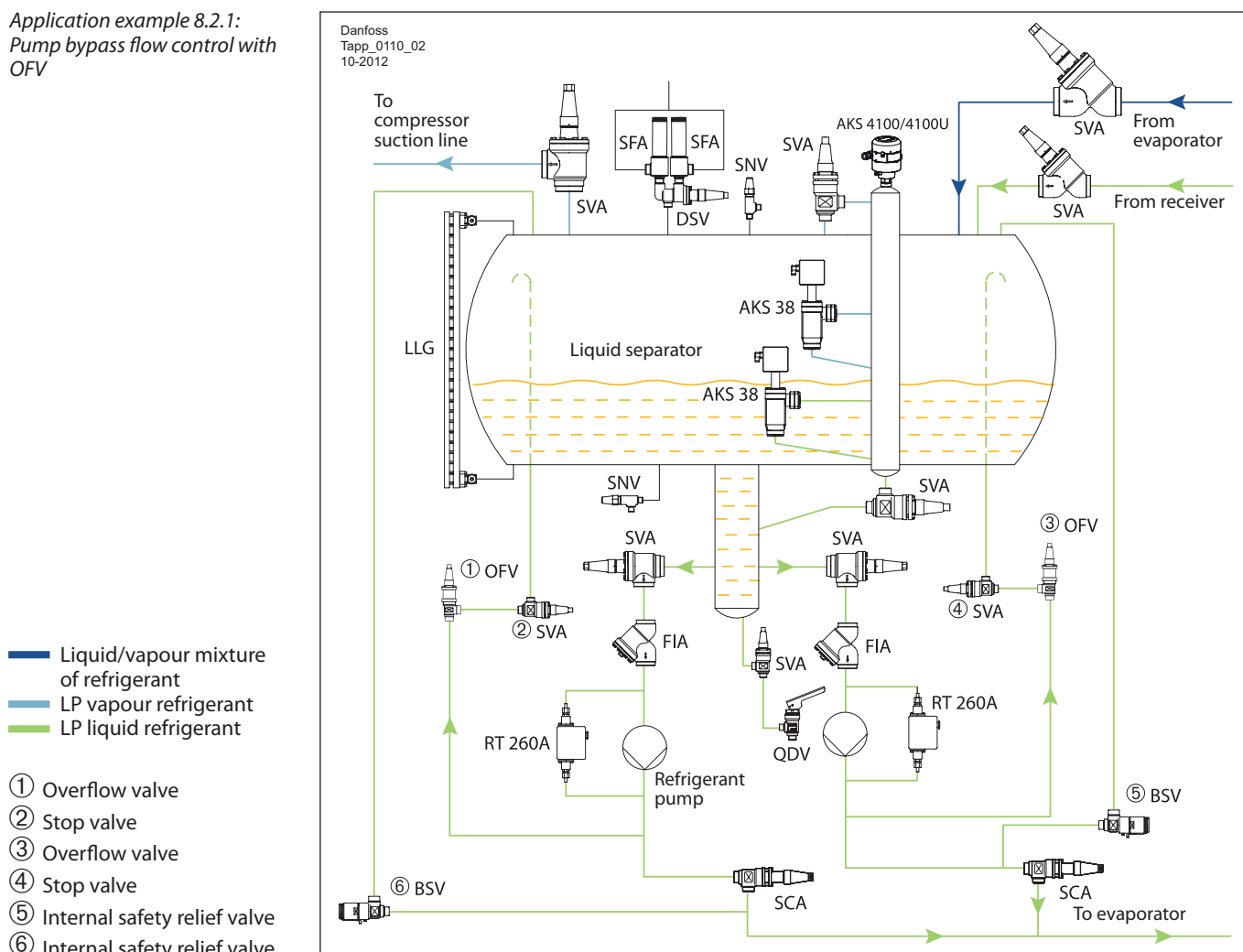
8.2 Pump Bypass Flow Control

The most common way to keep the flow through the pump above the minimum permissible value (Q_{\min} in fig. 8.2) is to design a bypass flow for the pump.

The bypass line can be designed with regulating valve REG, differential pressure overflow valve OFV, or even just an orifice.

Even if the liquid supply to all evaporators in the system is stopped, the bypass line can still keep a minimum flow through the pump.

Application example 8.2.1:
Pump bypass flow control with OFV



The bypass line is designed for each pump with overflow valve OFV. The internal overflow valve BSV is designed for safety relief when there is excessive pressure. For

example, when the stop valves are closed, the liquid refrigerant trapped in the pipes may be heated to excessive high pressure.

Technical data

	Overflow valve - OFV
Material	Body: steel
Refrigerants	All common refrigerants, incl. R717
Media temp. range [°C]	-50 to 150
Max. working pressure [bar]	40
DN [mm]	20/25
Opening differential pressure range [bar]	2 to 8

	Safety relief valve - BSV (Back pressure independent)
Material	Housing: special steel approved for low temperature operation
Refrigerants	R717, R744, HFC, HCFC and other refrigerants (depending on the sealing material compatibility)
Media temp. range [°C]	-30 to 100 as an external safety relief valve -50 to 100 as a pilot valve for POV
Set pressure [bar]	10 to 25
Flow area [mm ²]	50

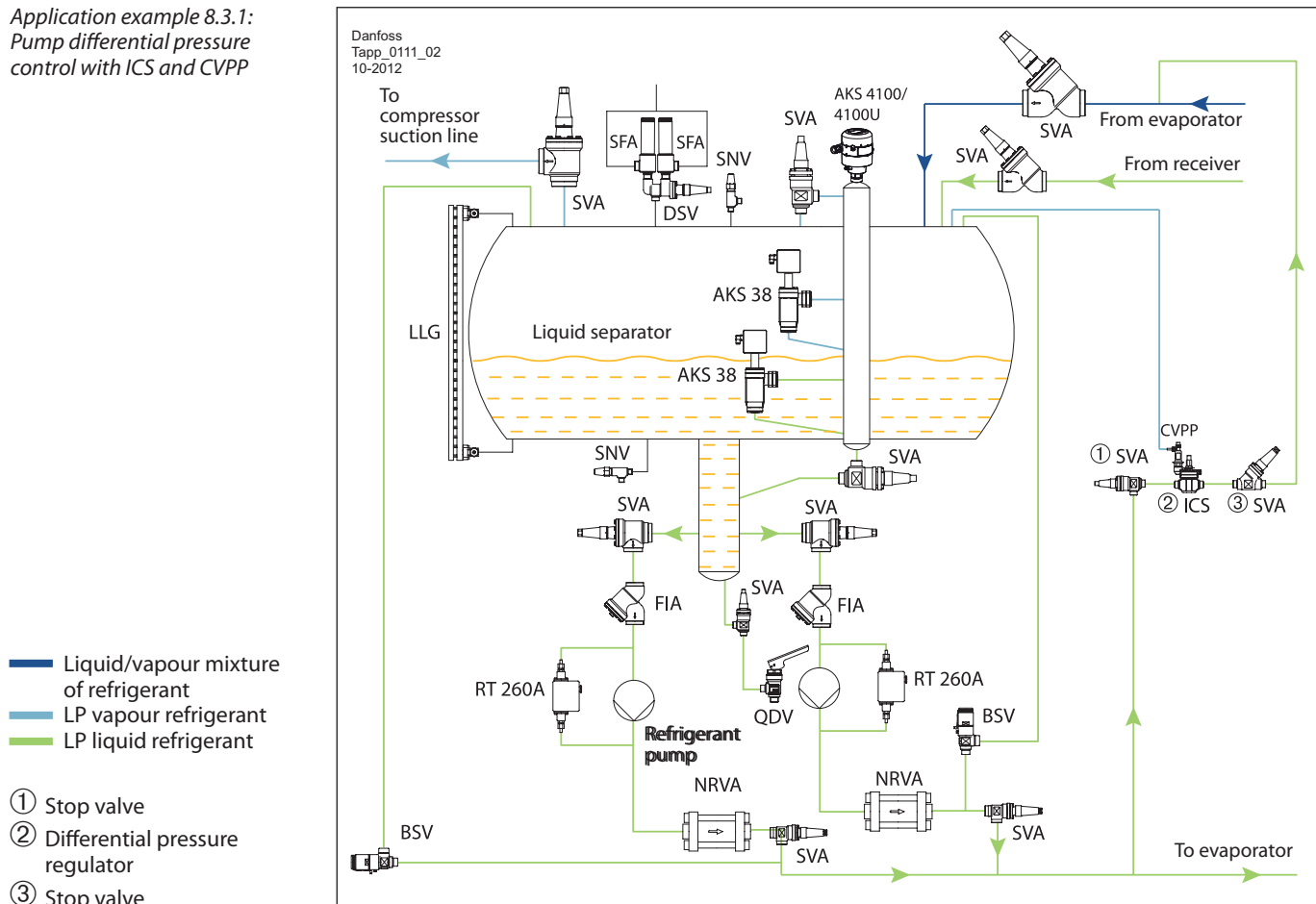
Not all valves are shown.
Not to be used for construction purposes.

8.3 Pump Pressure Control

Application example 8.3.1:
Pump differential pressure control with ICS and CVPP

It is of great importance to some types of pump circulation systems that a constant differential pressure can be maintained across the permanently set throttle valve before the evaporator.

By using pilot controlled servo valve ICS and pilot valve CVPP, it is possible to maintain a constant differential pressure across the pump, and therefore a constant differential pressure across the throttle valve.



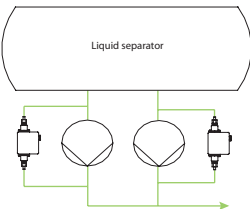
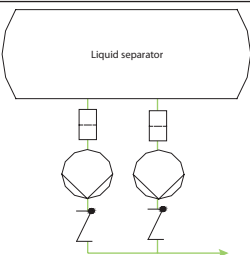
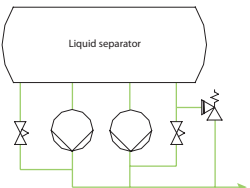
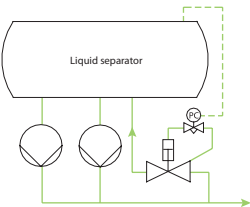
Technical data

	Pilot-operated servo valve - ICS
Material	Body: low temp. steel
Refrigerants	All common refrigerants, incl. R717 and R744
Media temperature range [°C]	-60 to 120
Max. working pressure [bar]	52
DN [mm]	20 to 150

	Differential pressure pilot valve-CVPP
Refrigerants	All common non-flammable refrigerants incl. R717
Media temp. range [°C]	-50 to 120
Max. working pressure [bar]	CVPP (LP): 17 CVPP (HP): up to 40
Regulating range [bar]	CVPP (LP): 0 to 7 CVPP (HP): 0 to 22
K _v value m ³ /h	0.4

Not all valves are shown.
Not to be used for construction
purposes.

8.4 Summary

Solution		Application	Benefits	Limitations
Pump Protection with Differential Pressure Control				
Pump protection with differential pressure control RT 260A		Applicable to all pump circulation systems.	Simple. Effective in protecting the pump against low differential pressure (corresponding to high flow).	Not applicable to flammable refrigerants.
Filter and Check Valve				
Filter FIA and check valve NRVA on the pump line		Applicable to all pump circulation systems.	Simple. Effective in protecting the pump against back flow and particles.	Filter on the suction line may lead to cavitation when blocked. Filter on the discharge line still allows particles to enter the pump.
Pump Bypass Flow Control				
Pump bypass flow control with REG and protection with safety relief valve BSV		Applicable to all pump circulation systems.	Simple. Effective and reliable in keeping the minimum flow for the pump. Safety valve can effectively prevent excessive pressure.	Part of pump power wasted.
Pump Pressure Control				
Pump pressure control with ICS and CVPP		Applicable to pump circulation systems that require constant differential pressure across the regulating valves before evaporators.	Provides a constant differential pressure and circulation ratio for the evaporators.	Part of pump power wasted.

8.5 Reference Documents

For an alphabetical overview of all reference documents please go to page 146

Technical Leaflet / Manual

Type	Literature no.	Type	Literature no.
BSV	PD.IC0.A	NRVA	PD.FK0.A
CVPP	PD.HN0.A	REG	PD.KM1.A
FIA	PD.FM1.A	RT 260A	PD.CB0.A
ICS	PD.HS2.A	SVA	PD.KD1.A

Product instruction

Type	Literature no.	Type	Literature no.
BSV	PI.IC0.A	NRVA	PI.FK0.A
CVPP	PI.HN0.C	REG	PI.KM1.A
FIA	PI.FN1.A	RT 260A	RI5BB
ICS 25-65	PI.HS0.A	SVA	PI.KD1.A
ICS 100-150	PI.HS0.B		

To download the latest version of the literature please visit the Danfoss website.

9. Others

9.1 Filter Driers in Fluorinated Systems

Water, acids and particles appear naturally in fluorinated refrigeration systems. Water may enter the system as a result of installation, service, leakage, etc..

Acid is formed as a result of the breakdown of the refrigerant and oil.

Particles usually result from soldering and welding residue, the reaction between refrigerant and oil, etc.

Failure to keep the contents of acids, water and particles within acceptable limits will significantly shorten the lifetime of the refrigeration system and even burn out the compressor.

Too much moisture in systems with evaporating temperatures below 0°C could form ice which may block control valves, solenoid valves, filters, and so on. Particles increase the wear-and-tear of the compressor and valves, as well as the possibility of creating a blockage. Acids are not corrosive if there is no water. But in water solution, acids can corrode the pipe work and plate the hot bearing surfaces in the compressor.

This plating builds up on to the hot bearing surfaces including the oil pump, crankshaft, con rods, piston rings, suction and discharge valve reeds etc. This plating causes the bearings to run hotter as the lubrication gap in the bearings reduces as the plating gets thicker.

Cooling of the bearings is reduced due to less oil circulating through the bearing gap. This causes these components to get hotter and hotter. Valve plates start to leak by causing higher discharge superheating effect. As the problems escalate the compressor failure is imminent.

Filter driers are designed to prevent all the above circumstances. Filter driers serve two functions: drying function and filtering function.

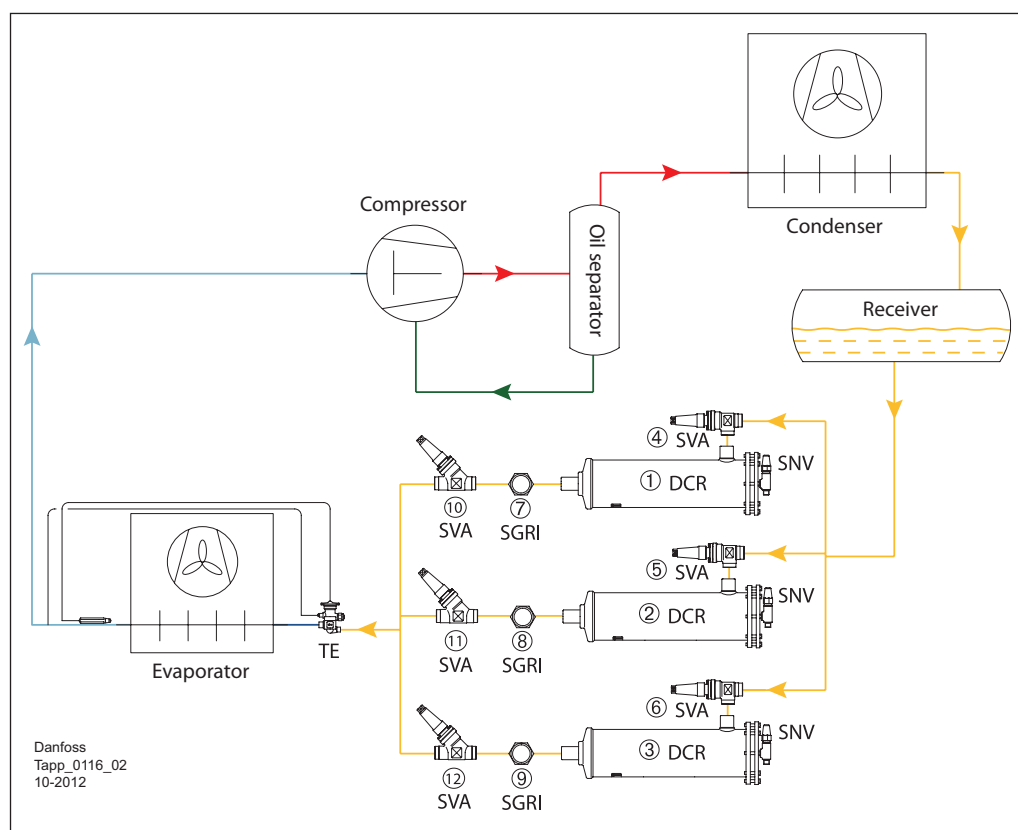
The drying function constitutes the chemical protection and includes the adsorption of water and acids. The purpose is to prevent corrosion of the metal surface, decomposition of the oil and refrigerant and avoid burn-out of motors.

The filter function constitutes the physical protection and includes retention of particles and impurities of any kind. This minimizes the wear and tear of the compressor, protects it against damage and significantly prolongs its life.

Application example 9.1.1:
Filter drier in fluorinated systems

- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- Oil

- ① Filter drier
- ② Filter drier
- ③ Filter drier
- ④ Stop valve
- ⑤ Stop valve
- ⑥ Stop valve
- ⑦ Sight glass
- ⑧ Sight glass
- ⑨ Sight glass
- ⑩ Stop valve
- ⑪ Stop valve
- ⑫ Stop valve



For fluorinated systems, filter driers are normally installed in the liquid line before the expansion valve. In this line, there is only pure liquid flow through the filter drier (unlike the two-phase flow after the expansion valve).

The pressure drop across the filter drier is small, and the pressure drop in this line has little influence on the performance of the system. The installation of filter drier could also prevent ice formation in the expansion valve.

In industrial installations the capacity of one filter drier is not normally sufficient to dry the whole system, therefore several filter driers could be installed in parallel.

DCR is a filter drier with interchangeable solid cores. There are three types of solid cores: DM, DC and DA.

- **DM** - 100% molecular sieve solid core suitable for HFC refrigerants and CO₂;
- **DC** - 80% molecular sieve and 20% activated alumina solid core suitable for CFC & HCFC refrigerants and compatible with HFC refrigerants;
- **DA** - 30% molecular sieve and 70% activated alumina solid core suitable for clean up after compressor burn-out and compatible with CFC / HCFC / HFC refrigerants.

In addition to the above normal solid cores, Danfoss also provide other customer-tailored solid cores. And Danfoss also provide filter driers with fixed solid cores. For more information, please refer to the product catalogue or contact your local sales companies.

The sight glass with indicator for HCFC/CFC, type SGRI, is installed after the filter drier to indicate the water content after drying. Sight glasses with indicator for other types of refrigerants can also be provided. For more information, please refer to Danfoss product catalogue.

Technical data

	Filter drier - DCR
Refrigerants	CFC/HFC/HCFC/R744
Material	Housing: steel
Max. working pressure [bar]	HP: 46
Operating temp. range [°C]	-40 to 70
Solid cores	DM/DC/DA

Not all valves are shown.
Not to be used for construction purposes.

9.2 Water Removal for Ammonia Systems

The issue of water in ammonia systems is unique compared with fluorinated systems and CO₂ systems:

The molecular structure of ammonia and water are similar, both small and polar and as a result both ammonia and water are completely soluble.

As a result of the similarity of ammonia and water molecular, there has been no efficient filter drier for ammonia. Furthermore, because of the high solubility of water in ammonia, free water is difficult to extract from the solution.

Water and ammonia will co-exist and act as a kind of zeotropic refrigerant, whose saturated P-T relationship is no longer the same as anhydrous ammonia.

These are factors as to why ammonia systems are seldom designed as DX systems: on one hand, the liquid ammonia is hard to completely vaporize when water is present, which will lead to liquid hammer; on the other hand, how can a thermostatic expansion valve function correctly when the saturated P-T relationship changes?

Pumped liquid circulation systems could well avoid the potential damages of water to the compressors. With only vapour entering in the suction line, liquid hammer is avoided; and so long as there is not too much water in the liquid, the vapour will contain nearly no water (< the recommended max. of 0.3%), which could effectively avoid the oil pollution by water.

While pumped liquid circulation systems effectively avoid damage to the compressors, it also keeps the other penalties of water unnoticed:

- **COP of the system is reduced**
When there is water content, the saturated P-T relationship of the refrigerant will be different from pure ammonia. Specifically, the refrigerant will evaporate at a higher temperature for a given pressure. This will decrease the refrigeration capacity of the system and increase power consumption.
- **Corrosion**
Ammonia becomes corrosive with water present and start to corrode the pipe work, valves, vessels, etc.
- **Compressor problems**
If water is taken into the compressors, e.g. due to inefficient liquid separators, it will also lead to oil and corrosion problems to the compressors.

Therefore, to keep the system in efficient and trouble-free mode, it's recommended to detect water regularly, and employ some water removal method when the water content is found to be above the acceptable level.

Basically, there are three ways to deal with water contamination:

- **Change the charge**
This is suitable for systems with small charges (e.g. chillers with plate type evaporators), and it should comply with local legislation.
- **Purging from some evaporators**
This is suitable for some gravity driven systems without hot gas defrost. In these systems, water remains in the liquid when ammonia vaporizes, and accumulates in the evaporators.
- **Water rectifier**
Part of contaminated ammonia is drained into the rectifier, where it is heated, with the ammonia vaporising and the water drained. This is the only way of removing water for pumped liquid circulation systems.

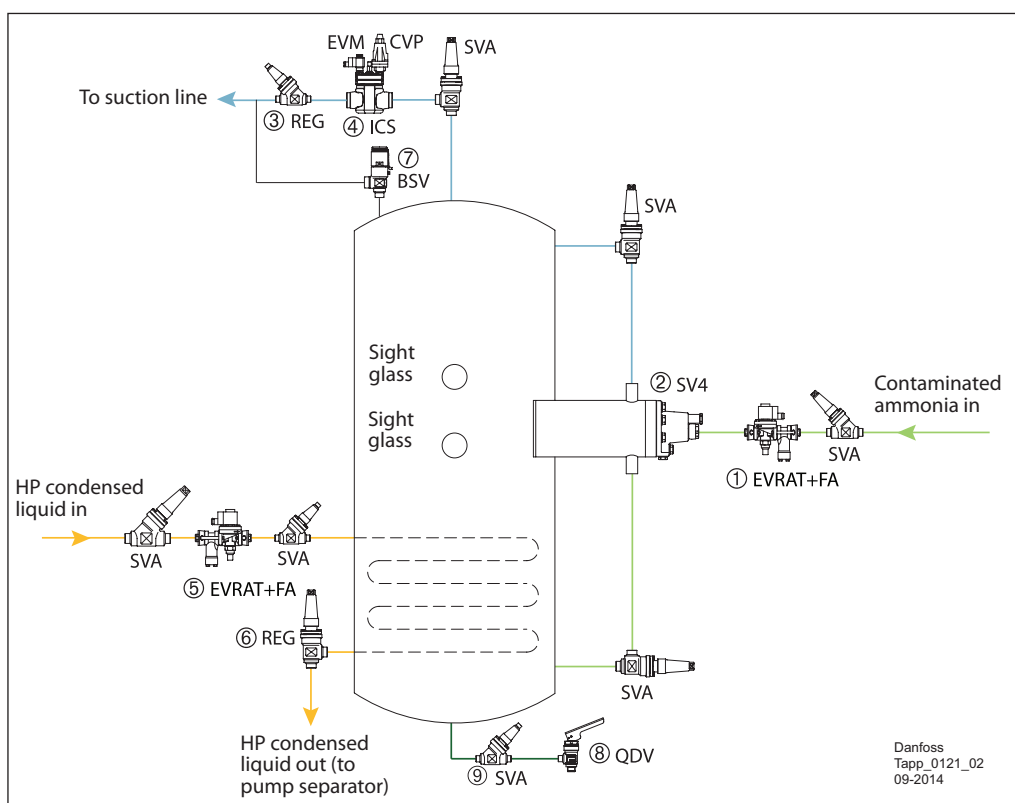
For more information on water contamination and water removal in ammonia refrigeration systems, please refer to IIR bulletin 108.

It is necessary to mention that there is a down side to too low water content - the possibility of a special kind of steel corrosion. However it is not likely in a real plant.

Application example 9.2.1:
Water rectifier heated by hot gas
controlled by float valves

— HP vapour refrigerant
— HP liquid refrigerant
— LP vapour refrigerant
— LP liquid refrigerant
— Oil

- ① Solenoid valve
- ② Float valve
- ③ Hand regulating valve
- ④ Pressure regulating valve
- ⑤ Solenoid valve
- ⑥ Hand regulating valve
- ⑦ Internal safety valve
- ⑧ Quick drain valve
- ⑨ Stop valve



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Procedure for removing water:

1. Energise the solenoid valve EVRAT ① and ICS+EVM ④. Contaminated ammonia is drained into the rectifying vessel. The float valve SV4 ② will close when the liquid level in the vessel reaches the set level. Energise the solenoid valve EVRAT ⑤.
2. Condensed liquid is fed to the coil inside the vessel and starts to heat the contaminated ammonia. Ammonia starts to evaporate, and contaminated liquid remains in the vessel. When ammonia evaporates in the vessel and the liquid level drops, the float valve SV4 ② will open and drain more contaminated ammonia into the vessel. After a certain time, based on experience, preparation for draining the contaminated liquid can start.
3. De-energise the solenoid valve EVRAT ①. After a certain time all ammonia will be evaporated, and only contaminated liquid will be left in the vessel. To drain the contaminated liquid from the vessel the pressure inside the vessel must be increased to a pressure above 0°C. This is done by de-energizing the solenoid valve ICS+EVM ④. Now the pressure inside the vessel is controlled by the ICS+CVP ④. Open a few turn the stop valve SVA ⑨ and carefully open drain valve QDV ⑧, and drain off the contaminated liquid remaining in the vessel.
4. Close the drain valve QDV ⑧ and stop valve SVA ⑨. Then de-energise the solenoid valve ⑤, to stop the contaminated liquid removal process, or if necessary, repeat step 1 to continue the process.

For safety considerations, safety relief valve BSV ⑦ is installed on the vessel to avoid excessive pressure build up.

Not all valves are shown.
Not to be used for construction
purposes.

9.3 Air purging systems

Presence of Non Condensable Gases

Non-condensable gases are present in refrigeration systems at the outset of the installation process, with pipes and fittings being full of air. Therefore, if a good vacuum process is not undertaken air can remain within the system.

Additionally, air can enter the system as a result of the system leaking, when the system is open for maintenance, penetration through the system components, leaking at welded connections where the pressure of the ammonia is lower than atmospheric pressure (below -34°C evaporating temperature), when adding oil, etc.

Moreover, impurities in the refrigerant and / or decomposition of the refrigerant or the lubricating oil due to high discharge temperatures may generate non-condensable gases (e.g. Ammonia decomposes into nitrogen and hydrogen).

Location & Detection

Non-condensable gases are contained within the high pressure side of the refrigeration system, mainly in the coldest and less agitated points in the condenser.

A simple way to check for the presence of non-condensable gases in the system, is to compare the pressure difference between the actual condensing pressure, read at the pressure gauge of the receiver and the saturated pressure corresponding to the temperature measured at the condenser outlet.

For example if 30°C is measured at the outlet of the condenser in an ammonia system, the related saturated temperature is 10.7 bar g and if the pressure gauge reading is 11.7 bar g then there is 1 bar difference and this is due to the presence of non-condensable gases.

Problems generated

The air tends to form a film over the condenser pipes isolating the heat transfer surface from the refrigerant in the condenser. The result is a reduction of the condenser capacity and thus an increase in the condensing pressure. The energy efficiency will then decline and depending on the condensing pressure, the potential for oil related problems would increase.

The capacity reduced in the condenser is a fact but is very hard to determine. Air purger manufacturers have provided some data, which indicates a 9-10 % capacity reduction for every bar of increased condensing pressure. If a more accurate calculation is required, ASHRAE gives some guidelines on how to estimate it as well as some examples of research undertaken with the results achieved. (HVAC Systems & Equipment Manual, Non-Condensable Gases).

Other manufacturers estimate the risks and the associated costs rising from the compressor side. As the condensing pressure and discharge temperature increase, there will be higher risks to the bearings due to oil problems, as well as an increase in the running cost of a compressor. The cost estimation is related to the compressor type and size in the plant.

All in all the presence of non-condensable gases is as undesirable as unavoidable and air purging equipment is often used.

Air purging systems

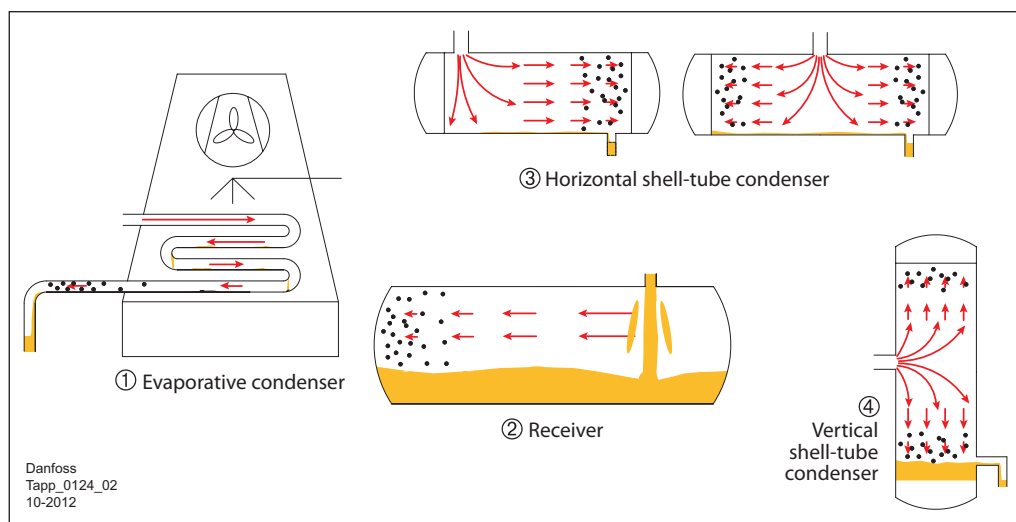
The air or non-condensable gases can be purged out of the system manually. This is performed by maintenance personnel and may lead to excessive refrigerant losses.

Another way of purging is called refrigerated purging: gases coming from the sampling points are cooled down inside a chamber with a cooling coil in order to condense the refrigerant and return it back to the system. The gases then left in the chamber should be purged out to the atmosphere. The idea of cooling down and condensation is to reduce the amount of refrigerant released.

The refrigerant used for the cooling coil could be the same as the refrigeration plant; it can also be another different refrigerant.

Location for purge connection is quite difficult and depends on the system and condenser type. Below are some examples of purge points. In the picture, the arrows in the condenser coils and the vessels represent the flow velocities. The length of arrow decreases as the velocity decreases.

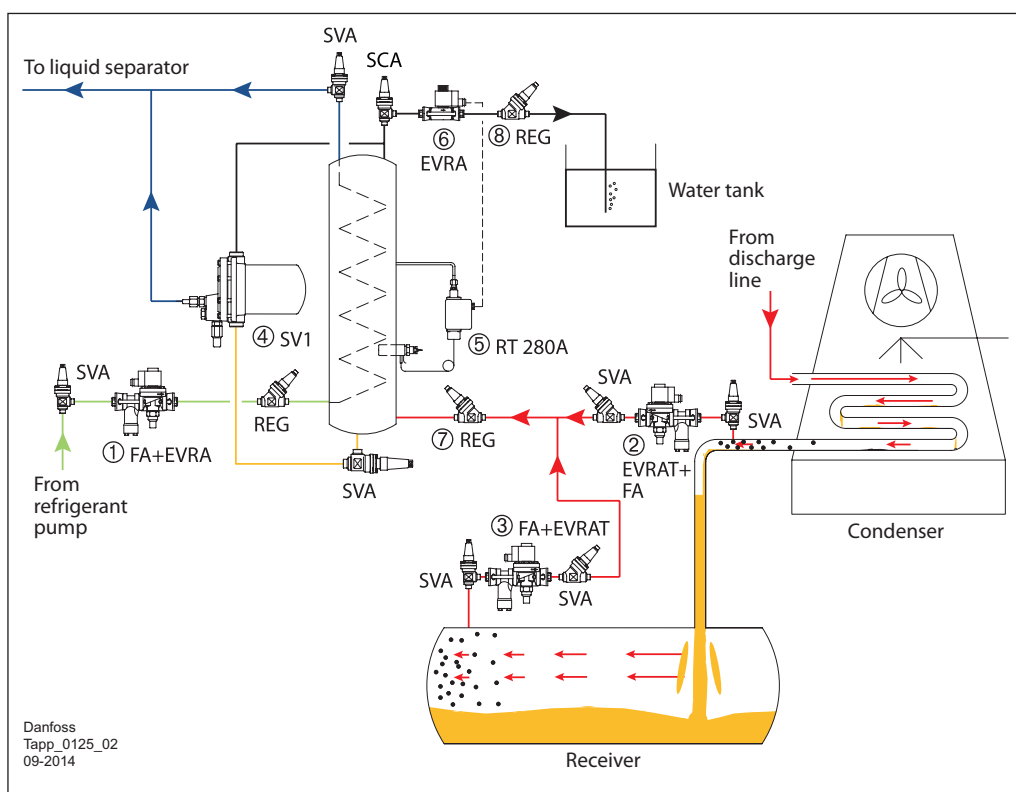
The air accumulation is shown by the black dots. These places with high content of air are where samples for purging should be taken.



Application example 9.3.1:
Automatic air purging system
using the refrigerant from the
plant

— HP vapour refrigerant
— HP liquid refrigerant
— Liquid/vapour mixture
of refrigerant
— LP liquid refrigerant
— Air

- ① Solenoid valve
- ② Solenoid valve
- ③ Solenoid valve
- ④ Float valve
- ⑤ Pressure switch
- ⑥ Solenoid valve
- ⑦ Hand regulating valve
- ⑧ Hand regulating valve



Steps for air purging:

1. Energise the solenoid valve EVRA ①, so that low pressure liquid refrigerant enters the coil and cools down the refrigerant contained in the vessel.
2. Energise the solenoid valve EVRAT ② or ③ (only ONE of them). Gas refrigerant with accumulated air is drawn into the vessel, inside which refrigerant vapour condenses and air rises to the top of the vessel. The float valve SV1 ④ drains the condensed liquid refrigerant automatically.
3. With the air that accumulates in the top of the vessel, the total pressure inside the vessel compared with the saturated pressure of the liquid refrigerant increases. When this pressure reaches the setting on the pressure switch RT 280A ⑤ opens the solenoid valve EVRA ⑥ and purges some air from the vessel.

The regulating valve ⑧ must be adjusted to a relatively small opening degree to have a controlled/slow purging of the air from the vessel.

The regulating valve ⑦ must be adjusted to a relatively small opening degree as it must create a pressure drop to enable as low a pressure inside the airpurger. Alternatily a small orifice can be fitted down-strem of the regulating valve ⑦.

9.4 Heat Recovery System

The free heat from de-superheating and/or condensing in the condenser can be reclaimed if there are requirements for heating in the plant. These include heating of air in offices or shops, heating water for washing or processing, preheating boiler feed water, etc.

To make heat recovery an economic solution, it is important to ensure that the free heat and the heating requirements match in terms of timing, temperature level and heat flow. For example, for production of hot water, i.e. when heat at high temperature level is required, the de-superheating heat could be recovered; whilst for office heating, usually the recovery of all the condenser heat could be considered.

A well designed control system is crucial for trouble free and efficient operation of refrigeration systems with heat recovery.

The purpose of control is to coordinate heat recovery and refrigeration:

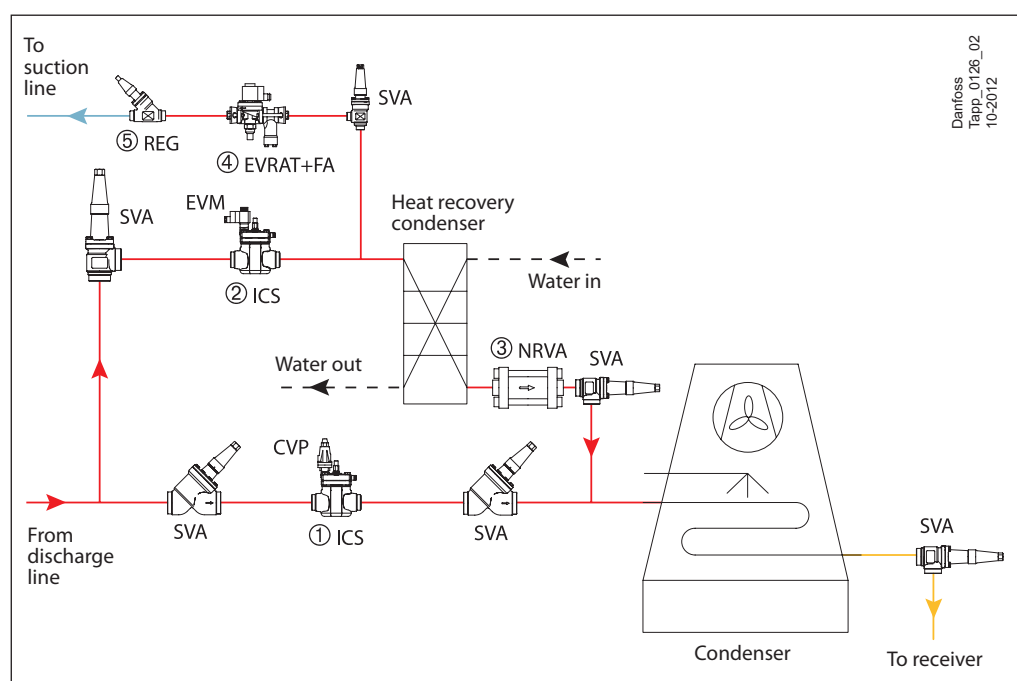
1. The basic function of refrigeration should be ensured whether the heat recovery is running or not. The condensing pressure should not be too high when heat recovery stops. Furthermore for DX systems, the condensing pressure should not be too low either (See section 3).
2. The requirements for heat recovery, e.g. the temperature and the heat flow, should be fulfilled.
3. Trouble free on/off control of the heat recovery loop according to the demand.

Heat recovery control needs very sophisticated design, which may vary from plant to plant. The following are some examples:

*Application example 9.4.1:
Control for series arrangement
of recovery heat exchanger and
condenser*

- HP vapour refrigerant
- HP liquid refrigerant
- LP vapour refrigerant
- Water

- ① Pressure regulator
- ② Solenoid valve
- ③ Check valve
- ④ Solenoid valve
- ⑤ Hand regulating valve



This heat recovery system is applicable to air as well as water.

Refrigerating cycle without heat recovery

Hot gas from the discharge line is led directly to the main condenser through the pilot-operated servo valve ICS ① with constant pressure pilot CVP (HP). The check valve NRVA ③ prevents the flow back towards the heat recovery condenser.

Heat recovery cycle

The pilot operated servo valve ICS ② is controlled by the on/off switching of the pilot solenoid valve EVM, through a time clock, thermostat etc. Hot gas enters the recovery condenser.

ICS ① will normally close because of the increased condensing capacity and decreased discharge pressure. If the discharge pressure increases, constant pressure pilot CVP (HP) will open the servo valve ICS ① so that part of the hot gas can flow towards the main condenser.

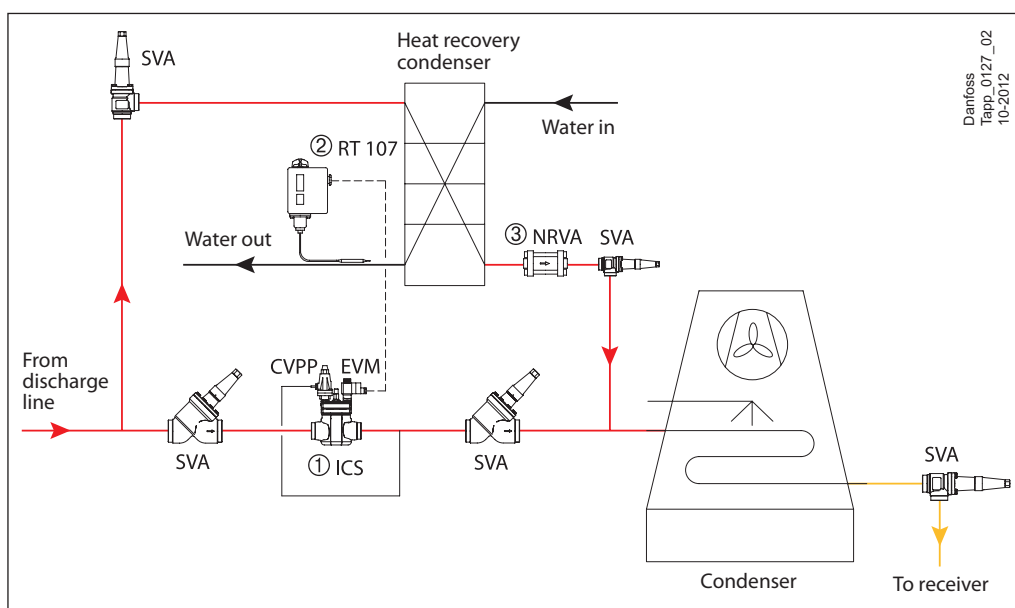
In summertime the heat recovery condenser is idle for extended periods of time. To avoid the risk of accumulation of liquid in this condenser, a solenoid valve EVRA ④ and a regulating valve REG ⑤ ensure periodic evaporation of any condensate in the recovery condenser.

Not all valves are shown.
Not to be used for construction
purposes.

Application example 9.4.2:
Control for series arrangement
of recovery heat exchanger and
condenser

— HP vapour refrigerant
— HP liquid refrigerant
— Water

- ① Differential pressure regulator
- ② Thermostat
- ③ Check valve



This heat recovery system is applicable to central refrigeration plant with several compressors.

Provided only a small proportion of compressor capacity is used, all the discharge gas will pass through the recovery condenser and then to the main condenser.

The greater the amount of compressor capacity used, the higher becomes the pressure drop in the recovery condenser.

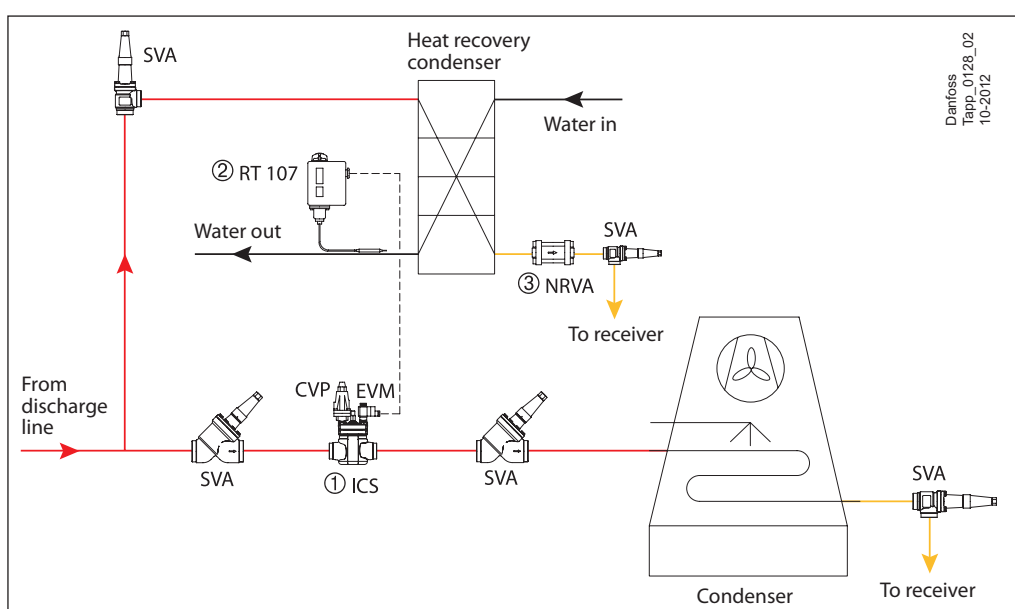
When this pressure drop exceeds the setting of differential pressure pilot CVPP(HP) on the servo valve ICS ① partially opens and excess pressure gas is led direct into the main condenser.

When the desired water or air temperature has been achieved by means of the heat recovery condenser, the thermostat RT 107 ② activates the on/off pilot EVM, and the servo valve ICS ① will open fully.

Application example 9.4.3:
Control for parallel arrangement
of recovery heat exchanger and
condenser

— HP vapour refrigerant
— HP liquid refrigerant
— Water

- ① Pressure regulator and solenoid valve
- ② Thermostat
- ③ Check valve



This heat recovery system is applicable to systems with several compressors - e.g. for the heating of central heating water.

In normal operation the servo valve ICS ① is kept open by the on/off switching of the solenoid valve pilot EVM, activated by an external control connected to the thermostat RT 107.

In wintertime, when the heating demand necessitates heat recovery, the solenoid valve

pilot EVM is closed, which in turn causes the servo valve ICS ① to close. If the condensing pressure exceeds the setting of the constant pressure pilot CVP (HP), the servo valve ICS 3 will open and excess pressure gas will be led to the main condenser.

The check valve NRVA prevents flow back of refrigerant to the recovery condenser.

Not all valves are shown.
Not to be used for construction
purposes.

9.5 Reference Documents

For an alphabetical overview of all reference documents please go to page 146

Technical Leaflet / Manual

Type	Literature no.	Type	Literature no.
BSV	PD.IC0.A	REG	PD.KM1.A
CVP	PD.HN0.A	RT 107	PD.CB0.A
DCR	PD.EJ0.A	SGR	PD.EK0.A
EVM	PD.HN0.A	SNV	PD.KB0.A
EVRA(T)	PD.BM0.B	SVA	PD.KD1.A
ICS	PD.HS2.A	SV 1-3	PD.GE0.B
NRVA	PD.FK0.A	SV 4-6	PD.GE0.D

Product instruction

Type	Literature no.	Type	Literature no.
BSV	PI.IC0.A	REG	PI.KM1.A
CVP	PI.HN0.C	SGR	PI.EK0.A
DCR	PI.EJ0.B	SNV	PI.KB0.A
EVM	PI.HN0.N	SVA	PI.KD1.A
EVRA(T)	PI.BN0.L	SV 1-3	PI.GE0.C
ICS 25-65	PI.HS0.A	SV 4-6	PI.GE0.B
ICS 100-150	PI.HS0.B		
NRVA	PI.FK0.A		

To download the latest version of the literature please visit the Danfoss website.

10. Using CO₂ in refrigeration systems

The use of carbon dioxide (CO₂) in refrigeration systems is not new. Carbon dioxide was first proposed as a refrigerant by Alexander Twining (ref. [1]), who mentioned it in his British patent in 1850. Thaddeus S.C. Lowe experimented with CO₂ for military balloons, but he also designed an ice machine with CO₂ in 1867. Lowe also developed a machine onboard a ship for transportation of frozen meat.

From the literature it can be seen that CO₂ refrigerant systems were developed during the following years and they were at their peak in the 1920's and early 1930's. CO₂ was generally the preferred choice for use in the shipping industry because it was neither toxic nor flammable, whilst ammonia (NH₃ or R717) was more common in industrial applications (ref. [2]). CO₂ disappeared from the market, mainly because the new "miracle refrigerant" Freon had become available and was marketed very successfully.

Ammonia has continued to be the dominant refrigerant for industrial refrigeration applications over the years. In the 1990's there was renewed interest in the advantages of using CO₂, due to ODP (Ozone Depletion Potential) and GWP (Global Warming Potential), which has restricted the use of CFCs and HFCs and imposed limits on refrigerant charges in large ammonia systems.

CO₂ is classified as a natural refrigerant, along with ammonia, hydrocarbons such as propane and butane, and water. All of these refrigerants have their respective disadvantages.

Ammonia is toxic, hydrocarbons are flammable, and water has limited application potential. By contrast, CO₂ is non-toxic and non-flammable.

CO₂ differs from other common refrigerants in many aspects and has some unique properties. Technical developments since 1920 have removed many of the barriers to using CO₂, but users must still be highly aware of its unique properties, and take the necessary precautions to avoid problems in their refrigeration systems.

The chart in figure 10.1 shows the pressure/temperature curves of CO₂, R134a and ammonia. Highlights of CO₂'s properties relative to the other refrigerants include:

- Higher operating pressure for a given temperature
- Narrower range of operating temperatures
- Triple point at a much higher pressure
- Critical point at a very low temperature.

While the triple point and critical point are normally not important for common refrigerants, CO₂ is different. The triple point is relatively high at 5.2 bar [75.1 psi], but more importantly, higher than normal atmospheric pressure.

This can create problems unless suitable precautions are taken. Also, CO₂'s critical point is very low: 31.1°C [88.0°F], which strongly affects design requirements.

In the table below, various properties of CO₂ are compared with those of R134a and ammonia.

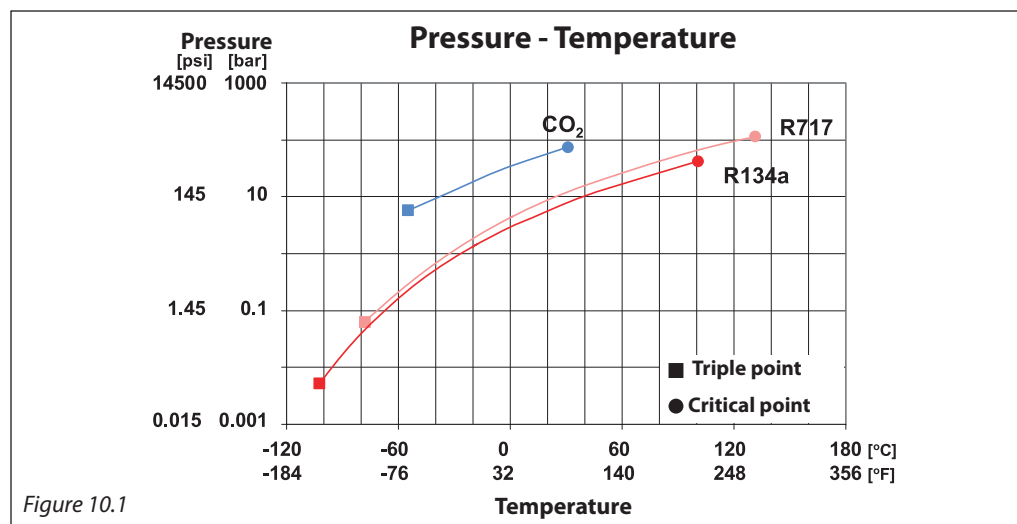


Figure 10.1

Refrigerant		R 134a	NH ₃	CO ₂
Natural substance		NO	YES	YES
Ozone Depletion Potential (ODP)*		0	0	0
Global Warming Potential (GWP)*		1300	-	1
Critical point	bar [psi]	40.7 [590]	113 [1640]	73.6 [1067]
	°C [°F]	101.2 [214]	132.4 [270]	31.1 [87.9]
Triple point	bar [psi]	0.004 [0.06]	0.06 [0.87]	5.18 [75.1]
	°C [°F]	-103 [-153]	-77.7 [-108]	-56.6 [-69.9]
Flammable or explosive		NO	(YES)	NO
Toxic		NO	YES	NO

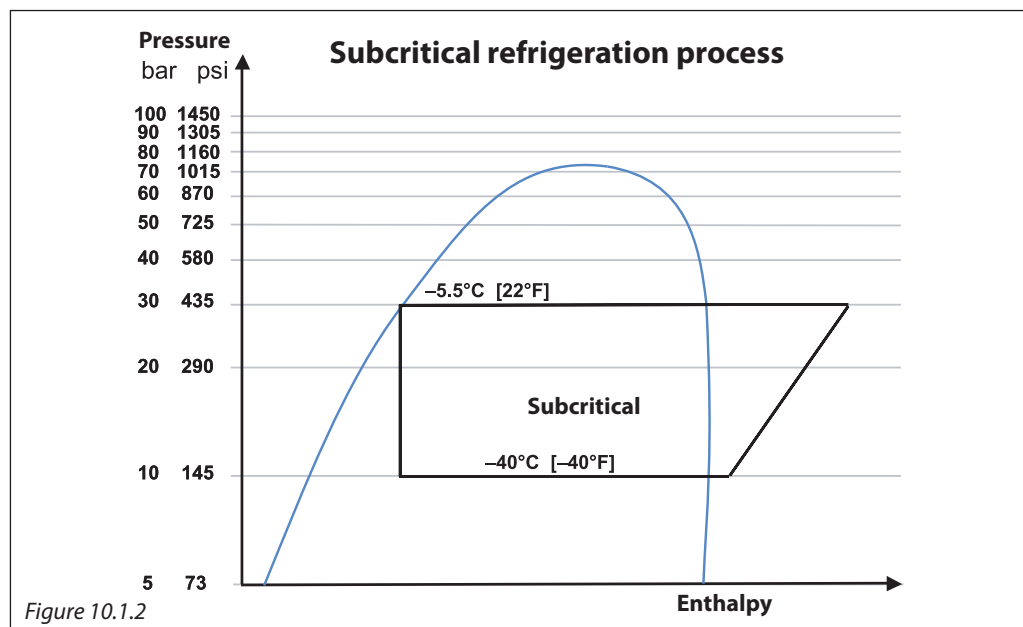
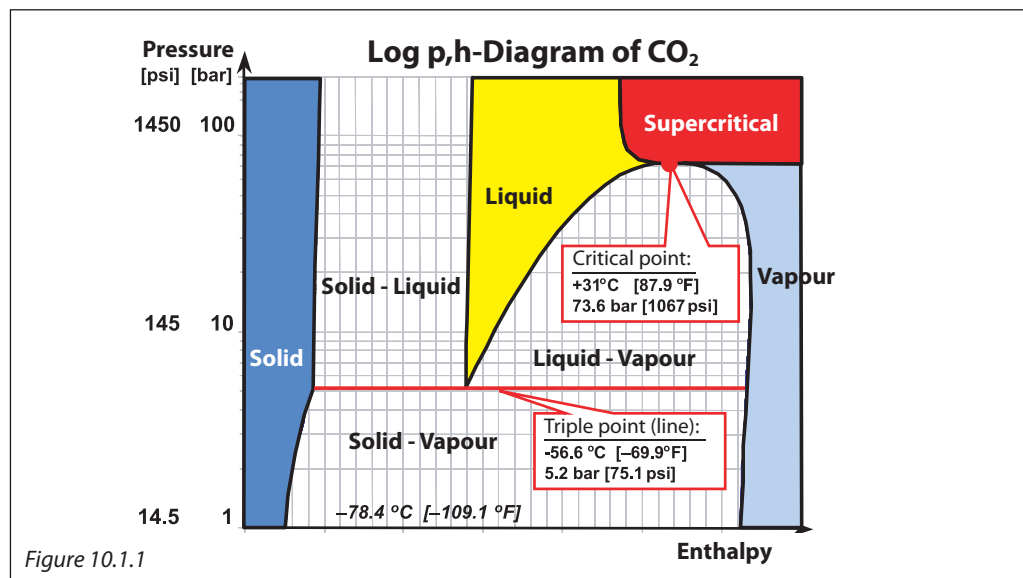
10.1 CO₂ as a refrigerant

CO₂ may be employed as a refrigerant in a number of different system types, including both subcritical and supercritical. For any type of CO₂ system, both the critical point and the triple point must be considered.

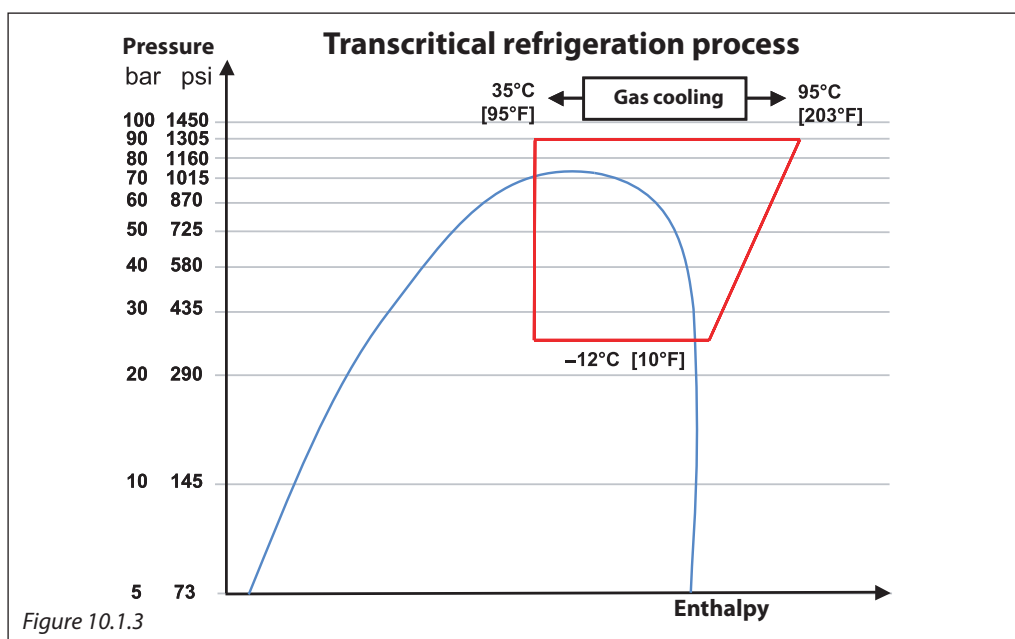
The classic refrigeration cycle we are all familiar with is subcritical, i.e., the entire range of temperatures and pressures are below the critical point and above the triple point. A single-stage subcritical CO₂ system is simple, but it also has disadvantages because of its limited temperature range and high pressure (figure 10.1.2).

Transcritical CO₂ systems are at presently only of interest for small and commercial applications, e.g., mobile air conditioning, small heat pumps, and supermarket refrigeration, but not for industrial systems (figure 10.1.3). Transcritical systems are not described in this handbook.

Operating pressures for subcritical cycles are usually in the range 5.7 to 35 bar [83 to 507 psi], corresponding to -55 to 0°C [-67 to 32°F]. If the evaporators are defrosted using hot gas, then the operating pressure is approximately 10 bar [145 psi] higher.



10.1 CO₂ as a refrigerant (Continued)

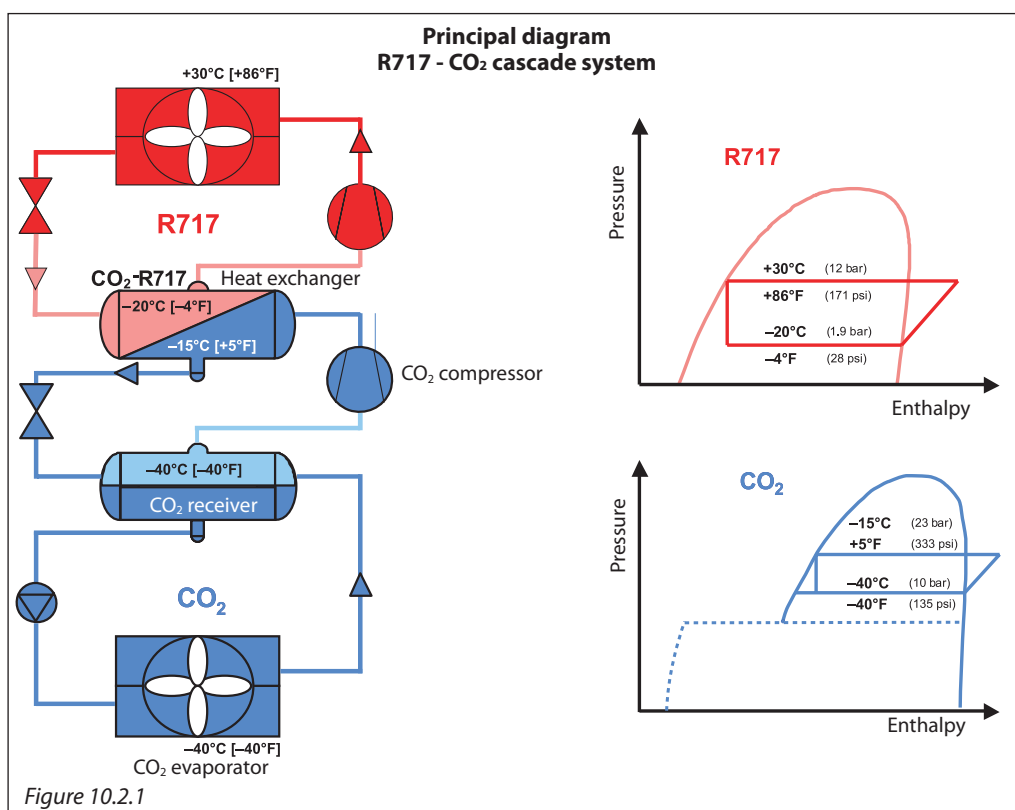


CO₂ is most commonly used in cascade or hybrid system designs in industrial refrigeration, because its pressure can be limited to such extent that commercially available components like compressors, controls and valves can be used.

CO₂ cascade systems can be designed in different ways, e.g., direct expansion systems, pump circulating systems, CO₂ in volatile secondary "brine" systems, or combinations of these.

10.2 CO₂ as a refrigerant in industrial systems

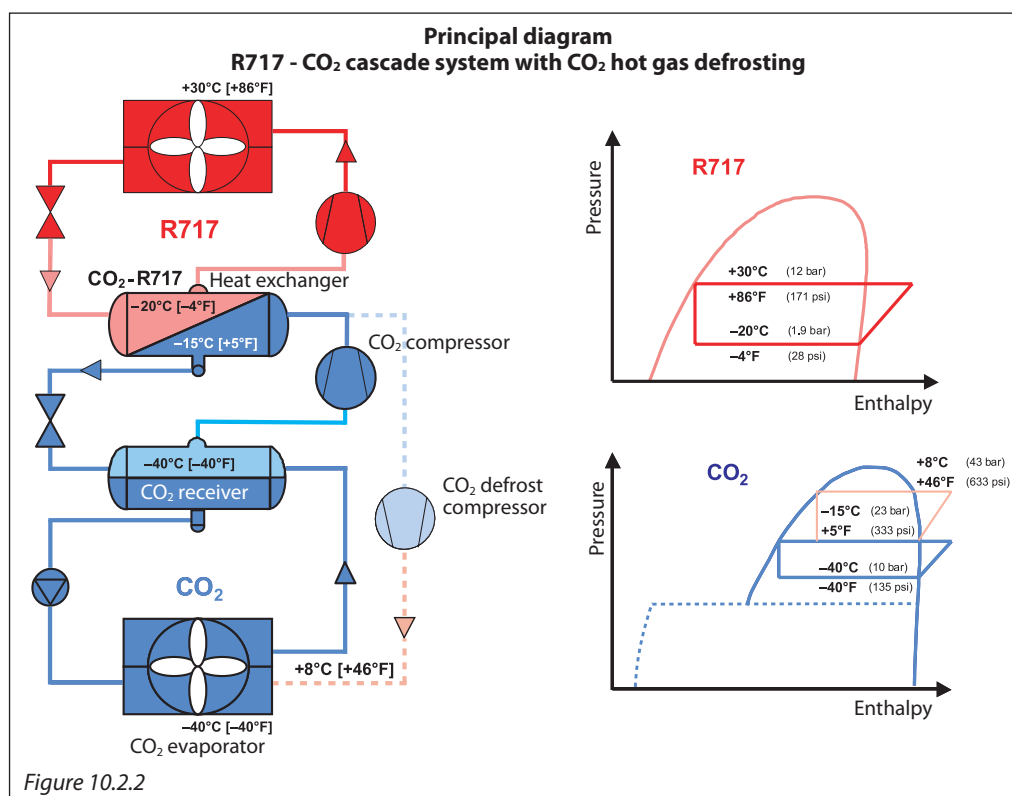
Figure 10.2.1 shows a low temperature refrigerating system -40°C [-40°F] using CO₂ as a phase change refrigerant in a cascade system with ammonia on the high-pressure side.



10.2

CO₂ as a refrigerant in industrial systems

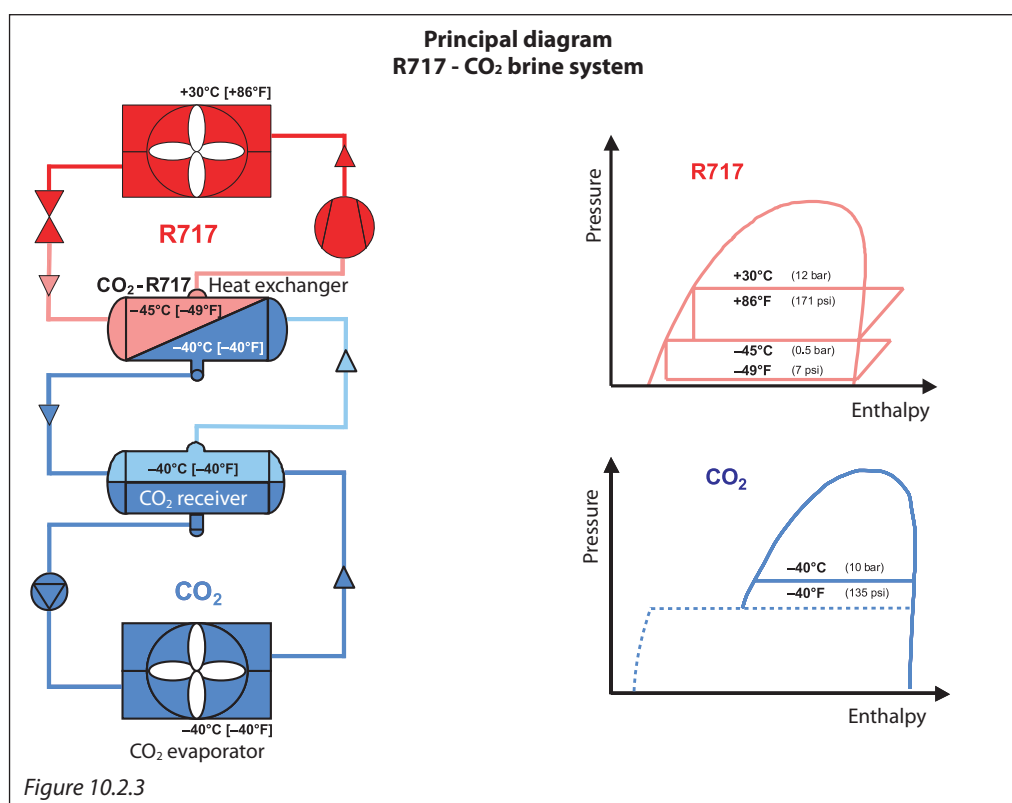
(Continued)



The CO₂ system is a pump circulating system where the liquid CO₂ is pumped from the receiver to the evaporator, where it is partly evaporated, before it returns to the receiver. The evaporated CO₂ is then compressed in a CO₂ compressor, and condensed in the CO₂-NH₃ heat exchanger. The heat exchanger acts as an evaporator in the

NH₃ system. Compared to a traditional ammonia system, the ammonia charge in the above mentioned cascade system can be reduced by a factor of approximately 10.

Figure. 10.2.2 shows the same system as in figure 10.2.1, but includes a CO₂ hot gas defrosting system.



10.2 CO₂ as a refrigerant in industrial systems (Continued)

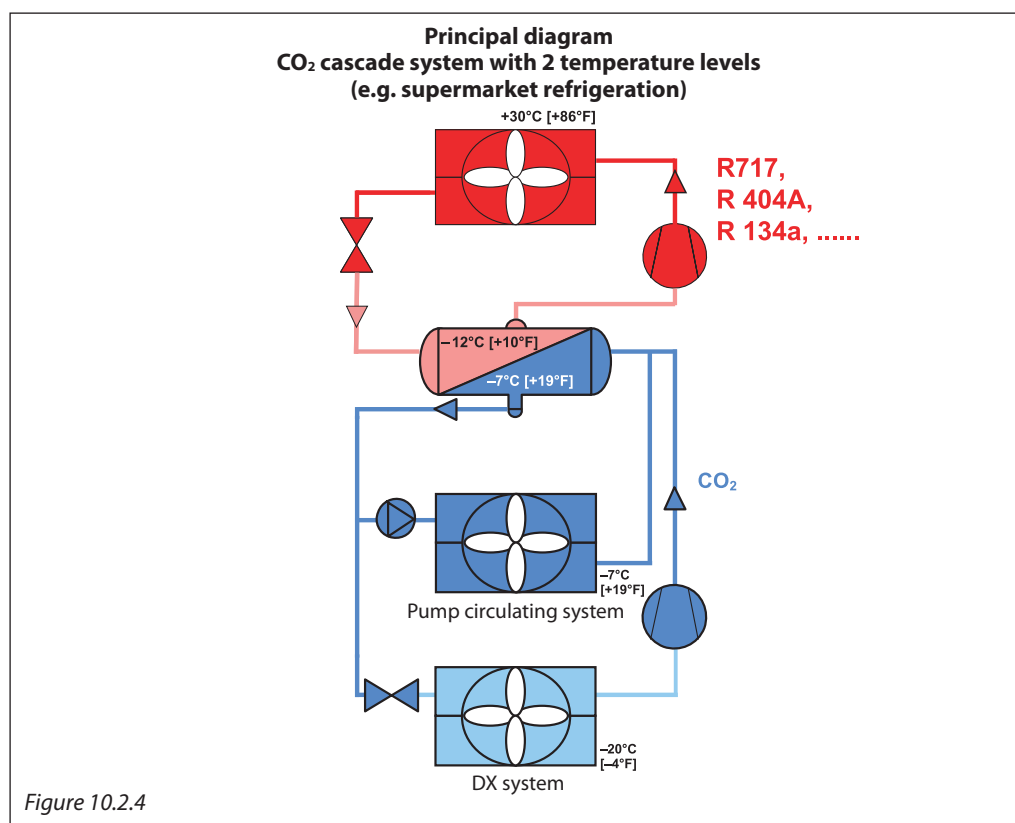


Figure 10.2.3 shows a low temperature refrigerating system -40°C [-40°F] using CO₂ as a “brine” system with ammonia on the high-pressure side. The CO₂ system is a pump circulating system, where the liquid CO₂ is pumped from the receiver to the evaporator. Here it is partly evaporated, before it returns to the receiver.

The evaporated CO₂ is then condensed in the CO₂- NH₃ heat exchanger. The heat exchanger acts as an evaporator in the NH₃ system. Figure 10.2.4 shows a mixed system with flooded and DX-system, e.g. for a refrigeration system in a supermarket, where 2 temperature levels are required

10.3 Design pressure

When determining the design pressure for CO₂ systems, the two most important factors to consider are:

- Pressure during *stand still*
- Pressure required during defrosting

Importantly, without any pressure control, at *stand still*, i.e., when the system is turned off, the system pressure will increase due to heat gain from the ambient air. If the temperature were to reach 0°C [32°F], the pressure would be 34.9 bar [505 psi] or 57.2 bar [830 psi] @ 20°C [68°F]. For industrial refrigeration systems, it would be quite expensive to design a system that can withstand the equalizing pressure (i.e., saturation pressure corresponding to the ambient temperature) during *stand still*. Therefore, installing a small auxiliary condensing unit is a common way to limit the maximum pressure during *stand still* to a reasonable level, e.g., 30 bar [435 psi].

With CO₂, many different ways of defrosting can be applied (e.g., natural, water, electrical, hot gas). Hot gas defrosting is the most efficient, especially at low temperatures, but also demands the highest pressure. With a design pressure of 52 bar-g [754 psig], it is possible to reach a defrosting temperature of approx. 10°C [50°F].

The saturated pressure at 10°C [50°F] is 45 bar [652 psi]. By adding 10% for the safety valves and approximately 5% for pressure peaks, the indicated maximum allowable working pressure would be ~ 52 barg [~754 psig] (figure 10.3.2 & 10.3.3).

10.3 Design pressure (Continued)

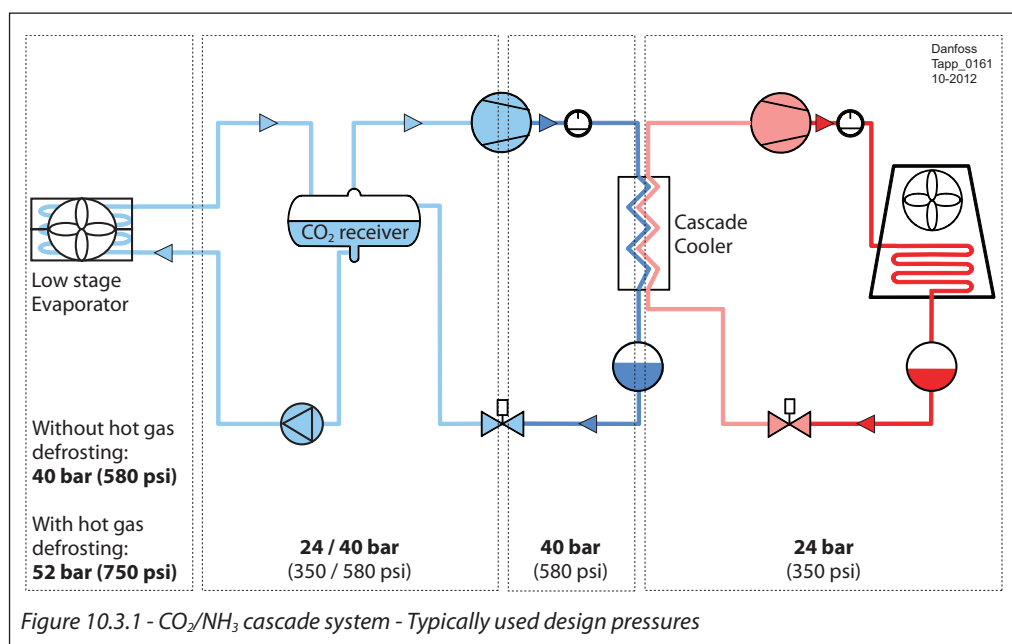


Figure 10.3.1 - CO₂/NH₃ cascade system - Typically used design pressures

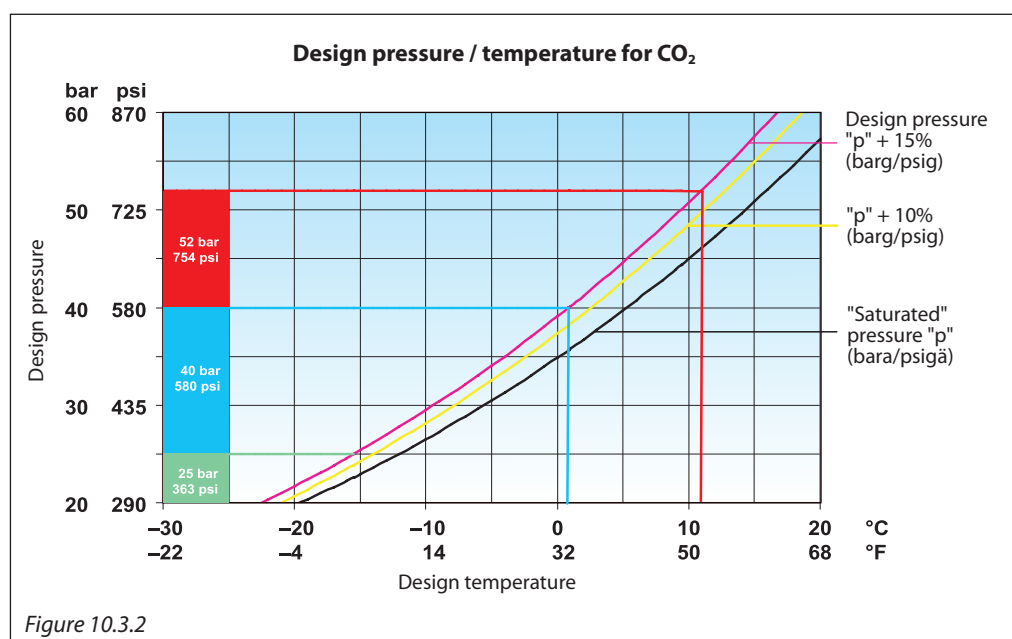


Figure 10.3.2

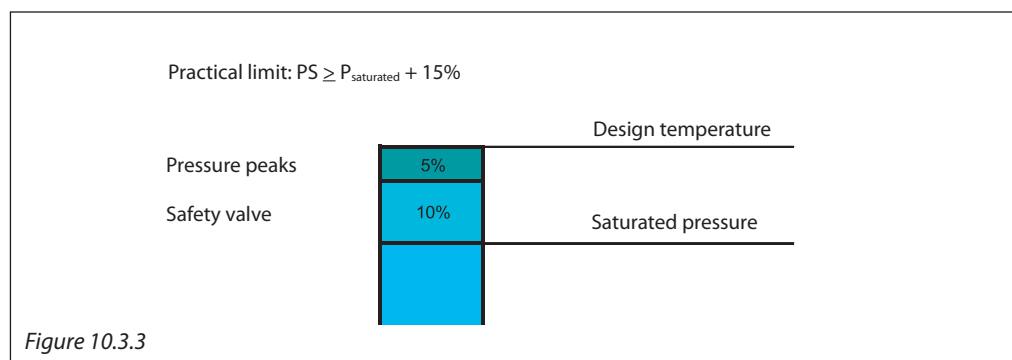


Figure 10.3.3

10.4 Safety

CO₂ is an odourless, colourless substance classified as a non-flammable and non-toxic refrigerant, but even though all the properties seem very positive, CO₂ also has some disadvantages.

Due to the fact that CO₂ is odourless, it is not self-alarming if leaks occur (ref. [6]).

CO₂ is heavier than air, so it sinks to the ground or floor level. This can create dangerous situations, especially in pits or confined spaces. CO₂ can displace oxygen so much that the resulting mixture is lethal. The relative density of CO₂ is 1.529 (air=1 @ 0°C [32°F]). This risk requires special attention during design and operation. Leak detection and / or emergency ventilation are always necessary.

Compared to ammonia, CO₂ is a safer refrigerant. The TLV (threshold limit value) is the maximum

concentration of vapour CO₂ in air, which can be tolerated over an eight-hour shift for 40 hours a week. The TLV safety limit is 25 ppm for ammonia and 5000 ppm (0.5%) for CO₂.

Approx. 0.04% CO₂ is present in the air. With higher concentration, some adverse reactions are reported:

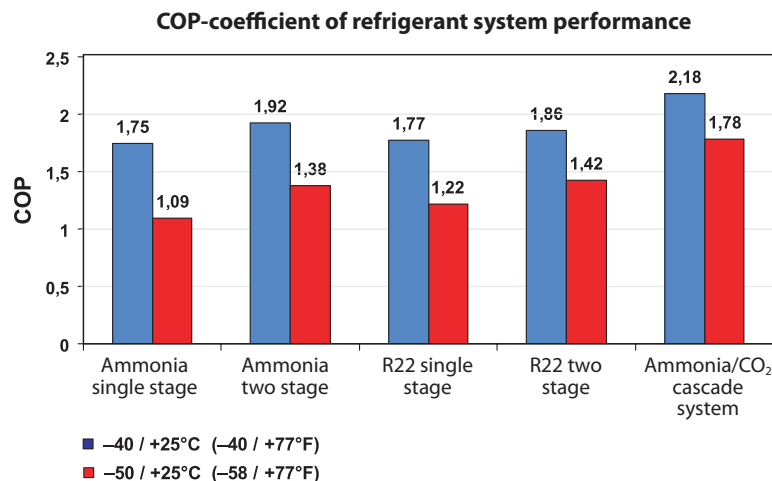
2%	50% increase in breath rate
3%	100% increase in breath rate
5%	300% increase in breath rate
8-10%	Natural respiration is disrupted and breathing becomes almost impossible. Headache, dizziness, sweating and disorientation.
> 10%	Can lead to loss of consciousness and death.
> 30%	Quickly leads to death.

10.5 Efficiency

In CO₂- NH₃ cascade systems it is necessary to use a heat exchanger. Using heat exchangers reduces system efficiency, due to the necessity of having a temperature difference between the fluids. However, compressors running with CO₂ have a

better efficiency and heat transfer is greater. The overall efficiency of a CO₂- NH₃ cascade system is not reduced when compared to a traditional NH₃ system (figure 10.5.1 & ref. [3]).

Example:



Source: IAR Albuquerque, New Mexico 2003, P.S Nielsen & T.Lund
Introducing a New Ammonia/CO₂ Cascade Concept for Large Fishing Vessels

Figure 10.5.1

10.6 Oil in CO₂ systems

In CO₂ systems with traditional refrigeration compressors, both miscible and immiscible oil types are used (see the table below).

For immiscible lubricants, such as polyalphaolefin (PAO), the lubricant management system is relatively complicated. The density of PAO is lower than the density of the liquid CO₂. The lubricant therefore floats on top of the refrigerant, making it more difficult to remove than in ammonia systems. Also, to avoid fouling evaporators, compressor oil separation with non- miscible oils must be highly effective; basically, a virtually oil-free system is desirable.

With miscible lubricants, such as polyol ester (POE), the oil management system can be much simpler. POE oils have high affinity with water, so the challenge when using POE is to ensure the stability of the lubricant.

In *volatile brine* systems using CO₂ as a secondary refrigerant, and in recirculating systems with oil free compressors, no oil is present in the circulated CO₂. From an efficiency point of view, this is optimum because it results in good heat transfer coefficients in the evaporators. However, it requires that all valves, controls and other components can operate *dry*.

CO₂ and oil

Oil type	PAO Poly-alpha-olefin oil (synthetic oil)	POE Polyol ester oil (ester oil)
Solubility	Low (immiscible)	High (miscible)
Hydrolysis	Low	High affinity to water
Oil separation system	Special requirements: <ul style="list-style-type: none"> High filtration performance Multistage coalescing filters Active carbon filter 	No special requirements (System requirements like HCFC/HFC)
Oil return system	Special requirement: <ul style="list-style-type: none"> Oil drain from low temperature receiver (oil density lower than CO₂ - opposite of NH₃) 	Simple (System requirements like HCFC/HFC)
Challenge	<ul style="list-style-type: none"> Oil separation and return system Long term oil accumulation in e.g. evaporators 	<ul style="list-style-type: none"> High affinity to water Long term stability of oil "Clean" refrigerant system required

10.6 Oil in CO₂ systems (Continued)

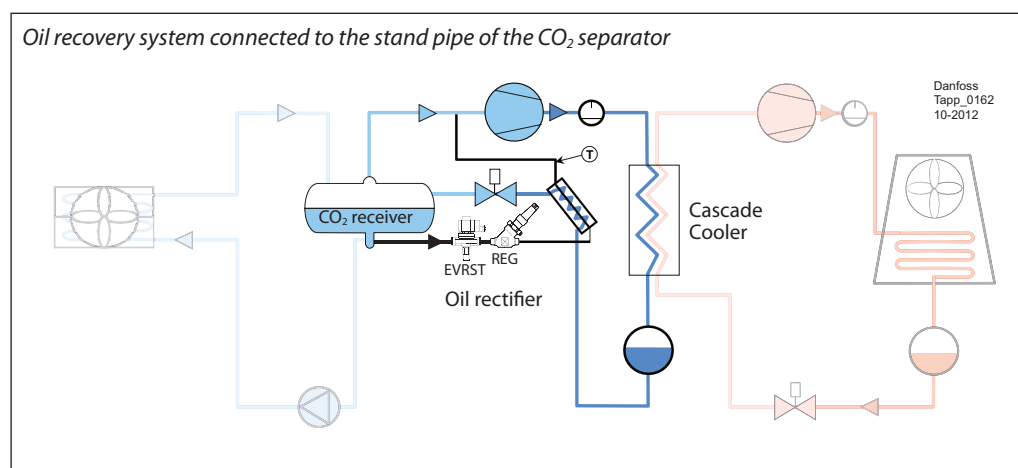
The oil concentration in the pump separator increases gradually because the oil cannot be directly sucked back to the compressor with the gas. If the oil concentration in the evaporator becomes too high, the adhesive forces will make the oil "stick" to the heat transferring surfaces. This reduces the capacity of the plant.

Pure CO₂ liquid must never be returned back to compressor as this will damaged the compressor, therefor it is imperative that the CO₂ at the heat exchanger outlet is superheated.

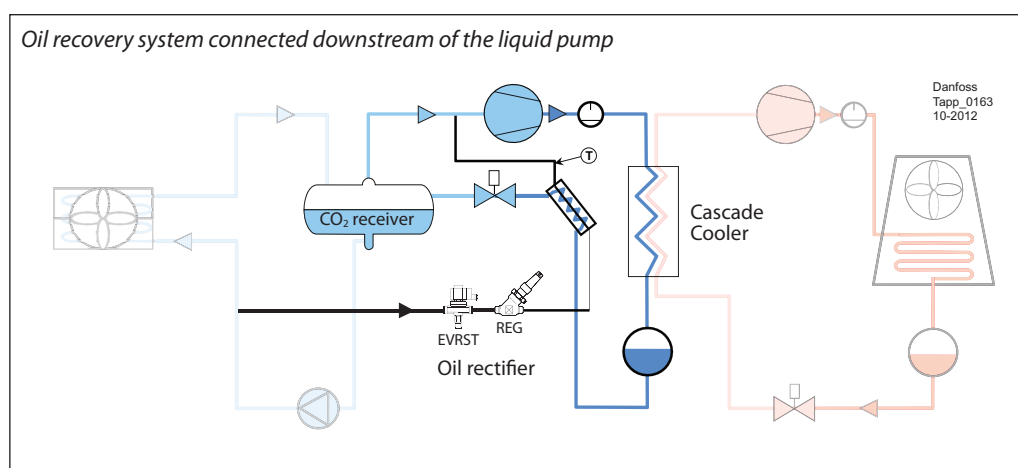
The superheat can be controlled by a REG valve fitted down streams of the solenoid valve.

By constantly boiling of part of the oil/CO₂ liquid from the pump separator, the oil concentration in the plant remains low. During the boiling process in the oil rectifier, the CO₂ liquid is sub-cooled and the oil/CO₂ liquid mixture from the CO₂ separator is boiled off and sucked back to the CO₂ compressor.

*Example 10.6.1
Oil management system
for systems with soluble
(miscible) oils*



*Example 10.6.2
Oil management system
for systems with soluble
(miscible) oils*

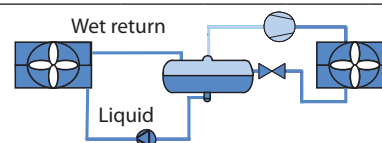








Not all valves are shown.
Not to be used for construction
purposes.

10.7 Comparison of component requirements in CO₂, ammonia and R134a systems

Compared to ammonia and R134a, CO₂ differs in many respects. The following comparison illustrates this fact; to allow an "true" comparison, operating conditions such as evaporating temperature, condensing temperature, are kept constant.

Comparison of pipe cross section area Wet return / Liquid lines

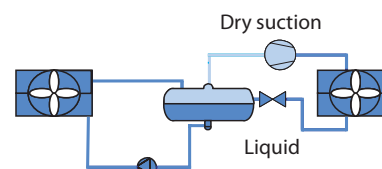








Refrigerant				R 134a	R 717	CO ₂
Capacity				250 [71]	250 [71]	250 [71]
"Wet return" line	ΔT	K [F]		0.8 [1.4]	0.8 [1.4]	0.8 [1.4]
	Δp	bar [psi]		0.0212 [0.308]	0.0303 [0.439]	0.2930 [4.249]
	Velocity	m/s [ft/s]		11.0 [36.2]	20.2 [66.2]	8.2 [26.9]
						
Diameter				215 [8.5]	133 [5.2]	69 [2.7]
"Wet return" area				36385 [56.40]	13894 [21.54]	3774 [5.85]
"Liquid" line	Velocity	m/s [ft/s]		0.8 [2.6]	0.8 [2.6]	0.8 [2.6]
						
	Diameter	mm [inch]		61 [2.4]	36 [1.4]	58 [2.3]
	"Liquid" area	mm ² [inch ²]		2968 [4.6]	998 [1.55]	2609 [4.04]
Total pipe cross section area	"Wet return" + "liquid" area	mm ² [inch ²]		39353 [61.0]	14892 [23.08]	6382 [9.89]
Liquid cross section area				8	7	41

L_{eqv} = 50 [m] / 194 [ft] - Pump circ.: n_{circ} = 3 - Evaporating temp.: TE = -40[°C] / -40[°F]

Table 1

Comparison of pipe cross section area Dry suction / Liquid lines



Refrigerant				R 134a	R 717	CO ₂
Capacity				250 [71]	250 [71]	250 [71]
"Dry suction" line	ΔT	K [F]		0.8 [1.4]	0.8 [1.4]	0.8 [1.4]
	Δp	bar [psi]		0.0212 [0.308]	0.0303 [0.439]	0.2930 [4.249]
	Velocity	m/s [ft/s]		20.4 [67]	37.5 [123]	15.4 [51]
						
Diameter				168 [6.6]	102 [4.0]	53 [2.1]
"Dry suction" area				22134 [34.31]	8097 [12.55]	2242 [3.48]
"Liquid" line	Velocity	m/s [ft/s]		0.8 [2.6]	0.8 [2.6]	0.8 [2.6]
						
	Diameter	mm [inch]		37 [1.5]	21 [0.8]	35 [1.4]
	"Liquid" area	mm ² [inch ²]		1089 [1.69]	353 [0.55]	975 [1.51]
Total pipe cross section area	"Dry suction + liquid" area	mm ² [inch ²]		23223 [36.00]	8450 [13.10]	3217 [4.99]
Liquid cross section area				5	4	30

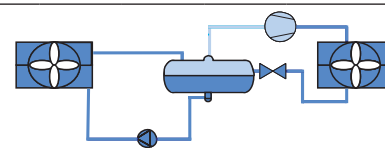
L_{eqv} = 50 [m] / 194 [ft] - Evaporating temp.: TE = -40[°C] / -40[°F] - Condensing temp.: TE = -15[°C] / -5[°F]

Table 2

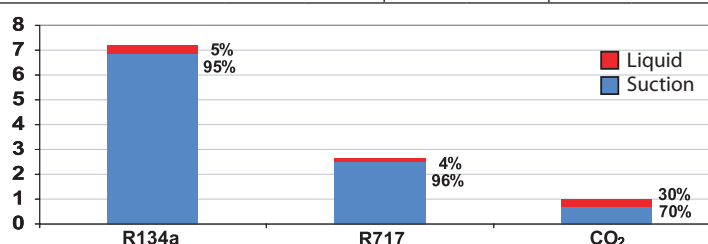
10.7

Comparison of component requirements in CO₂, ammonia and R134a systems (Continued)

Comparison of pipe cross section area Dry suction / Liquid lines



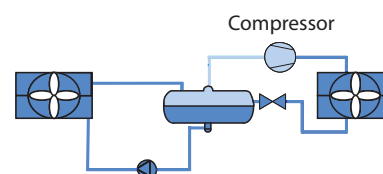
Refrigerant			R 134a	R 717	CO ₂
Capacity	kW	[TR]	250 [71]	250 [71]	250 [71]
"Dry suction" line	"Dry suction" area	mm ² [inch ²]	22134 [34.31]	8097 [12.55]	2242 [3.48]
"Liquid" line	"Liquid" area	mm ² [inch ²]	1089 [1.69]	353 [0.55]	975 [1.51]
Total pipe cross section area	"Dry suction + liquid" area	mm ² [inch ²]	23223 [36.00]	8450 [13.10]	3217 [4.99]
Relative cross section area	-	-	7.2	2.6	1.0
Liquid cross section area	-	%	5	4	30
Vapour cross section area	-	%	95	96	70



L_{eqv} = 50 [m] / 194 [ft] - Evaporating temp.: TE = -40[°C] / -40[°F] - Condensing temp.: TE = -15[°C] / -5[°F]

Table 3

Comparison of compressor displacement

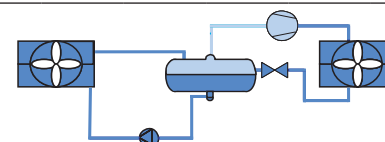


Refrigerant			R 134a	R 717	CO ₂
Refrigerant capacity	kW	[TR]	250 [71]	250 [71]	250 [71]
Required compressor displacement	m ³ /h	[ft ³ /h]	1628 [57489]	1092 [38578]	124 [4387]
Relative displacement	-	-	13.1	8.8	1.0

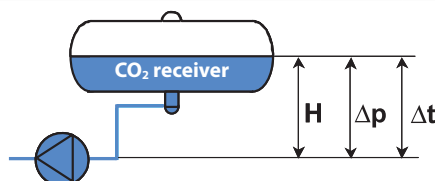
Evaporating temp.: TE = -40[°C] / -40[°F] - Condensing temp.: TE = -15[°C] / -5[°F]

Table 4

Comparison of pressure / subcooling produced in liquid risers



Refrigerant			R 134a	R 717	CO ₂
Hight of liquid riser "H"	m	[ft]	3 [9.8]	3 [9.8]	3 [9.8]
Pressure produced in liquid riser "Δp"	bar	[psi]	0.418 [6.06]	0.213 [2.95]	0.329 [4.77]
Subcooling produced in liquid riser "Δt"	K	[°F]	14.91 [26.8]	5.21 [9.4]	0.88 [1.6]



Evaporating temp.: TE = -40[°C] / -40[°F]

Table 5

Wet return lines in recirculation systems:

A comparison of pump circulating systems shows that for "wet return" lines, CO₂ systems require much smaller pipes than ammonia or R134a (table 3). In CO₂ "wet return" lines, the allowable pressure drop for an equivalent temperature drop is approximately 10 times higher than

for ammonia or R134a wet return lines. This phenomenon is a result of the relatively high density of the CO₂ vapor. The above comparison is based on a circulating rate of 3. The results are slightly different if the circulating rate is optimized for each refrigerant.

Suction lines in dry expansion systems:

In the comparison of "dry suction" lines, the results are very nearly the same as in the previous comparison, in terms of both pressure drop and line size (table 2).

Liquid lines:

For both recirculating and dry expansion systems, calculated sizes for CO₂ liquid lines are much larger than those for ammonia, but only slightly larger than those for R134a (table 1 and 2). This can be explained by ammonia's much larger latent heat relative to CO₂ and R134a. With reference to the table showing the relative liquid and vapor cross-sectional areas for the three refrigerants (table 1), the total cross-section area for the CO₂ system is approximately 2.5 times smaller than that of an ammonia system and approximately seven times smaller than that of R134a. This result has interesting implications for the relative installation costs for the three refrigerants. Due to the relative small vapor volume of the CO₂ system and large volumetric refrigeration capacity, the CO₂ system is relatively sensitive to capacity fluctuations. It is therefore important to design the liquid separator with sufficient volume to compensate for the small vapor volume in the pipes.

The required compressor capacity for identical refrigeration loads is calculated for the three refrigerants (table 4). As can be seen, the CO₂ system requires a much smaller compressor than the ammonia or R134a systems.

For compressors of identical displacement, the capacity of the compressor capacity for CO₂ is 8.8 times greater than for ammonia and 13 times greater than for R134a.

The subcooling produced in a liquid riser of a given height "H" is calculated for the three refrigerants (table 5). The subcooling for the CO₂ liquid riser is much smaller than that for ammonia and R134a. This characteristic must be noted when designing CO₂ systems in order to prevent cavitation and other problems with liquid CO₂ pumps.

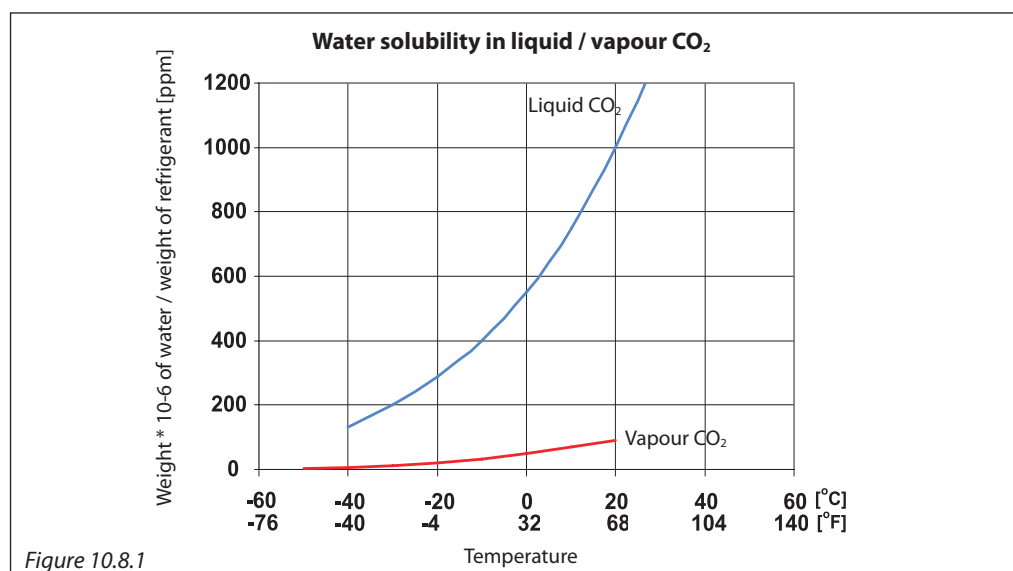
10.8 Water in CO₂ Systems

In ammonia systems, the oil is changed regularly and non-condensibles are purged frequently to minimise the accumulation of oil, water and solid contaminants that can cause problems.

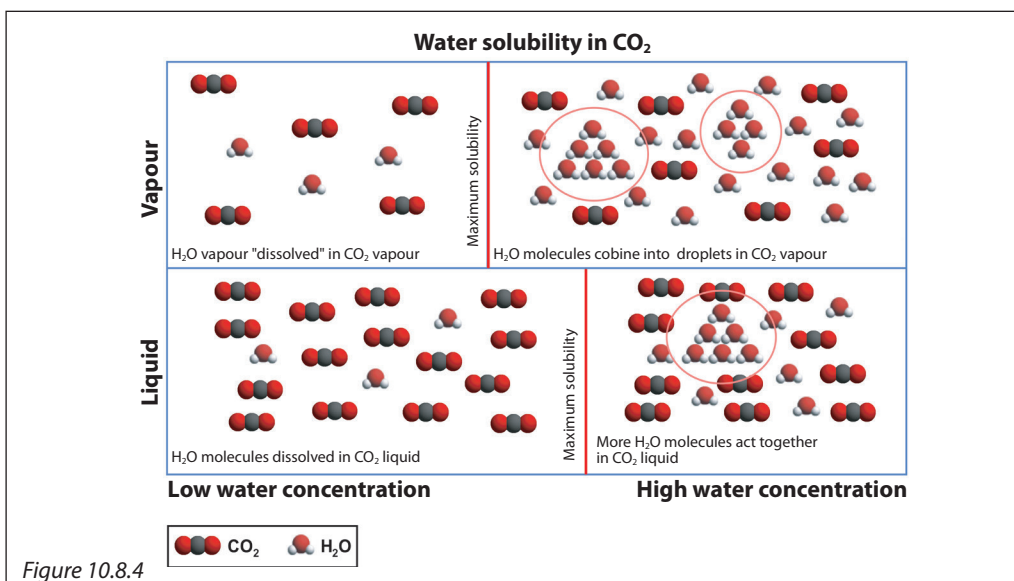
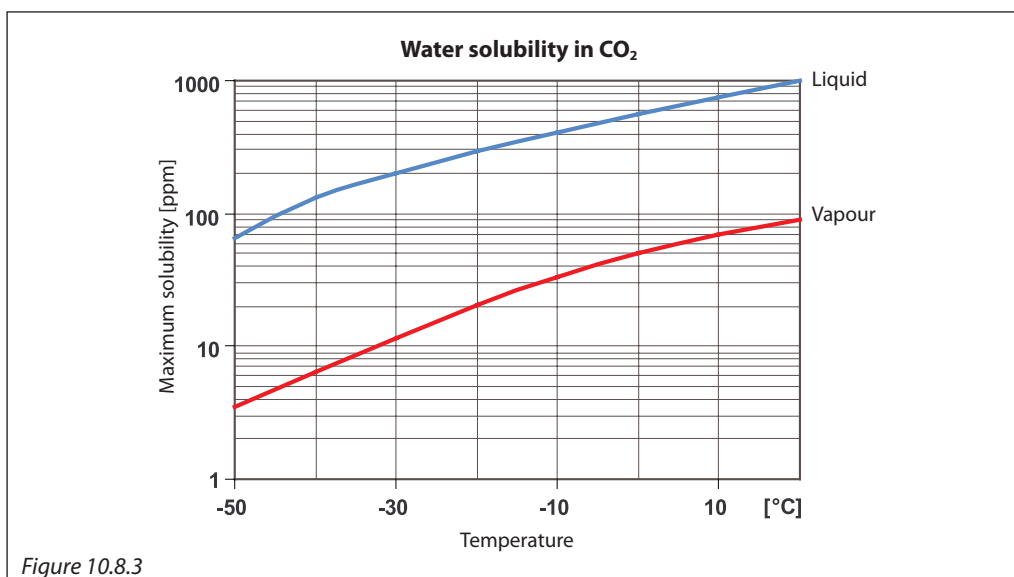
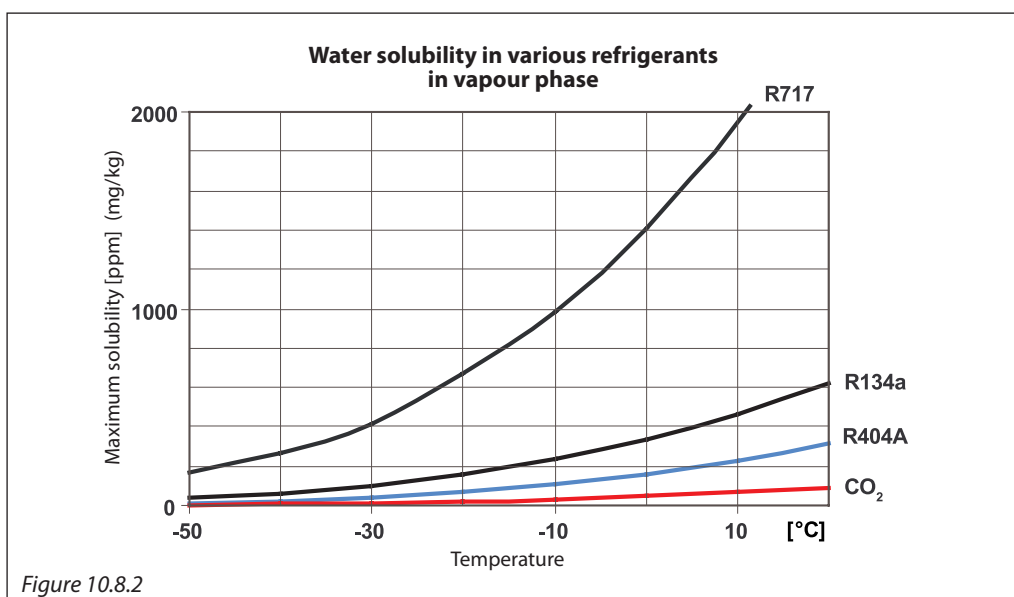
Compared to ammonia systems, CO₂ is less sensitive, but if water is present, problems may occur. Some early CO₂ installations reported problems with control equipment, among other components. Investigations revealed that many of these problems were caused by water freezing in the system. Modern systems use filter driers to

maintain the water content in the system at an acceptable level.

The acceptable level of water in CO₂ systems is much lower than with other common refrigerants. The diagram in figure 10.8.1 shows the solubility of water in both the liquid and vapor phases of the CO₂ liquid and vapor as a function of temperature. The solubility in the liquid phase is much higher than in the vapor phase. The solubility in the vapor phase is also known as the *dew point*.



10.8 Water in CO₂ Systems (Continued)



10.8 Water in CO₂ Systems (Continued)

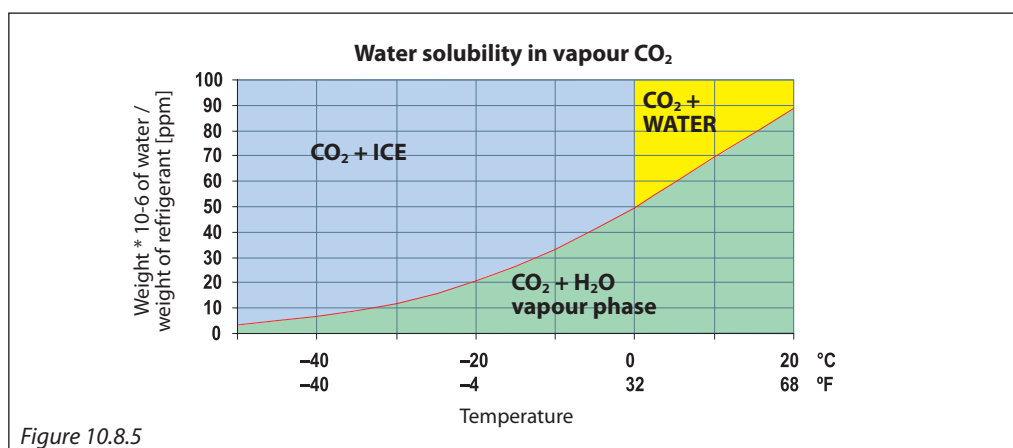


Figure 10.8.5

The diagram in figure 10.8.1 shows that the water solubility in CO₂ is much lower than for R134a or ammonia. At -20°C [-4°F], water solubility in the liquid phase is:

- 20.8 ppm for CO₂
- 158 ppm for R134a
- 672 ppm for ammonia

Below these levels, water remains dissolved in the refrigerant and does not harm the system. As illustrated in figure 10.8.4, water (H₂O) molecules are dissolved if the concentration is lower than the maximum solubility limit, but they

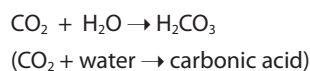
precipitate out of solution into droplets if the water concentration is higher than the maximum solubility limit.

If the water is allowed to exceed this limit in a CO₂ system, problems may occur, especially if the temperature is below 0°C. In this case, the water will freeze, and the ice crystals can block control valves, solenoid valves, filters and other equipment (figure 10.8.5). This problem is especially significant in flooded and direct expansion CO₂ systems, but less so in volatile secondary systems because less sensitive equipment is used.

Chemical reactions

It should be noted that the reactions described below do not occur in a well-maintained CO₂ system, where the water content is below the maximum solubility limit.

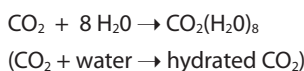
In a closed system such as a refrigeration system, CO₂ can react with oil, oxygen, and water, especially at elevated temperatures and pressures. For example, if the water content is allowed to rise above the maximum solubility limit, CO₂ can form carbonic acid, as follows (ref. [4] and [5]):



In CO₂ production systems, where water concentrations can rise to high levels, it is well known that carbonic acid can be quite corrosive to several kinds of metals, but this reaction does not take place in a well-maintained CO₂ system, because the water content in the system is kept below the maximum solubility limit.

Water in vapor phase

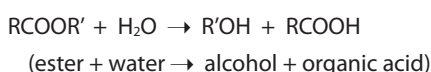
If the water concentration is relatively high, CO₂ and water in vapor phase can react to form a CO₂ gas hydrate:



The CO₂ gas hydrate is a large molecule and can exist above 0°C [32°F]. It can create problems in control equipment and filters, similar to the problems due to ice.

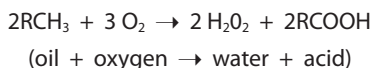
POE lubricant

Generally, esters such as POE react with water as follows:



As shown, if water is present POE will react with water to form alcohol and an organic acid (carboxylic acid), which is relatively strong and may corrode the metals in the system. It is therefore essential to limit the water concentration in CO₂ systems if POE lubricants are used.

PAO lubricant



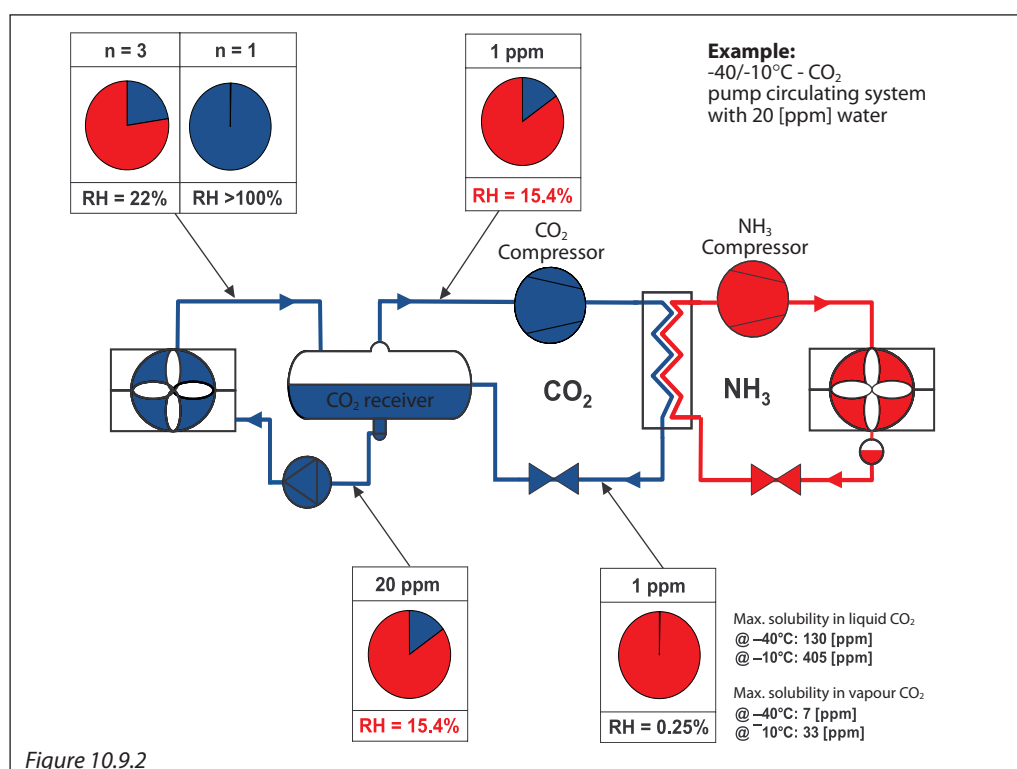
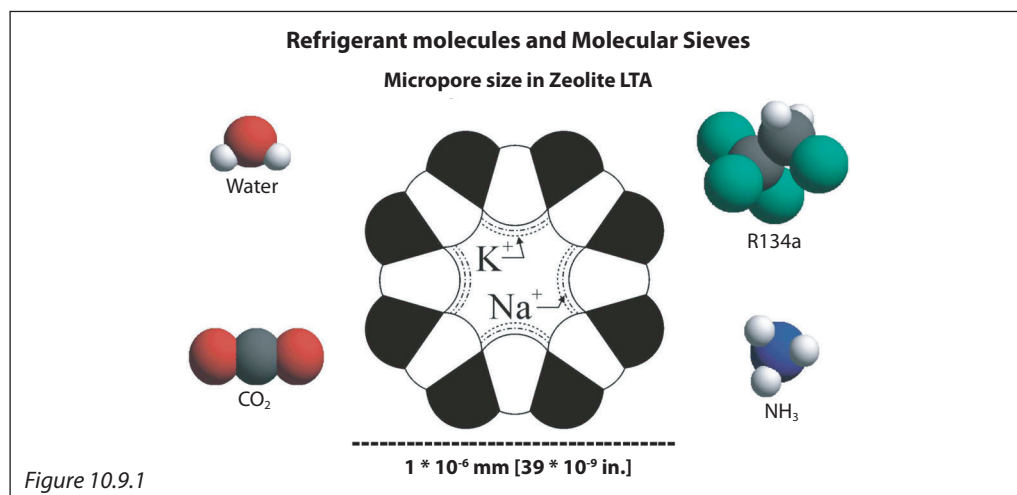
PAO lubricant is also called synthetic oil. Ordinarily, PAO is very stable. However, if sufficient free oxygen is present, such as might be available from corrosion in pipes, the oxygen will react with the lubricant to form carboxylic acid.

10.9 Removing water

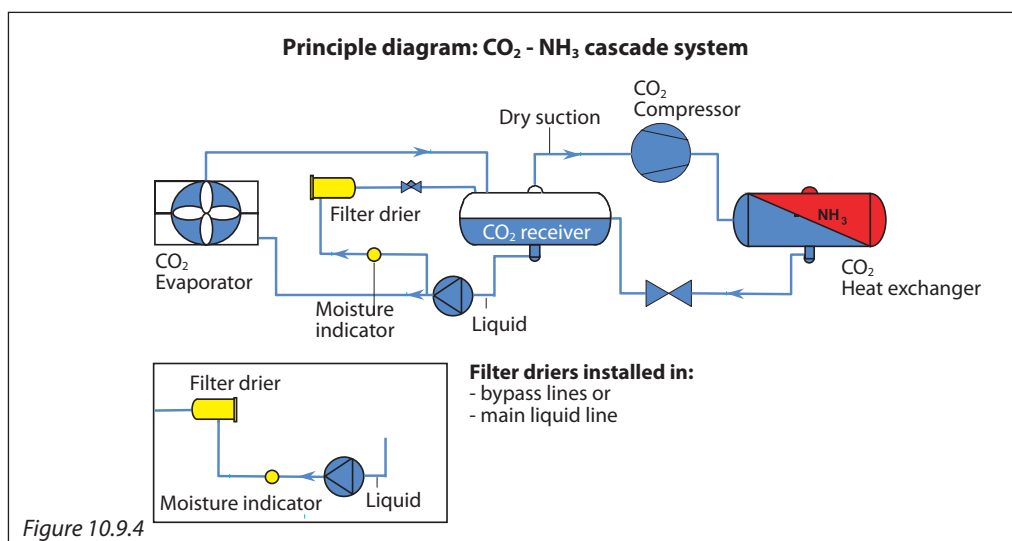
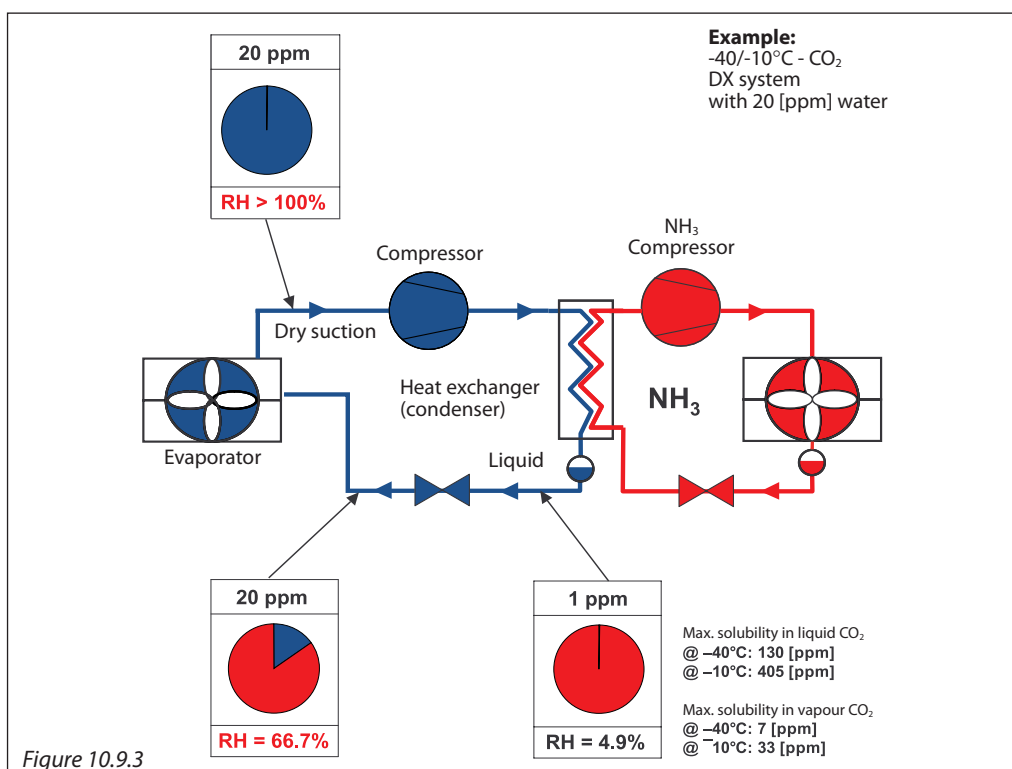
Controlling the water content in a refrigeration system is a very effective way to prevent the above-mentioned chemical reactions.

In Freon systems, filter driers are commonly used to remove water, usually the type with a zeolite core. The zeolite has extremely small pores, and acts like a molecular sieve (figure 10.9.1).

Water molecules are small enough to pass through the sieve, and being very polar, are adsorbed on the zeolite molecules. R134a molecules are too large to penetrate the sieve. When the replaceable core is removed, the water goes with it.



10.9 Removing water (Continued)



CO₂ is a non-polar molecule, so the removal process is different. Like water molecules, CO₂ molecules are small enough to pass through the molecular sieve. However, the water molecules adsorbed on the molecular sieve tend to displace the CO₂ molecules, due to the difference in polarity. Zeolite filter driers cannot be used in ammonia systems, because both water and ammonia are very polar. Even though the driers function differently in this respect in CO₂ systems, the efficiency is fairly good. The water retention capacity is approximately the same as in R134a systems.

The most effective location to detect and remove water is where the concentration is high. The solubility of water in CO₂ is much lower in the vapor phase than in the liquid phase, so more water can be transported in liquid lines.

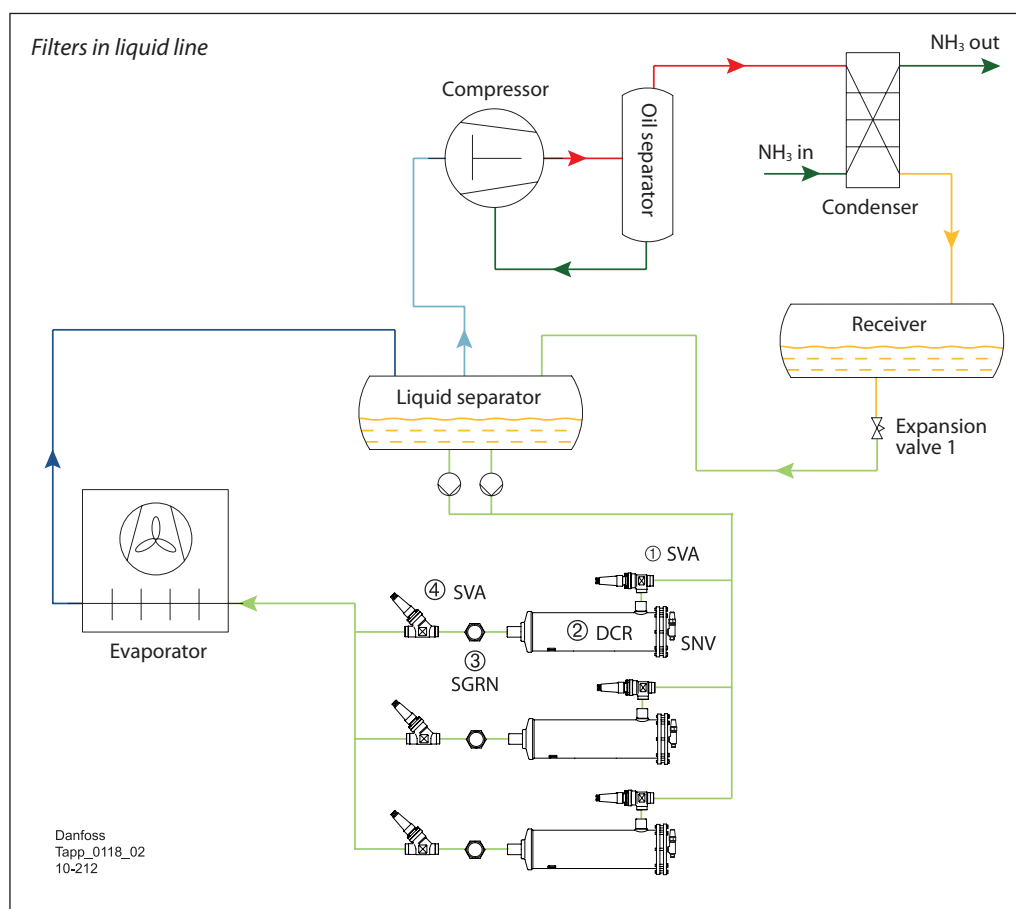
Fig. 10.9.2 illustrates the variation of the relative humidity in a pump circulation system operating at -40°C. The illustration shows that the relative humidity is highest in the wet return line, and that it depends on the circulating rate. In a DX system the variation of the relative humidity differs, but also in this case the highest concentration is located in the suction line (fig. 10.9.3).

Taking advantage of this principle, moisture indicators and filter driers are typically installed in a liquid line or liquid bypass line from the receiver (figure 10.9.4 and figure 10.9.5).

Application example 10.9.5:
Filter driers in CO₂ pumped
liquid circulation systems

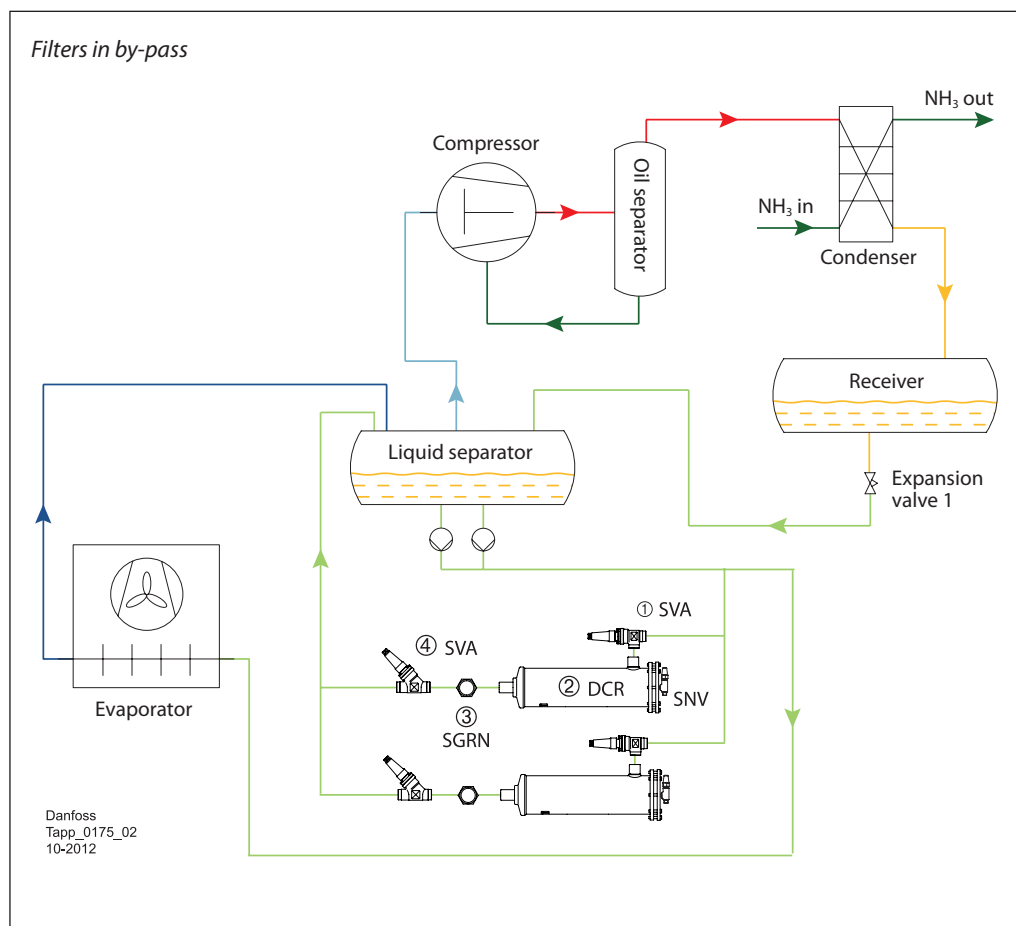
- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- LP liquid refrigerant
- Oil

- ① Stop valve
- ② Filter drier
- ③ Sight glass
- ④ Stop valve



- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- LP liquid refrigerant
- Oil

- ① Stop valve
- ② Filter drier
- ③ Sight glass
- ④ Stop valve



Not all valves are shown.
Not to be used for construction
purposes.

*Application example 10.9.6:
Filter driers in CO₂ pumped
liquid circulation systems
(Continued)*

To install a filter drier in a CO₂ system, the following criteria should be considered:

- **Relative Humidity**
The relatively humidity should be high.
- **Pressure Drop**
The pressure drop across the filter drier should be small. And the system performance should not be sensitive to this pressure drop.
- **Two Phase Flow**
Two phase flow through the filter drier should be avoided, which brings risk of freezing and blocking because of the unique water solubility characteristics.

In a CO₂ pumped liquid circulation systems, filter driers are recommended to be installed on the liquid lines before evaporators. On these lines, RH is high, there is no two phase flow, and it's not sensitive to pressure drop.

Installation in other positions is not recommended for the following reasons:

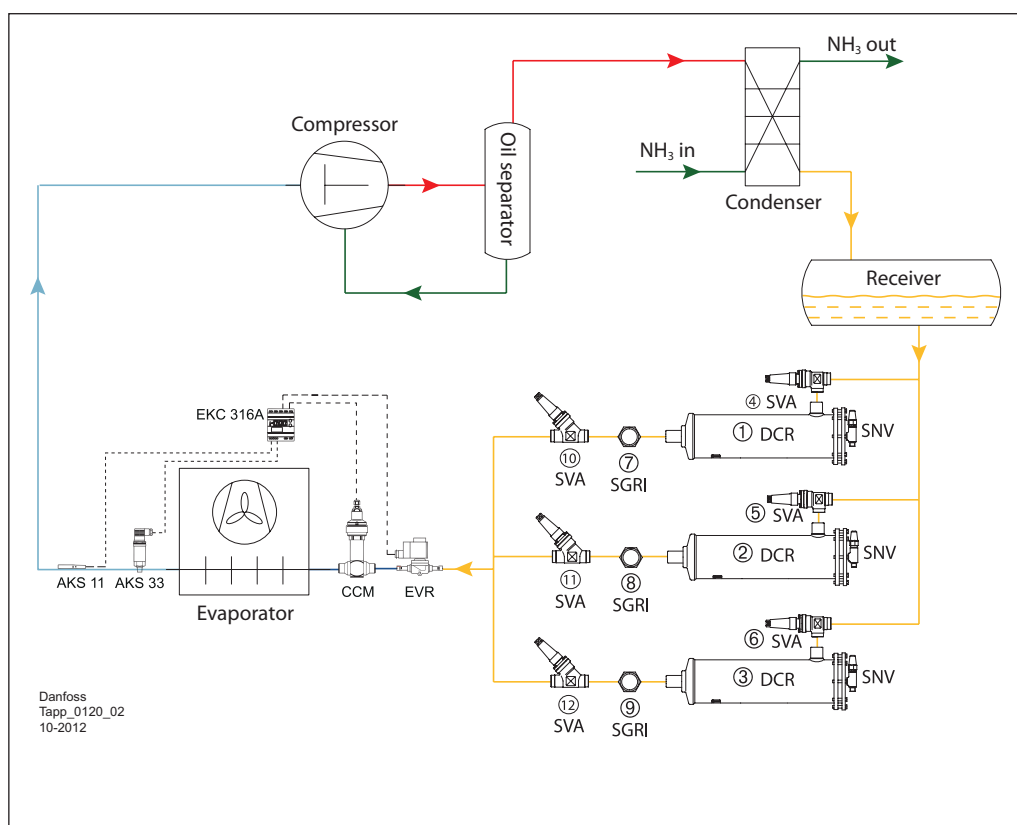
1. In the compressor-condenser-expansion valve loop the RH is low. In the liquid separator, more than 90% water exists in the liquid phase because of the much lower solubility of vapour CO₂ compared with liquid. Therefore, little water is brought into the compressor loop by the suction vapour. If filter driers are installed in this loop, the drier will have too little capacity.
2. In the wet suction line there is a risk of "freezing" because of the two phase flow as mentioned.
3. In the liquid line before the refrigerant pumps, pressure drop increases the risk of cavitation to the pumps.

If the capacity of one filter drier is not enough, several filters driers in parallel could be considered.

*Application example 10.9.7:
Filter driers in CO₂ DX systems*

- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- Oil

- ① Filter drier
- ② Filter drier
- ③ Filter drier
- ④ Stop valve
- ⑤ Stop valve
- ⑥ Stop valve
- ⑦ Sight glass
- ⑧ Sight glass
- ⑨ Sight glass
- ⑩ Stop valve
- ⑪ Stop valve
- ⑫ Stop valve



In a CO₂ DX system, the water concentration is the same throughout the system, so the RH is only up to the water solubility of the refrigerant.

Although the RH in the liquid line before the expansion valve is relatively small because of the high water solubility of the high temperature liquid CO₂, it's still recommended that filter driers be installed on this line (same position as in fluorinated system) for the following reasons:

1. In the suction line and discharge line, it is sensitive to the pressure drop, as well as the high risk of freezing in the suction line. Filter driers are not recommended to be installed here although the RHs are high.
2. In the liquid line after the expansion valve, installation of filter drier should also be avoided because of the two phase flow.

Not all valves are shown.
Not to be used for construction
purposes.

10.10 How does water enter a CO₂ system?

Unlike some ammonia systems, the pressure in CO₂ systems is always above atmospheric. However, water can still find its way into CO₂ systems.

Water may contaminate a CO₂ system through five different mechanisms:

1. Diffusion
2. Maintenance and repair activities
3. Incomplete water removal during installation/ commissioning
4. Water-contaminated lubricant charged into the system
5. Water-contaminated CO₂ charged into the system

Obviously, all these mechanisms should be avoided or minimized.

To illustrate a scenario in which water may contaminate a system, think of a contractor, who, believing CO₂ is a very safe refrigerant, thinks that it may be handled without following the normal ammonia safety requirements. He might open up the system to perform a repair. Once the system is opened up, air enters, and the moisture in the air condenses inside the piping. If the contractor does not evacuate the system very thoroughly, some water may be left in the system.

In another scenario, the contractor forgets that the lubricant used in the system, POE, has a high affinity for water, and leaves the cap off the container. After the POE is charged into the system, the water may start to cause problems in the system.

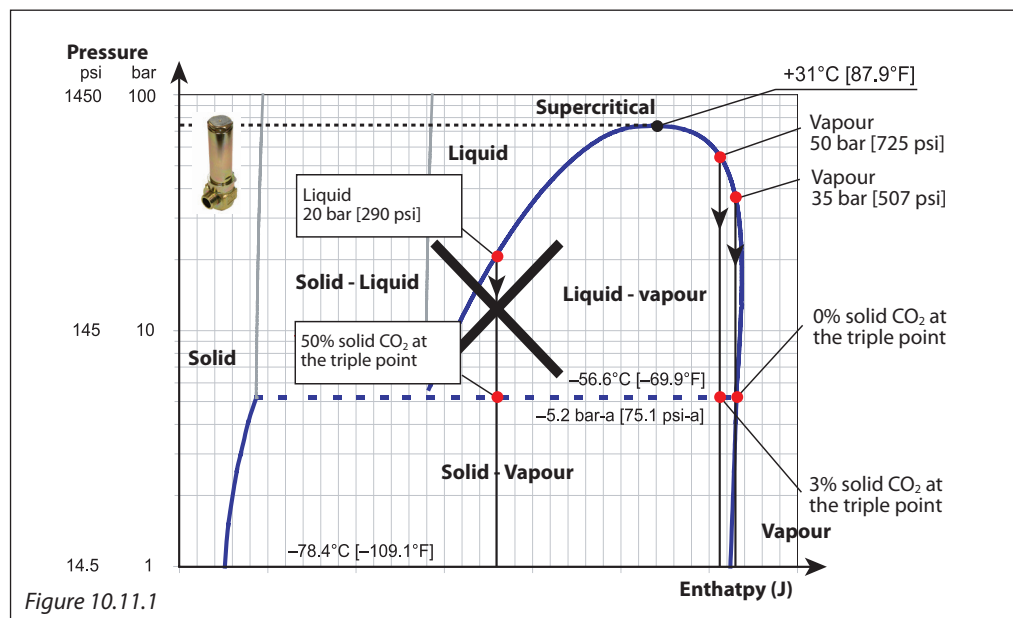
10.11 Miscellaneous features to be taken into consideration in CO₂ refrigeration systems

CO₂ expansion - phase changes
Safety valves

Safety valve

CO₂'s particularly high triple point can cause solid CO₂ to form under certain conditions. Figure 10.11.1 shows the expansion processes occurring in pressure relief valves starting at three different conditions. If the set pressure of a pressure relief

valve in the vapor phase is 35 bar [507 psi] or less, e.g., the rightmost line, the pressure in the relief line will pass through the triple point at 5.2 bar [75.1 psi]. Once below the triple point, the CO₂ will be pure vapor.



If the set pressure of a safety valve in the vapor phase is 50 bar [725 psi], e.g., the middle line, the relief line pressure will pass the triple point and 3% of the CO₂ will change into solid during relief. In a worst-case scenario (e.g., a long relief line with many bends), solid CO₂ may block this line. The most effective solution to this problem would be to mount the safety valve without an outlet line, and relieve the system directly to the atmosphere. The phase change of the CO₂ does not take place in the valve, but just after the valve, in this case, in the atmosphere.

If a pressure relief valve is set to relieve liquid at 20 bar [290 psi], the relief products would pass through the triple point, whereupon 50% of the CO₂ would change into solid upon further relief, subjecting the relief line to a high risk of blockage. Thus, to safely protect liquid lines against formation of dry ice, connect safety relief valves to a point in the system at a pressure higher than the triple point pressure of 5.2 bar [75.1 psi].

Charging CO₂

It is important to start up with CO₂ in the vapor phase, and continue until the pressure has reached 5.2 bar [75.1 psi]. It is therefore strongly recommended to write a procedure for charging a CO₂ system. One must be aware when charging a refrigerant system that until the pressure reaches the triple point, the CO₂ can only exist

as a solid or vapor inside the refrigeration system. Also, the system will exhibit very low temperatures until the pressure is sufficiently raised (figure 10.11.1). For example, at 1 bar [14.5 psi], the sublimation temperature will be -78.4°C [-109°F].

Filter cleaning

The same considerations apply to cleaning liquid strainers or filters. Even though CO₂ is non-toxic, one cannot just drain the liquid outside the system. Once the liquid CO₂ contacts the atmosphere, the liquid phase will partly change into the solid phase, and the temperature will drop dramatically, as in the example described

above. This sudden temperature drop is a thermal shock to the system materials, and can cause mechanical damage to the materials. Such a procedure would be considered to be a code violation because this equipment is not normally designed for such low temperatures.

Trapped liquid

Trapped liquid is a potential safety risk in refrigerant systems, and must always be avoided. This risk is even higher for CO₂ systems than for ammonia or R134a systems. The diagram in figure 10.11.2 shows the relative liquid volume

change for the three refrigerants. As shown, liquid CO₂ expands much more than ammonia and R134a, especially when the temperature approaches CO₂'s critical point.

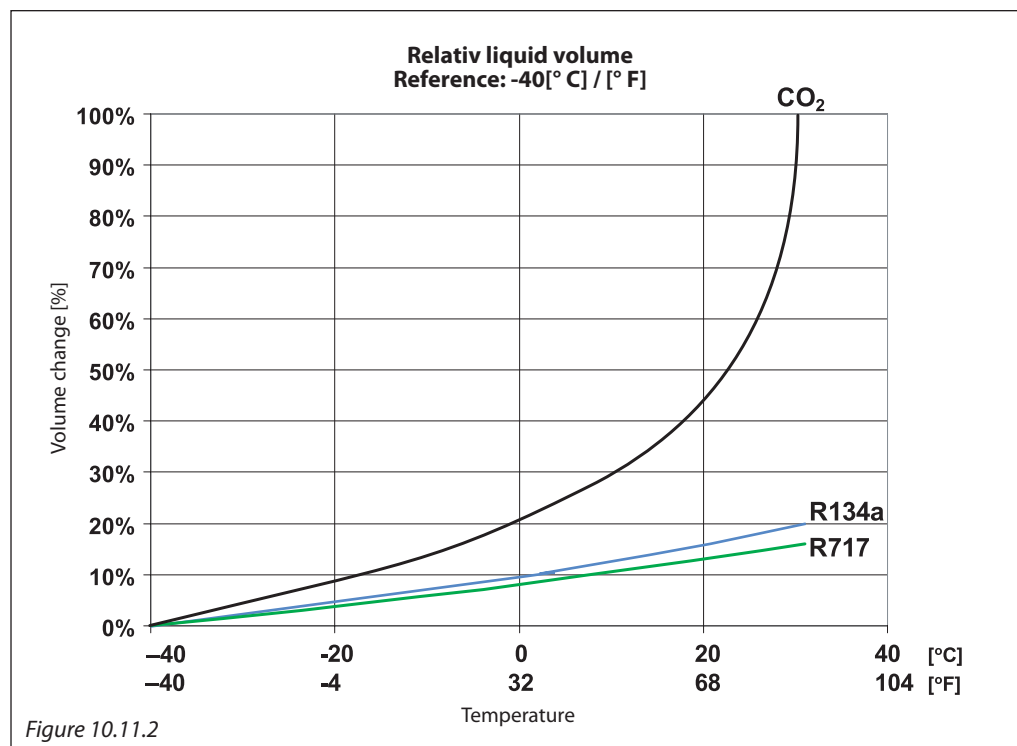
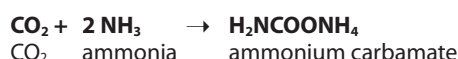


Figure 10.11.2

Leaks in CO₂-NH₃ cascade systems

The most critical leak in a CO₂-NH₃ cascade system is in the heat exchangers between CO₂ and NH₃. The pressure of the CO₂ will be higher than the NH₃, so the leak will occur into the NH₃ system, which will become contaminated.

The solid substance ammonium carbamate is formed immediately when CO₂ is in contact with NH₃. Ammonium carbamate is corrosive (ref. [5]).



Material compatibility

CO₂ is compatible with almost all common metallic materials, unlike NH₃. There are no restrictions from a compatibility point of view, when using copper or brass. The compatibility of CO₂ and polymers is much more complex. Because CO₂ is a very inert and stable substance critical, chemical reactions with polymers are not a problem. The main concern with CO₂ is the physiochemical effects, such as permeation, swelling and the generation of cavities and internal fractures. These effects are connected with the solubility and diffusivity of CO₂ in the material concerned.

These tests have shown that CO₂ is different, and modifications have to be made on some products. The large amount of CO₂ that can dissolve in polymers must be taken into consideration. Some commonly used polymers are not compatible with CO₂, and others require different fixing methods e.g. sealing materials. When the pressure is close to the critical pressure and the temperature is high, the impact on polymers is much more extreme. However, these conditions are not important for industrial refrigeration, as pressure and temperatures are lower in these systems.

Danfoss has carried out a number of tests to ensure that components released for use with CO₂ can withstand the impact of CO₂ in all aspects.

Conclusion

CO₂ has good properties, in particular at low temperature, but it is not a substitute for ammonia. The most common industrial CO₂ refrigeration systems are hybrid systems with ammonia on the high temperature side of the system.

The availability of components for industrial CO₂ refrigeration systems with pressures up to approximately 40 bar is good. Several manufacturers of equipment for traditional refrigerants can also supply some components for CO₂ systems. The availability of components for high-pressure industrial CO₂ refrigeration systems is limited, and the availability of critical components is an important factor in the growth rate of CO₂ use.

CO₂ is in many aspects a very uncomplicated refrigerant, but it is important to realize that CO₂ has some unique features compared with other common refrigerants. Knowing the differences, and taking these into account during design, installation, commissioning and operation, will help avoid problems.

References

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11. Pumped CO₂ in Industrial Refrigeration Systems

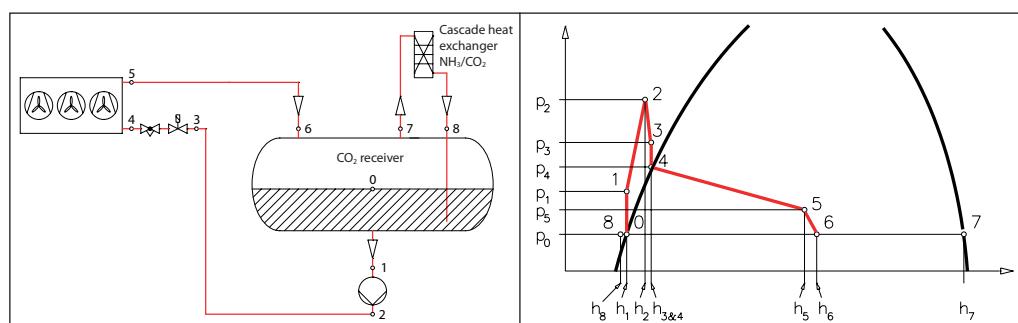


Figure 11.1 - General diagram of CO₂ pumped system.

General description of the systems

A typical schema of a low/medium temperature NH₃/CO₂ system (fig. 11.1) consisting of

- a standard NH₃ refrigeration system with a cascade heat exchanger acting as evaporator
- CO₂ acts as a volatile fluid in the evaporators (flooded system (1-6))

CO₂ is circulated by gravity in the cascade heat exchanger, which gives good control of the CO₂ temperature in the receiver.

The CO₂ gas flows up (7) into the cascade heat exchanger, where it is cooled by NH₃, condenses and flows back down into the CO₂ receiver as liquid CO₂ (8). On the ammonia side the refrigeration cycle can be controlled using a high pressure float valve (HFI) or by direct expansion into the evaporator (e.g. with an electronic expansion valve type ICM, and a cascade controller type EKC 313).

Differences to traditional NH3/brine systems

System performance:

NH₃/CO₂ fluid systems have significantly lower energy consumption compared to traditional systems with NH₃ and water based brines. COP of the system is higher due to the following:

- **Evaporation temperature and PHE efficiency**
Typically the high side NH₃ system evaporation temperature is a few degrees higher. The reason for this is the better CO₂ heat transfer coefficient in the air coolers and the PHE, resulting in a lower temperature difference in the heat exchangers. This directly reduces the energy consumption of the NH₃ compressors. Some figures indicate that the COP of NH₃/CO₂ systems is close to that of pure NH₃ systems.

- **Pump energy**

The pump energy needed to circulate the CO₂ through the air coolers is significantly lower, due to the fact that less CO₂ needs to circulate, but also thanks to the lower density of CO₂. The pump recirculation rate for CO₂ is relatively low as well (typically between 1.1 and 2), and this also makes it possible to use a smaller pump.

Line and component sizes in a flooded system:

Due to the high specific heat content of CO₂ and its lower density, smaller components and line sizes can be used compared to a traditional brine system, for both the outward and the return lines.

The smaller volume of circulating CO₂ to circulate means that smaller pumps can be used which yields lower energy consumption for the circulated cooling capacity.

The smaller CO₂ pipes have a smaller surface and therefore lower heat loss compared to larger brine/glycol pipes.

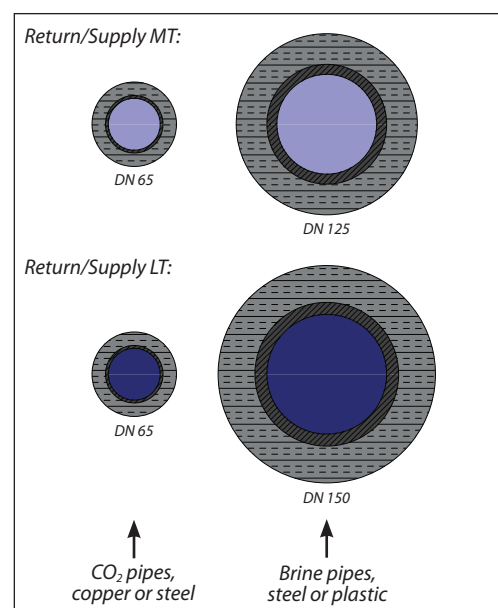


Figure 11.2 - Comparative pipe size

Differences to traditional
NH₃/brine systems.
(Continued)

Optimising energy management:
Further reduction of energy consumption by NH₃/CO₂ systems is possible using smart control algorithms. A good way to improve the efficiency (COP) of the system is to reduce the pressure ratio in the NH₃ compressor. There are two ways to do this:

- Keep the condenser at the lowest possible pressure.
- Keep evaporation at the highest possible pressure

The condenser control is similar to that of traditional systems, where fans can be controlled by an AKD102 variable frequency drive, and the condensing pressure can vary depending on the ambient temperature.

That can be done using the Danfoss pack controller AK-PC 730/840.

Management of the suction pressure is another area where there are differences between CO₂ cascade systems and brine/glycol systems. Assuming a system design as shown in fig. 11.3, a pressure signal from the CO₂ receiver can be used to control the capacity of the cascade compressors (the NH₃ system). If the pressure in the CO₂ receiver decreases, then the speed of the cascade compressors also decreases in order to maintain the CO₂ pressure. This function can be provided by the AK-PC 730 / 840 Pack Controller.

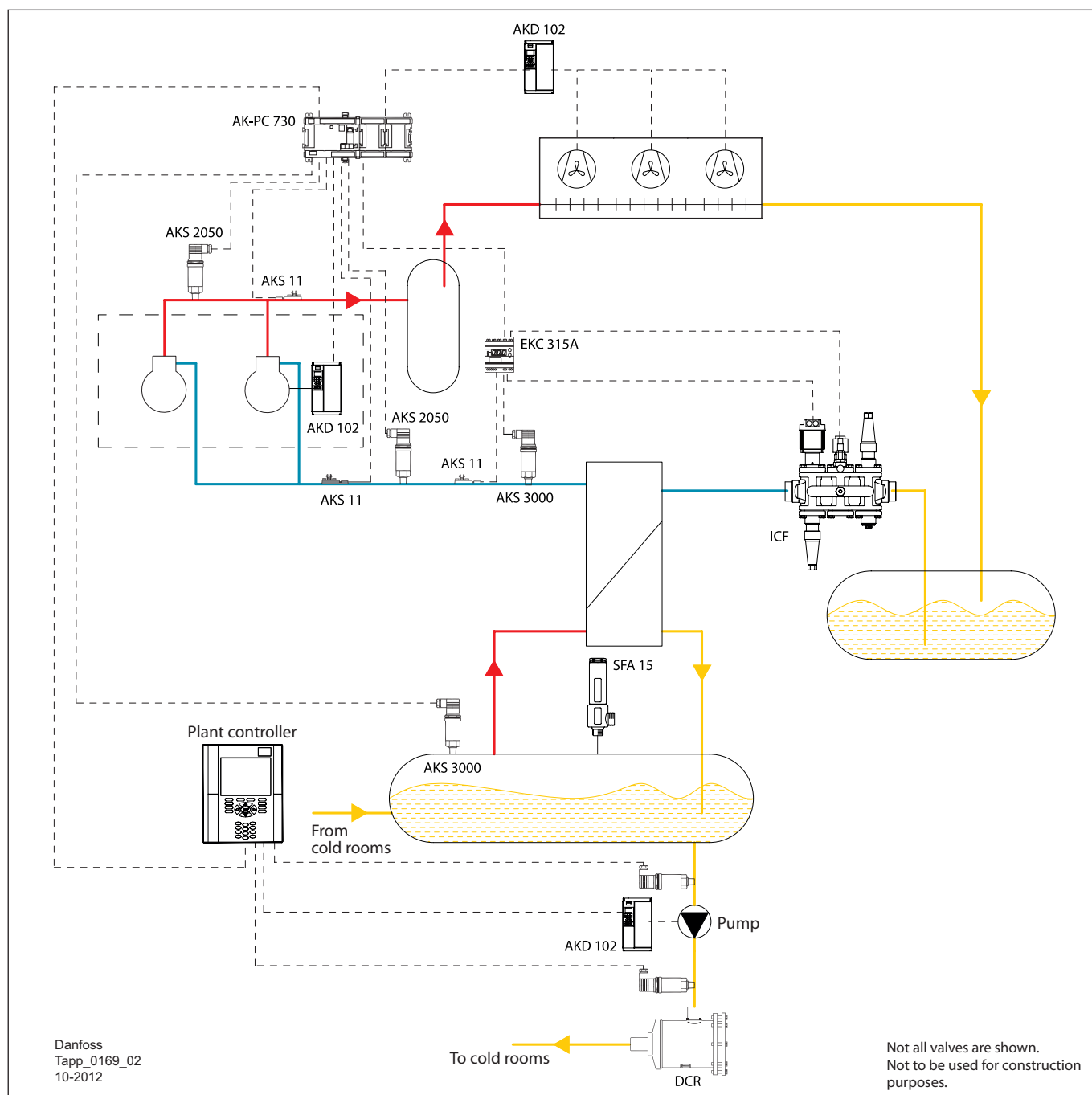


Figure 11.3 - Integrated control of pump-circulated CO₂ systems

Frequency control
of the CO₂ pumps

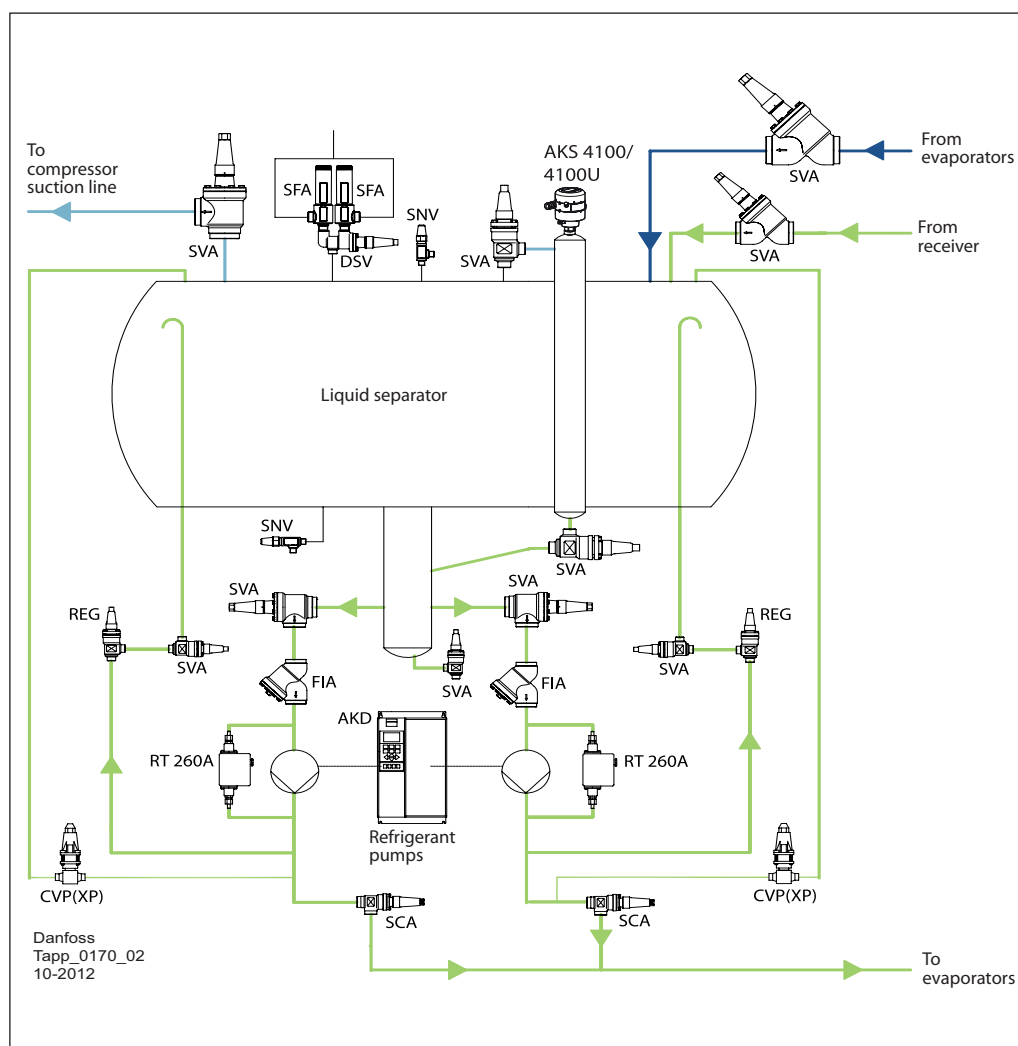


Figure 11.4 - CO₂ pump stations with AKD and the necessary valves and controls

There are two ways to control the liquid CO₂ pumps: using a simple on/off step control or using a frequency converter (type AKD). Frequency converter operation is becoming increasingly popular for two reasons: energy savings and better liquid distribution in the evaporator coils.

Energy savings

CO₂ pumps are typically controlled by a constant pressure difference. Under standard conditions the energy consumption is the same as or slightly higher than that of a fixed-speed pump. When running under partial load conditions, a fixed-speed pump would still consume the same energy due to the increased pressure difference. A liquid CO₂ pump using a frequency converter will run at a lower speed and consume less energy.

The savings will vary depending on the running time and the actual running conditions. Savings can, however, be up to 50% compared to pumps operating on/off at full speed.

Better liquid distribution in the evaporators

A requirement for good performance of the evaporators / air coolers is a good distribution of the refrigerant liquid in the system.

A precondition for good distribution of refrigerant liquid is having a stable pressure differential across the evaporators.

Pumps controlled by frequency converters can ensure that the pressure is kept at a stable level under all load conditions. At low capacity the energy consumption will be low and at high capacity there will be sufficient flow of CO₂.

A typical piping layout with CO₂ pumps controlled by frequency converters (AKD 102 type) is shown in figure 11.4. Another benefit of pumps driven by frequency converters is that the Q-max orifices can be omitted.

Defrosting pumped CO₂ systems

There are several ways to defrost pumped CO₂ systems

- **Electrical defrosting.** This is the simplest and least energy efficient method of defrosting. The additional power consumption for defrosting can be quite significant in some cases.
- **Hot gas defrosting.** CO₂ hot gas defrosting can be used if a compressor is built into the system to support defrosting. This compressor runs only when defrosting is needed. This method is more economical than electrical defrosting.

The process is similar to a traditional NH₃ defrosting system.

- **Brine defrosting.** By using brine it is possible to utilize the heat from the cascade system to defrost CO₂ evaporators. This method is especially attractive if the ammonia condenser is water cooled.
- **Water defrosting.** In some cases (especially in rooms with temperatures above zero) evaporators can be defrosted using sprayed water.

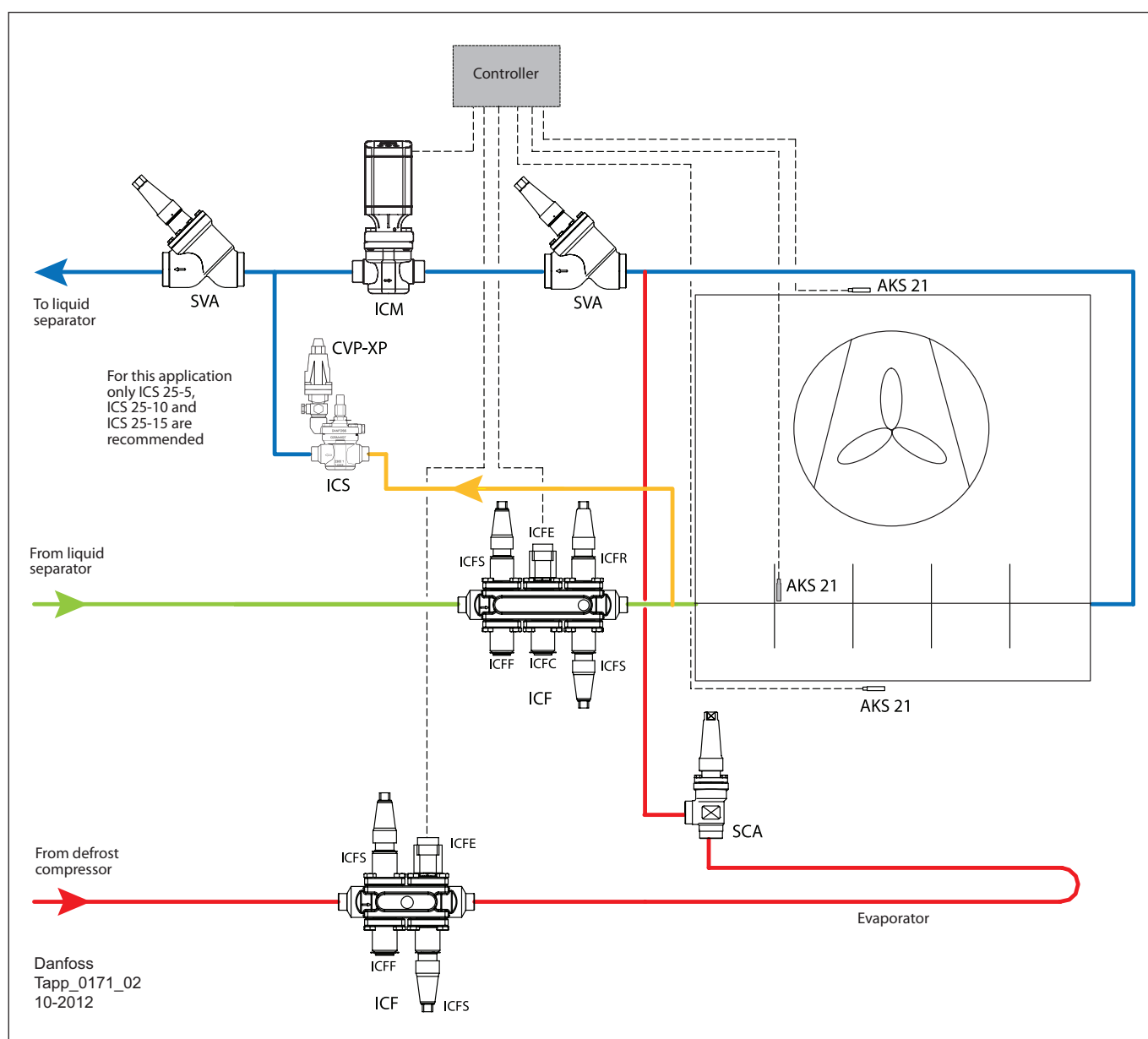


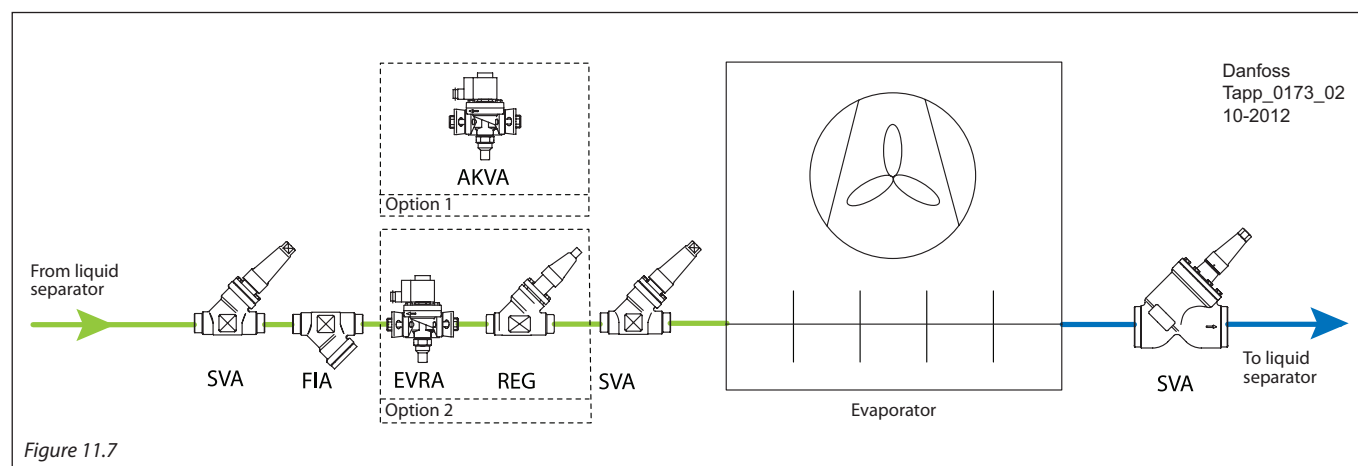
Figure 11.5 - CO₂ hot gas defrosting

Not all valves are shown.
Not to be used for construction purposes.

Evaporator control in pumped CO₂ systems (continued)

Temperature control in evaporators can be managed as follows:

- Regulating valve for distribution control + ON-OFF solenoid valve for temperature control
- Regulating valve for distribution control + pulse-width modulated solenoid valve for temperature control
- AKV valves for both distribution control (orifice size) and PWM temperature control



Traditional injection valves in pumped CO₂ systems

In a traditional flooded system, liquid injection is controlled by a thermostat which constantly measures the air temperature.

The solenoid valve is opened for several minutes or longer until the air temperature has reached the set point. During injection the mass of the refrigerant flow is constant.

This is a very simple way to control the air temperature, however, the temperature fluctuation caused by the differential of the thermostat may cause unwanted side effects in some applications, like dehumidification and inaccurate control.

Air cooler capacity

The capacity of an air cooler is described by the following equations:

Refrigerant side:

$$Q_{\text{cooler}} = \text{mass flow} \times \Delta h \quad (1)$$

Mass flow [kg/s evaporated liquid]

Δh [kJ/K]

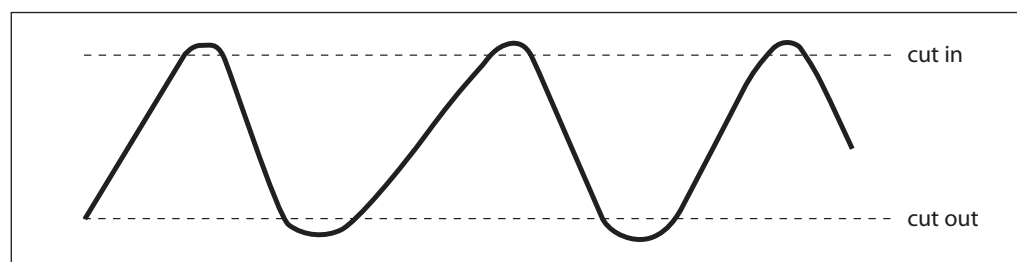
Refrigerant/Air side:

$$Q_{\text{cooler}} = k \times A \times \Delta T \quad (2)$$

k [W/(m².K)]: the total heat transfer coefficient, (depending on the heat transfer coefficient of the air and refrigerant, which depend on air/refrigerant flow) and the heat conductivity of the materials used in the coolers.

A [m²]: cooler surface

ΔT [K]: the difference between the evaporation and air temperatures.



*Injection into an air cooler
using a pulse width modulation
AKV(A) valve*

Instead of injecting periodically, as described above, one can also constantly adapt the liquid injection to the actual need. This can be done by means of a PWM AKV(A) valve type controlled by an AK-CC 450.

The air temperature is constantly measured and compared to the reference temperature. When the air temperature reaches the set point, the opening of the AKV(S) is reduced, giving it a smaller opening angle during a cycle, resulting in less capacity and vice versa. The duration of a cycle is adjustable between 30 sec. and 900 sec..

In principle, the regulation in this system is performed with a PI function. This results in reduced fluctuation of the regulated air temperature with stable loads, giving a more constant air humidity.

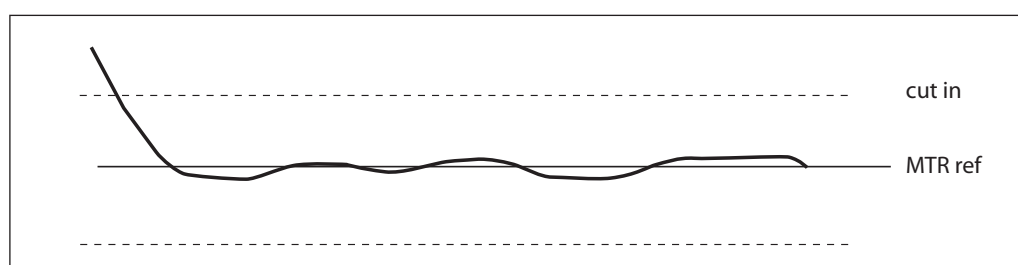
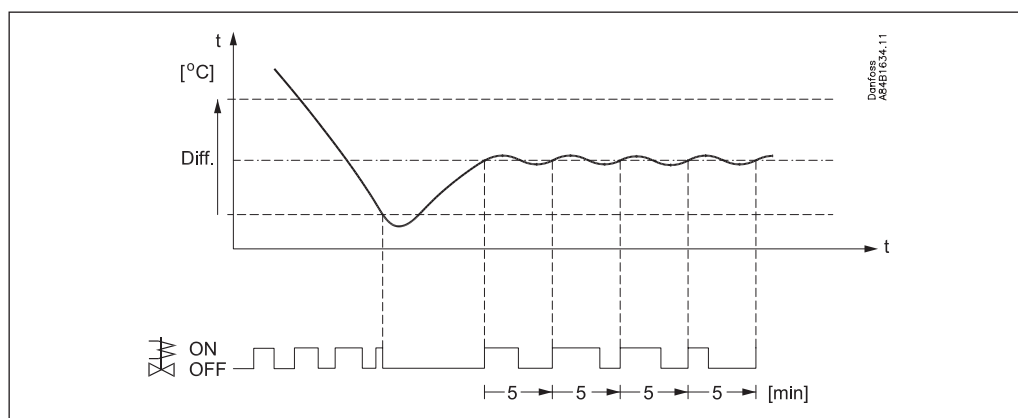
The function gives a constant temperature regulation with a temperature value, which lies

half-way between the on and off values of the thermostat.

The operating parameters of the PI regulation are automatically optimised via the preset on and off values and the degree of opening of the valve. The differential affects the amplification of the regulator and can therefore not be set to less than 2K in order to ensure regulation stability.

In a flooded system this means that the average refrigerant flow is constantly controlled and adapted to the demand, with circulation rate decreasing when less refrigerant is injected.

This approach to liquid injection in a flooded system is very versatile. The amount of injected liquid can be controlled exactly. A direct effect of this is a lower average surface temperature of the air cooler, resulting in a smaller ΔT between the refrigerant and the air. This increases the accuracy and the energy efficiency of the system.



Looking at the equations (1) and (2), it can be concluded that reducing injection results in:

- a decreasing ΔT (evaporating temperature comes nearer to ambient temperature)
- a decreasing k value

- a decreasing heat transfer surface on the air cooler (less "wetted" surface)

All resulting in smaller cooler capacity.

This approach to liquid injection in a flooded system yields a high degree of operational flexibility. The amount of injected liquid can be controlled exactly, which increases the accuracy and the energy efficiency of the system.

Typical applications are cool stores for fruit/vegetables, where adaptation to the actual load is frequently needed. A chilling cycle (AKV valve fully open) needs much more capacity than a storage cycle (AKV valves in PWM mode).

Also these types of cool rooms are often used for different amounts and types of fruit, so load adaptation is a must.

For more details, please refer to the manual of AK-CC 450 from Danfoss.

How to select an AKV(A) valve in a flooded CO₂ application?

When selecting a valve for a flooded system, we need to know the maximum cooler capacity required, given the highest circulation rate, which basically means the maximum amount of liquid to be injected. Secondly, we must define the net available pressure drop across the AKV(A) valve. The selection can be made easily using CoolSelector.

The minimum pressure drop in pressure needed in practice for an AKV(A) alone in a flooded system to operate satisfactorily has been shown to be 1 bar (or more if enough pump pressure is available).

Please be aware that the total pump pressure required depends on several factors, such as system pressure drop (distributors/nozzles of the air coolers, components, lines, bends, static height and so on)

Example:

- Refrigerant: CO₂
- N = 1.5
- T_o = -8°C
- Available drop in pressure across valve: 1 bar
- Cooler capacity: 30 kW

Coolselector® Version 1.0.1.26 Database Version 1.0.0.1

File Menu Preferences Code No Search Help **Danfoss**

Component Selector Line Design

Industrial Refrigeration Controls

Flooded evaporator, pump / Liquid line without phase change
Expansion Valves / AKVA

Search Criteria

Search Criteria

Refrigerant R744 Temperature -8 °C
Cooling Capacity 30 kW Pressure 27,98 bar
Mass Flow 637 Kg/h Circ. Rate 1,5
Pressure Drop 2 bar Pump Pres. Lift 2 bar
Design size, mm(in) Calculation basis Pressure Drop Calculation value 1 bar **Back**

Errors and omission expected the data are subject to change without notice

Type	Product	Feedback	Size DN	Size in	Dp. Actual Load (bar)	Dt. Actual Load (K)	Vel. Actual load (m/s)
AKVA	AKVA 15-3	⚠	DN25	1	1,05	1,29	0,285
AKVA	AKVA 15-4	⚠	DN25	1	0,417	0,508	0,285
AKVA	AKVA 15-2	⚠	DN20	3/4.	2,61	3,27	0,466

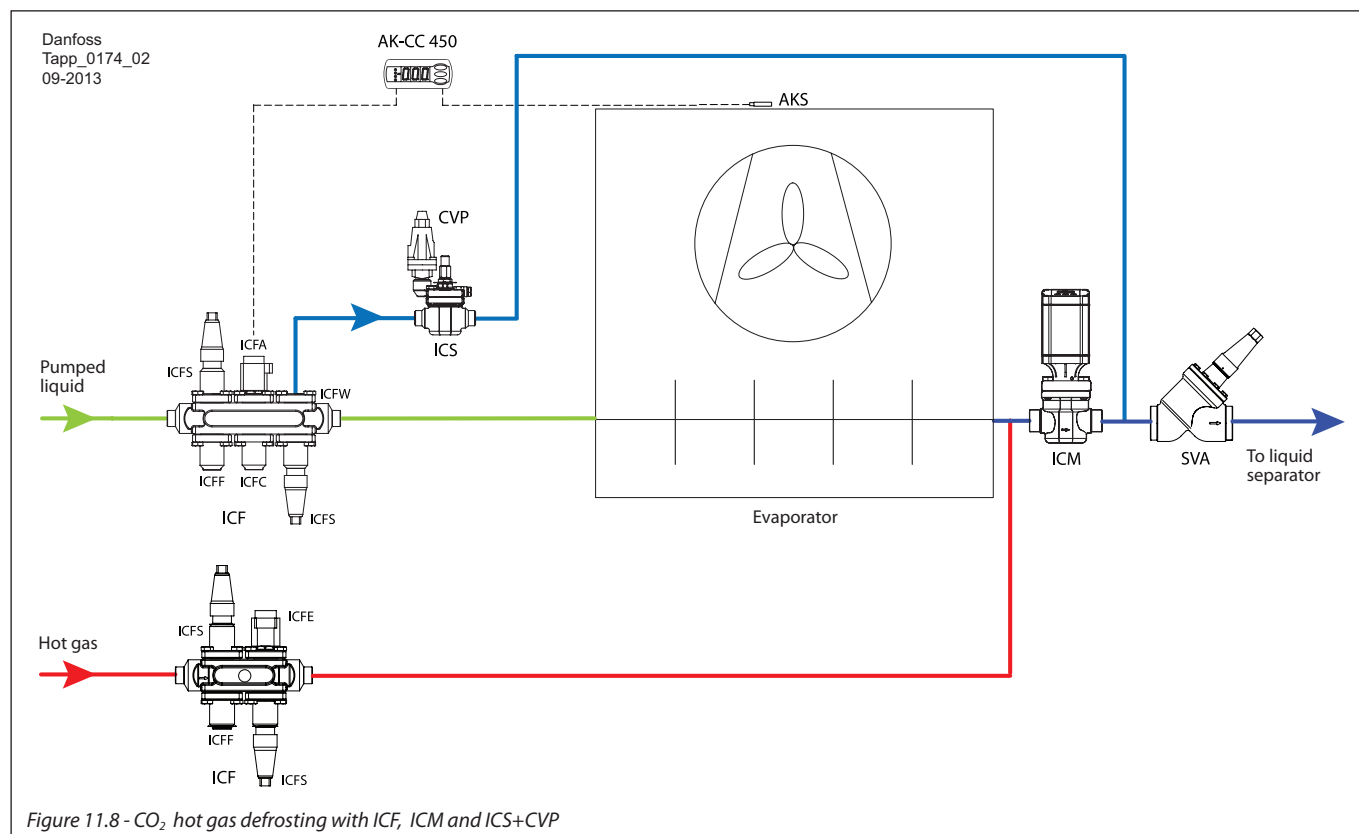
CoolSelector recommends an AKVA 15-3, (kV = 0.63 m³/h) which yields 30 kW at a circulation rate of 1.5 and a pressure drop across the valve of 1 bar. If more capacity is needed, a bigger valve or higher pressure drop in pressure across the valve should be provided.

Please keep in mind that all AKVA versions have a PS of 42 bar, AKV versions only have a PS of 42 bar in the AKV10 series and AKV15-1,2,3

Pumped systems with ICF

The example on the previous page is implemented with a standard AKVA valve. A multi-modular valve of type ICF would also be a good choice for this application.

If the coolers are defrosted using CO₂, a version with a check valve is needed.



Special care should be taken with the solenoid valve in the wet suction line. A commonly used defrost temperature is around 9-10°C, corresponding to a pressure of 44-45 bar (a) upstream of this solenoid valve.

Depending on the separator pressure, the MOPD of this valve could be too small to open. It is good practice to use a small bypass valve like EVRST (PS = 50 bar) to equalise the pressure first, before opening the main valve. The MOPD of the ICM 20-32 is 52 bar, so it is always able to open after a defrost cycle, even when the separator pressure is near the triple point of 5.2 bar a.

A benefit of using ICM is that the defrost pressure can be equalised by slowly opening the valve. A cost-effective way to do this is using the on/off mode on the ICM and selecting a very low speed (I04), or it can be achieved by using the modulating mode, so the PLC totally controls the opening degree and speed.

For an alphabetical overview of all reference documents please go to page 146

Technical Leaflet / Manual

Type	Literature no.
AK-CC 450	RS8EU
AKD 102	PD.R1.B
AK-PC 730	RS8EG
AKS 21	RK0YG
AKS 33	RD5GH
AKS 4100/4100U	PD.SC0.C
AKVA	PD.VA1.B
CVC-XP	PD.HN0.A
CVC-LP	PD.HN0.A
CVP	PD.HN0.A
CVPP	PD.HN0.A
DCR	PD.EJ0.A
EKC 315A	RS8CS

Type	Literature no.
EKC 347	PS.G00.A
EVM	PD.HN0.A
EVRA / T	PD.BM0.B
FIA	PD.FN1.A
ICF	PD.FT1.A
ICM / ICAD	PD.HT0.B
ICS	PD.HS2.A
NRV	PD.FE0.A
OFV	PD.HQ0.A
REG	PD.KM1.A
RT 260A	PD.CB0.A
SCA	PD.FL1.A
SGR	PD.EK0.A
SNV	PD.KB0.A
SVA-S/L	PD.KD1.A

Product instruction

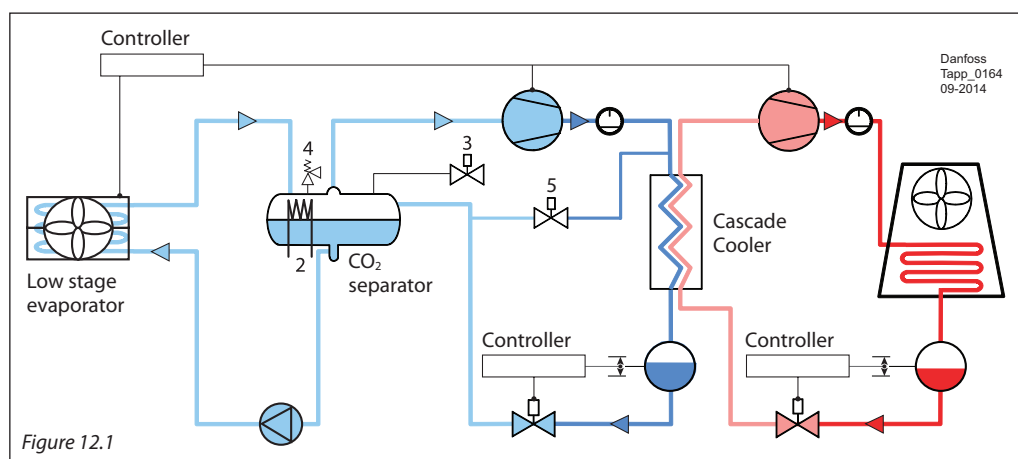
Type	Literature no.
AKD 102	MG11L
AKS 21	RI14D
AKS 32R	PI.SB0.A
AKS 33	PI.SB0.A
AKS 4100/4100U	PI.SC0.D / PI.SC0.E
AKVA	PI.VA1.B PI.VA1.C
CVC-XP	PI.HN0.A
CVC-LP	PI.HN0.M
CVP	PI.HN0.C
CVPP	PI.HN0.C
DCR	PI.EJ0.B
EKC 347	PI.RP0.A
EVM	PI.HN0.N

Type	Literature no.
EVRA / T	PI.BN0.L
FIA	PI.FN1.A
ICF	PI.FT0.C
ICM / ICAD	PI.HT0.A (ICM) PI.HT0.B (ICAD)
ICS	PI.HS0.A / PI.HS0.B
NRV	PI.FE0.A
OFV	PI.HX0.B
REG	PI.KM1.A
RT 260A	RI5BB
SCA	PI.FL1.A
SGR	PI.EK0.A
SNV	PI.KB0.A
SVA-S/L	PI.KD1.A

To download the latest version of the literature please visit the Danfoss website.

12. Control methods for CO₂ systems

- 2 Auxillary refrigeration system / stand still unit (cooling)
- 3 Solenoid valve
- 4 Safety valve
- 5 Bypass valve



Compressor control

There is no difference in the way the compressors can be controlled in CO₂ systems compared to a normal industrial refrigeration installation, but as they are cascade systems, it must be ensured that the NH₃ compressor is started / ready to start before the start signal is given to the CO₂ compressor (see the section regarding compressor control).

Liquid level control

There is no difference in the way the liquid level can be controlled in CO₂ systems compared to a normal industrial refrigeration installation (see the section regarding liquid level control).

Possible control devices in case of high pressure in the CO₂ separator

If the pressure in the CO₂ separator rises above the normal range, the following steps can be taken in order to minimize the escape of CO₂:

1. The CO₂ compressor can be forced to start and the CO₂ liquid pump forced to stop to prevent relatively warm liquid returning to the CO₂ separator.
2. If there is a fault preventing the CO₂ compressor from starting, the pressure will continue to increase. This will force the stand-still unit to start.
3. If the pressure continues to rise, a solenoid valve can be forced open to provide a controlled release of the CO₂ pressure down to a defined pressure.
4. The last device is the safety valve, which operates at its set pressure.

Possible control devices in case of low pressure in the CO₂ separator

If the pressure in the CO₂ separator drops below the normal operating range, the following steps can be taken to minimize the risk of formation of dry ice:

5. Opening a bypass valve enables the system to maintain a sufficient high suction pressure in the CO₂ separator. This also prevents stopping of the compressor to stop if there is a sudden drop in cooling load, e.g. if there is a freezing process with variations in the cooling load. This ensures that the compressor keeps running and maintains the system ready for a sudden increase in the cooling load.
6. The CO₂ compressor can be forced to stop, and thus avoid the forming of dry ice.

13. Design of a CO₂ sub-critical installation

In general the design and selection of valves for a CO₂ sub-critical installation are no different than for a traditional NH₃ installation, except for the higher working pressures and the oil recovery system.

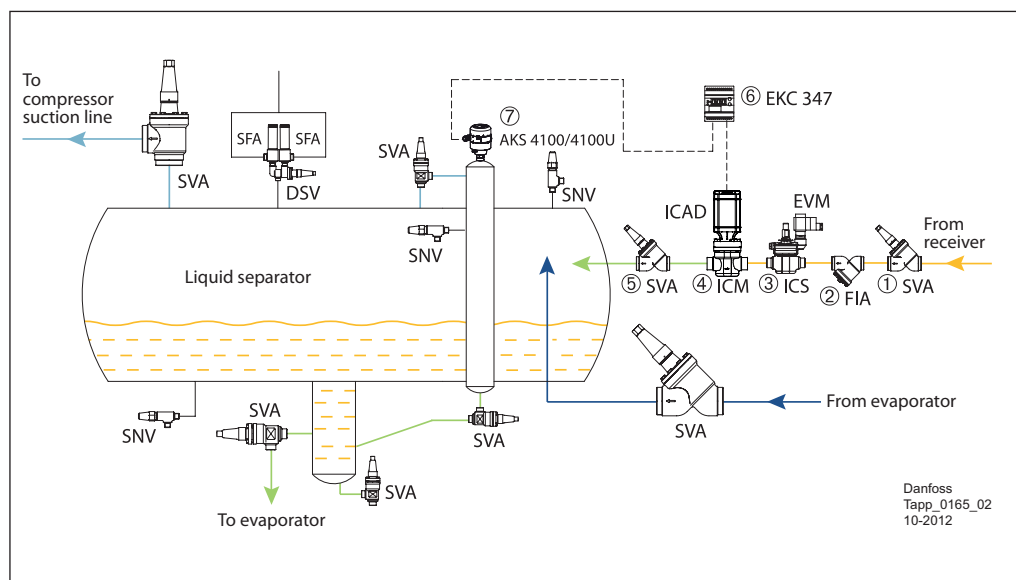
Therefore, the examples given in the previous sections of this handbook are also valid for CO₂. However, generally speaking it is recommended to avoid flange connections in CO₂ systems where possible.

13.1 Electronic solution for liquid level control

*Application example 13.1.1:
Electronic solution for LP liquid level control*

- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- LP liquid refrigerant

- ① Stop valve
- ② Filter
- ③ Solenoid valve
- ④ Motor valve
- ⑤ Stop valve
- ⑥ Controller
- ⑦ Level transmitter



The level transmitter AKS 4100/4100U ⑦, monitors the liquid level in the separator and sends a level signal to the liquid level controller EKC 347 ⑥, which sends a modulating signal to the actuator of the motor valve ICM ④. The ICM motor valve acts as an expansion valve.

The liquid level controller EKC 347 ⑥ also provides relay outputs for upper and lower limits and for alarm level.

*Application example 13.1.2:
Electronic solution for LP liquid level control*

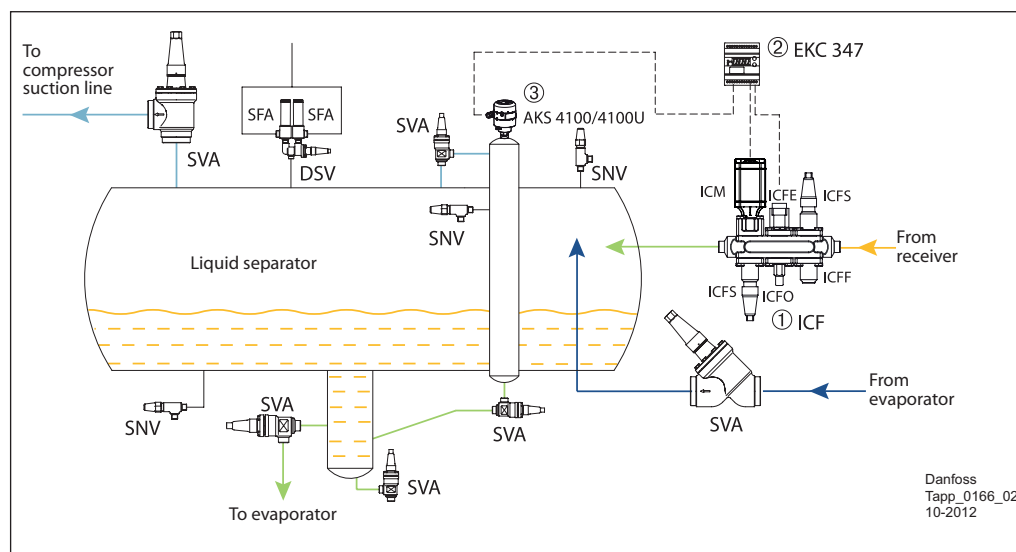
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- LP liquid refrigerant

- ① ICF valve station including:



- Stop valve
- Filter
- Solenoid valve
- Manual opener
- Motor valve
- Stop valve

- ② Controller
- ③ Level transmitter



Danfoss can supply a very compact valve solution ICF ①. Up to six different modules can be fitted in the same housing, which is easy to install. The module ICM acts as an expansion valve and the module ICFE is a solenoid valve.

This solution operates in the same way as example 13.1.1. Please refer to ICF literature for further information.

Not all valves are shown.
Not to be used for construction purposes.

13.2 Hot Gas Defrost for Pumped Liquid Circulation Air Coolers

Application example 13.2.1:
Pumped liquid circulation
evaporator, with hot gas defrost
system

- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture
of refrigerant
- LP liquid refrigerant

Liquid Line

- ① Liquid inlet stop valve
- ② Filter
- ③ Solenoid valve
- ④ Check valve
- ⑤ Manual expansion valve
- ⑥ Evaporator inlet stop valve

Suction Line

- ⑦ Evaporator outlet stop valve
- ⑧ Motor valve
- ⑨ Suction line stop valve

Hot gas line

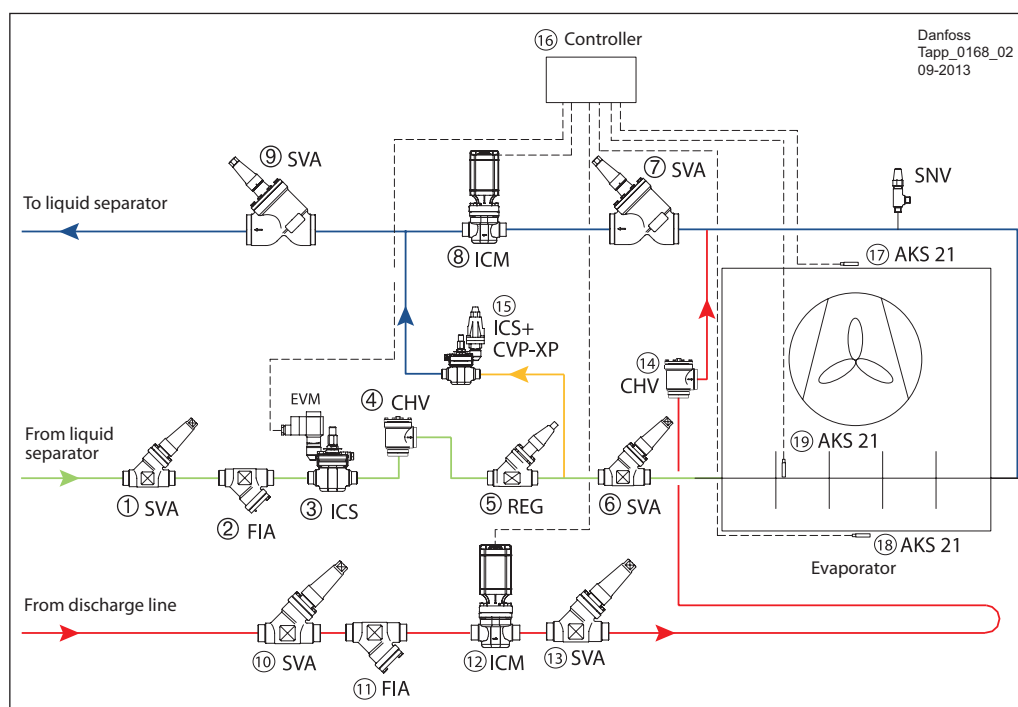
- ⑩ Stop valve
- ⑪ Filter
- ⑫ Motor valve
- ⑬ Stop valve
- ⑭ Check valve

Overflow line

- ⑮ Overflow valve

Controls

- ⑯ Controller
- ⑰ Controller
- ⑱ Controller
- ⑲ Controller



Application example 13.2.1 shows an installation for pumped liquid circulation evaporators with hot gas defrost using the ICV valves.

Refrigeration Cycle

The solenoid valve ICS ③ in the liquid line is kept open. Liquid injection is controlled by the manual regulating valve REG ⑤. The motor valve ICM ⑧ in the suction line is kept open, and the defrosting motor valve ICM ⑩ is kept closed.

Defrost Cycle

After initiation of the defrost cycle, the liquid supply solenoid module ICFE of the ICS ③ is closed. The fan is kept running for 120 to 600 seconds depending on the evaporator size in order to pump down the liquid in the evaporator.

The fans are stopped and the ICM valve ⑧ closed.

A delay of 10 to 20 seconds is provided to allow the liquid in the evaporator to settle down in the bottom without vapour bubbles. The motor valve ICM ⑩ is then opened and supplies hot gas to the evaporator.

Because of the high differential pressure between the hot gas line and the evaporator, it is recommended to increase the pressure slowly, allowing the pressure to be equalized before opening fully to ensure smooth operation and avoid liquid slugging in the evaporator.

A benefit of using the motor valve ICM ⑩, a benefit is that the defrost pressure can be equalized by slowly opening the valve. A cost effective way to do this is using the on/off mode on the ICM and selecting a very low speed, or it can be achieved by using the modulating mode, so the PLC totally controls the opening degree and speed.

During the defrost cycle, the condensed hot gas from the evaporator is injected into the low pressure side. The defrost pressure is controlled by the ICS+CVP ⑮.

When the temperature in the evaporator (measured by AKS 21) reaches the set value, defrost is terminated, the motor valve ICM ⑧ is closed, and after a small delay the motor valve ICM ⑩ is opened.

Because of the high differential pressure between the evaporator and the suction line, it is necessary to relieve the pressure slowly, allowing the pressure to be equalized before opening fully to ensure smooth operation and avoid liquid slugging in the suction line.

A benefit of using the motor valve ICM ⑩ a benefit is that the defrost pressure can be equalized by slowly opening the valve. A cost effective way to do this is using the on/off mode on the ICM and selecting a very low speed, or it can be achieved by using the modulating mode, so the PLC totally controls the opening degree and speed.

After the ICM fully opens, the liquid supply solenoid valve ICS ③ is opened to start the refrigeration cycle. The fan is started after a delay in order to freeze remaining liquid droplets on the surface of the evaporator.

In FIA, pos 2 and 11 (and in general in CO₂ systems), it is recommended to use a pleated insert with extra large surface and a more solid design.)

13.2 Hot Gas Defrost for Pumped Liquid Circulation Air Coolers

*Application example 13.2.2:
Pump circulated evaporator,
with hot gas defrost system,
fully welded, using ICF Valve
station for evaporator with hot
gas defrost*

■ HP vapour refrigerant
■ HP liquid refrigerant
■ Liquid/vapour mixture
of refrigerant
■ LP liquid refrigerant

① Liquid Line ICF with:



Stop valve
Filter
Solenoid valve
Check valve
Manual expansion valve
Evaporator inlet stop valve

② Evaporator outlet stop valve

③ Pressure regulator
(motor valve)

④ Suction line stop valve

⑤ Hot gas line ICF with:



Stop Valve
Filter
Solenoid valve
Stop valve

⑥ Check valve

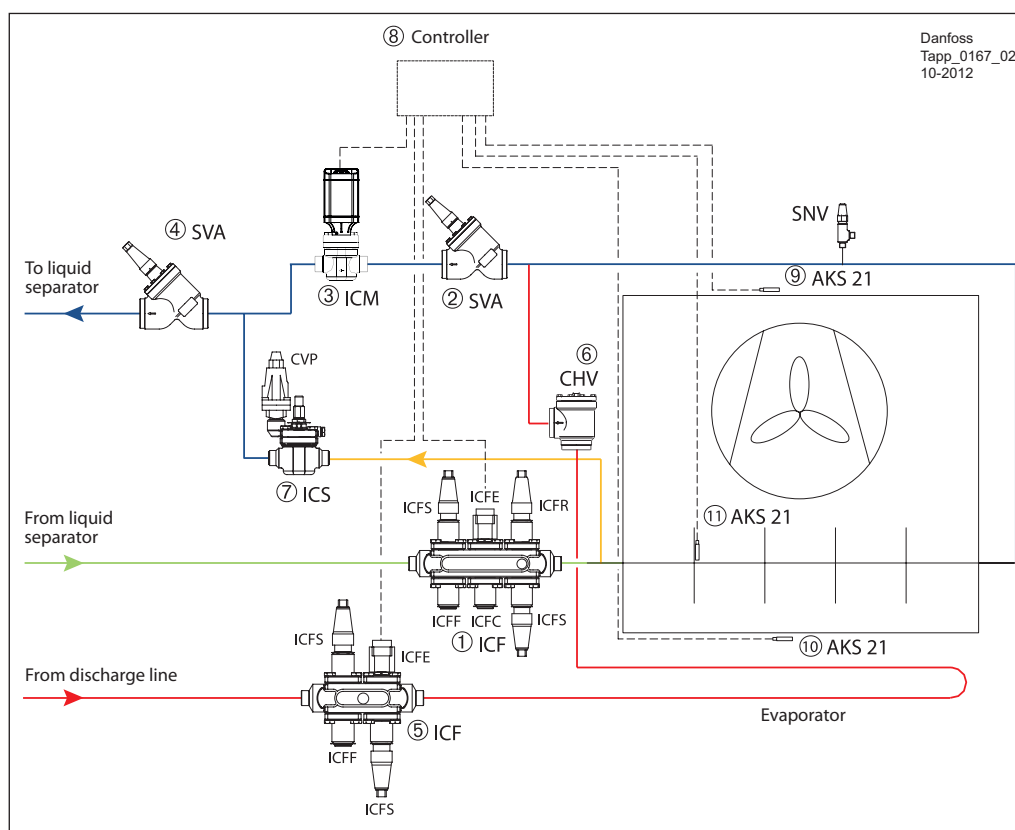
⑦ Pressure regulator

⑧ Controller

⑨ Temperature sensors

⑩ Temperature sensors

⑪ Temperature sensors



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Application example 13.2.2 shows an installation for pumped liquid circulation evaporators with hot gas defrost using the new ICF control solution.

The ICF will accommodate up to six different modules fitted in the same housing, offering a compact, easy to install control solution.

Refrigeration Cycle

The solenoid valve ICFE in ICF ① in the liquid line is kept open. The liquid injection is controlled by the hand regulating valve ICFR in ICF ①.

The motor valve ICM ③ in the suction line is kept open, and the defrosting solenoid valve ICFE in ICF ⑤ is kept closed.

Defrost Cycle

After initiation of the defrost cycle, the liquid supply solenoid module ICFE of the ICF ① is closed. The fan is kept running for 120 to 600 seconds depending on the evaporator size in order to pump down the liquid in the evaporator. The fans are stopped and the ICM valve closed. A delay of 10 to 20 seconds is provided to allow the liquid in the evaporator to settle down in the bottom without vapour bubbles. The solenoid valve ICFE in ICF ⑤ is then opened and supplies hot gas to the evaporator.

During the defrost cycle, the condensed hot gas from the evaporator is injected into the low pressure side. The defrost pressure is controlled by the ICS+CVP ⑦.

When the temperature in the evaporator (measured by AKS 21) reaches the set value, defrost is terminated, the solenoid valve ICFE in ICF ⑤ is closed, and after a small delay the motor valve ICM ③ is opened.

Because of the high differential pressure between the evaporator and the suction line, it is necessary to relieve the pressure slowly, allowing the pressure to be equalized before opening fully to ensure smooth operation and avoid liquid slugging in the suction line.

A benefit of using the motor valve ICM ③ is that the defrost pressure can be equalized by slowly opening the valve. A cost effective way to do this is to use the ICM on/off mode and select a very low speed. It can also be achieved by using the modulating mode, so that the PLC fully controls the opening degree and speed.

After the ICM fully opens, the liquid supply solenoid valve ICFE in ICF ① is opened to start the refrigeration cycle. The fan is started after a delay in order to freeze remaining liquid droplets on the surface of the evaporator.

14. Danfoss sub-critical CO₂ components



Today, Danfoss now offers a broad range of industrial components suitable for CO₂.

The majority of the components listed below have been evaluated and upgraded, and are therefore applicable for CO₂ within the pressure and temperature ranges stated in the technical documentation. In particular the pressure is the limiting factor for this group of components.

Special components for high-pressure CO₂ applications have been developed. The most common types of valves are listed on the following pages.


Please note that special high-pressure versions are generally only available on special order and extended delivery times should be taken into account.


Pressure Equipment Directive (PED)

The Industrial Refrigeration valves are approved in accordance with the European standards specified in the Pressure Equipment Directive and are CE marked.

Industrial Refrigeration products

Danfoss Sub Critical CO ₂ - components Industrial Refrigeration products			DN	PS 40bar [580 psi]	PS 52 bar [754 psi]
Main Valves, Solenoid Valves	ICS 1 ICS 3	all	20-150		
Multifunction valve	ICF	all	20-40		
Pilots for ICS Valves	CVP-XP				
	CVP-XP				
	CVC-XP				
	CVP-HP				
	CVPP-HP				
	EVM (NC)				65 bar
	EVM (NO)				
Stop Valves	SVA-S	all	6-200		
	SVA-L	all	15-40		
Regulating Valves	REG-SA/SB	all	15 - 65		
Stop Check Valves	SCA-X	all	15-125		
Filters	FIA	all	15-200		
Check Valves	CHV-X	all	15- 125		
Solenoid Valves	EVRS/EVRST	all	3-20		
	ICS + EVM	all	20-150		
Electrically operated expansion valve	AKVA	all	10-20		
	ICM	all	20-65		
	ICMTS	all	25		140 bar
	CCMT	all	15		140 bar
	CCM	all	15-25		90 bar
Safety Valves and Change Over Valves	SFA	15	-		
	DSV	1, 2	20-32		
	POV	40, 50, 80	40-80		
Filter drier	DCRH	High pressure version			46 bar
Liquid level transmitter	AKS 4100/4100U	-	-		
Gas detectors	GD				


 The product can be used in standard version.
All products are CE approved


 The product must be manufactured in a special version (higher test pressure, marking and documentation). All products are CE approved

14.
Danfoss sub-critical CO₂
components
(Continued)

Commercial Refrigeration products

Danfoss Sub Critical CO ₂ - components Commercial Refrigeration products		PS 42 bar [609 psi]	PS 46 bar [667 psi]	PS 52 bar [754 psi]	PS 90 bar [1305 psi]	PS 140 bar [2031 psi]
Solenoid Valves	EVR 2 - EVR 15					
	EVUL					
	EVUB					
Shutoff Valves (Ball Valves)	GBC for CO ₂					
Check Valves	NRV for CO ₂					
Electrically operated expansion valve	AKVH 10					
	AKV 15					
	ETS 12.5 - 100					
	CCM10 - 40					
	CCMT2 - 8					
	ICMTS					
Automatic Pressure Regulators	ICV					
	MBR					
Filter Driers	DCR					
	DML					
	DMT					
Moisture Indicator	SG (Inline)					
	SG (Socket)					

 The product can be used in standard version.
All products are CE approved

 The product must be manufactured in a special
version (higher test pressure, marking and
documentation). All products are CE approved

Electronic Controls for CO₂

Case controllers: AK-CC 550A, AK-CC 750
Evaporator controllers: EKC 315A, AKC 316, EKD 316, EKC 312
Pack controllers: EKC 331T, AK-PC 530, AK-PC 420, AK-PC 781, AK-PC 840
Chiller controls: AK-CH 650, AK-CH 650A
Cascade controller: EKC 313
Pressure controller: EKC 326A

Coils for solenoid valves



Due to the high pressure difference between the condenser and evaporator, the Maximum Opening Pressure Differential (MOPD) requirement for the solenoid valve in some applications may exceed the standard coil capabilities.

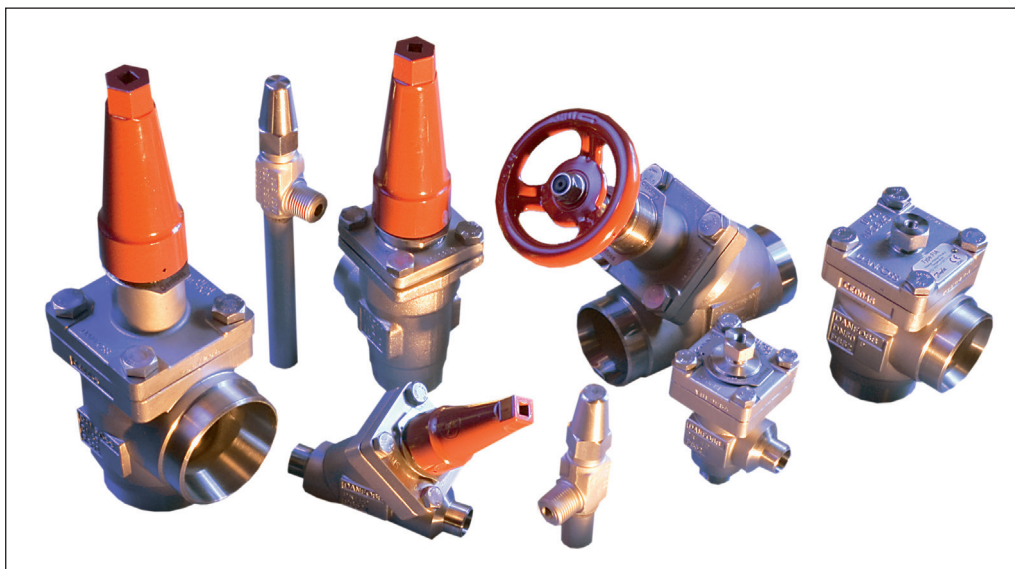
Examples of typical applications are:

- Liquid injection for cooling the compressor
- Hot gas defrost
- Shutoff valve before expansion valve

Therefore Danfoss offers a 20 W coil that covers a MOPD range up to 40 bar.

The 20 W coil range includes coils for 24, 110 and 230 V a.c. 50 Hz supply voltages.

15. Full range of stainless steel products



Surface protection is becoming increasingly important, especially for refrigeration systems in the food industry, where cleaning with strong cleaning agents is common.

Therefore Danfoss offers both angle flow and straight flow versions of stainless steel valves in the sizes DN 15 mm (1/2") to DN 125 mm (5") .

- Stop Valves SVA-SS
- Manual regulating valves REG-SS
- Stop Check Valves SCA-SS (only angleway)
- Check Valves CHV-SS (only angleway)
- Filters FIA-SS
- Overflow Valves OFV-SS (only angleway)
- Needle valves SNV-SS

This range of valves meets more stringent requirements resulting from:

1. The need for higher protection of external surfaces on valves and fittings
2. The need to accommodate current trends in plant design.

In certain specific areas such as outdoor applications and corrosive atmospheres, such as coastal installations, there is a need for high surface protection to prevent failure due to corrosion.

Today's food safety standards often call for daily cleaning with detergents to protect against bacteria growth, again producing a need for high surface protection.

- Compatible with all common non flammable refrigerants including R717 and non-corrosive gases/liquids dependent on sealing material.

- Optional accessories:

	Vented cap	Handwheel
SVA-SS	X	X
REG-SS	X	
SCA-SS	X	
CHV-SS		
FIA-SS		
OFV-SS	X	

- Designed to give favourable flow conditions.
- Internal backseating enables replacement of the spindle seal while the valve is in service, i.e. under pressure (SVA-SS, REG-SS, SCA-SS, OFV-SS).

- Housing is made of special cold resistant stainless steel approved for low temperature operation.
- Easy to disassemble for inspection and service.
- SVA-SS Stop Valves can accept flow in either direction.
- Butt-weld DIN connections.
- Max. operating pressure: 52 bar g (754 psig)
- Temperature range: -60 to +150°C (-76 to 302°F).
- Compact and light valves for easy handling and installation.
- Classification: contact your Danfoss sales company for a current product certification list.

Stainless steel solenoid valves EVRS and EVRST



EVRS 3 is direct operated.
EVRS 10, 15 and 20 are servo operated.
EVRST 10, 15 and 20 are forced servo operated valves used in liquid, suction, hot gas and oil return lines with ammonia or fluorinated refrigerants.

EVRS 3 and EVRST are designed for keeping open at a pressure drop of 0 bar.
EVRS/EVRST 10, 15 and 20 are equipped with spindle for manual opening.
EVRS and EVRST are supplied as components, i.e. valve body and coil must be separately ordered.

Features

- Stainless steel valve body and connections
- Max. working pressure 50 barg
- Used for ammonia and all fluorinated refrigerants
- MOPD up to 38 bar with 20 watt a.c. coil
- Wide choice of a.c. and d.c. coils
- Designed for temperatures of media up to 105°C
- Manual stem on EVRS and EVRST 10, EVRST 15 and EVRST 20

16. Appendix

16.1 Typical Refrigeration Systems

Refrigeration systems are basically characterized by the refrigeration cycle and the way of supplying refrigerant to the evaporator. By the refrigeration cycle, industrial refrigeration systems are categorized into three types:

Single-stage system

This is the most basic cycle: compression-condensation-expansion-evaporation.

Two-stage system

In this kind of system, compression is undertaken in two stages, typically by two compressors. Intermediate cooling is often used for optimizing the performance of the system.

Cascade system

This system is actually two basic cycles in cascade. The evaporator in the high temperature cycle acts also as the condenser of the low temperature cycle.

By the way of supplying refrigerant to evaporators, the systems could be categorized into two basic types:

Direct expansion system

The liquid/vapour mixture of refrigerant after expansion is directly fed into evaporators.

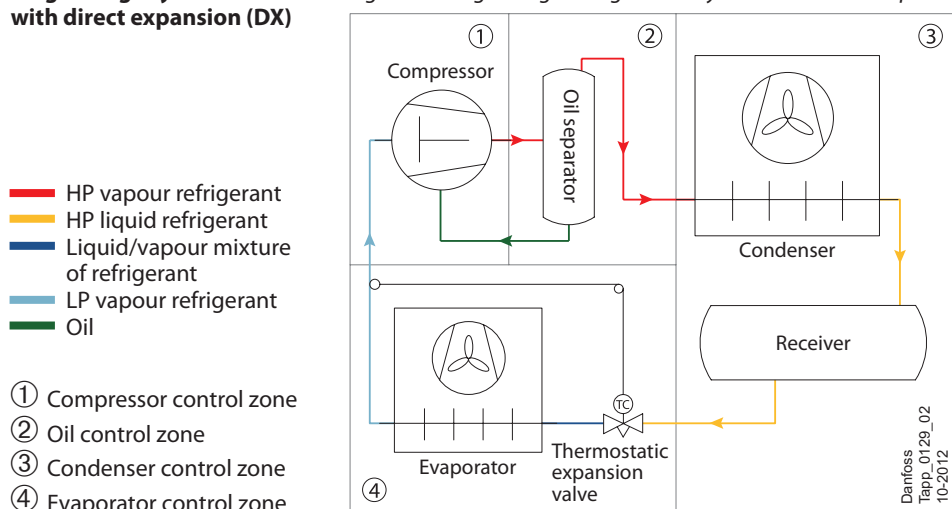
Circulated system

The liquid and vapour of refrigerant after expansion are separated in a liquid separator and only the liquid is fed into evaporators. The liquid circulation could be either gravity circulation or pump circulation.

These types of refrigeration systems will be illustrated by some examples:

Single-stage system with direct expansion (DX)

Fig. 16.1.1 Single-stage Refrigeration System with Direct Expansion



Single-stage refrigeration system with direct expansion is the most basic refrigeration system, which is very popular in air conditioning and small refrigeration systems, fig.16.1.1. The refrigeration cycle is: low pressure vapour refrigerant is compressed by the compressor into the condenser, where the high-pressure vapour condensates into high pressure liquid. The high-pressure liquid then expands through the thermal expansion valve into the evaporator, where the low pressure liquid evaporates into low-pressure vapour, and will be drawn into the compressor again.

The oil separator and the receiver have nothing to do with the refrigeration cycle, but they are important to the control:
The oil separator separates and collects the oil from the refrigerant, then sends the oil back to the compressor. This oil loop is important to secure safe and efficient running of the compressor, e.g. good lubrication. And oil control (Section 6) is essential for keeping the oil temperature and pressure at an acceptable level.

The receiver could absorb/release refrigerant when the refrigerant contents in different components vary with the load, or some components shut off for service. The receiver could also maintain a supply of liquid refrigerant at constant pressure to the expansion valve.

The thermostatic expansion valve is controlled by the superheat. This is of great importance for the functions of both the evaporator and the compressor:

- By keeping a constant superheat at the outlet of the evaporator, the thermostatic expansion valve supplies the right flow of liquid refrigerant to the evaporator according to the load.
- A certain superheat could ensure that only vapour enters the compressor suction. Liquid droplet in the suction will cause liquid hammering, which is equivalent to knocking in a motor.

Please notice that thermostatic expansion valve can only keep a constant superheat, instead of a constant evaporating temperature. Specifically, if no other controls happen, the evaporating temperature will rise with a load increase and drop with a load decrease. Since a constant evaporating temperature is the aim of refrigeration, some other controls are also necessary, e.g. compressor control and evaporator control. The **compressor control** could adjust the refrigeration capacity of the system, and the **evaporator control** could secure a right flow of refrigerant to the evaporator. Details of these two kinds of controls can be seen in Section 2 and Section 5, respectively.

Theoretically, the lower the condensing temperature, the higher the refrigeration efficiency is. But in a direct expansion system, if the pressure in the receiver is too low, the pressure difference across the expansion valve will be too low to provide enough flow of refrigerant. Therefore, controls should be designed to prevent a too low condensing pressure, if the condensing capacity of a direct expansion system is possible to vary too much. This is discussed in **Condenser Controls** (Section 3).

The main drawback of direct expansion is the low efficiency. Since a certain superheat has to be maintained:

- Part of the heat transfer area in the evaporator is occupied by vapour, and the heat transfer efficiency is lower.
- The compressor consumes more power to compress the superheated vapour than the saturated vapour.

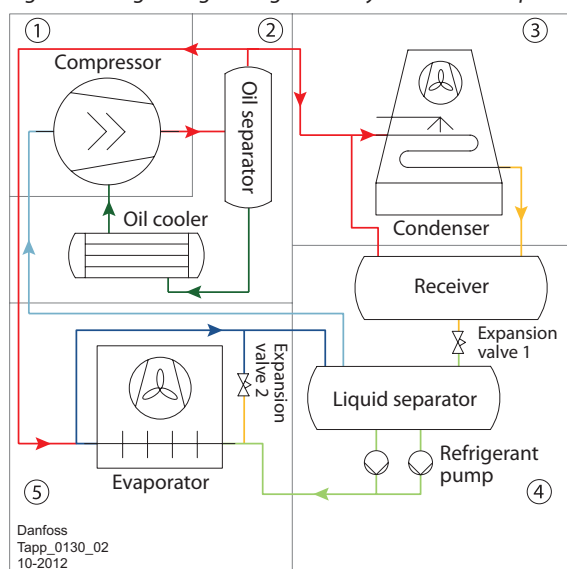
This drawback becomes especially terrible in a low-temperature refrigeration plant or a large refrigeration plant. In these refrigeration systems, circulated system with pump circulation or natural circulation is designed in order to save energy.

Single-stage system with pump circulation of refrigerant

- HP vapour refrigerant
- HP liquid refrigerant
- Liquid/vapour mixture of refrigerant
- LP vapour refrigerant
- LP liquid refrigerant
- Oil

- ① Compressor control zone
- ② Oil control zone
- ③ Condenser control zone
- ④ Liquid level control zone
- ⑤ Evaporator control zone

Fig.16.1.2 Single-stage Refrigeration System with Pump Circulation and Hot Gas Defrost



The circuit for a single-stage refrigeration system as shown in figure 16.1.2 has many similarities to the DX system shown in figure 16.1.1. The main difference is that in this system the refrigerant vapour entering the compressor suction is saturated vapour instead of superheated vapour.

This is caused by the installation of a liquid separator between the evaporator and the compressor. In the liquid separator the liquid from the liquid/vapour mix comes partly from the evaporator and partly from expansion valve 1. Only saturated vapour will pass to the compressor suction whilst only liquid is fed by the refrigerant pumps to the evaporator.

As the suction vapour is not superheated, the evaporation temperature will be lower than in a DX system. Due to the lower evaporation temperature the compressor will work more efficiently. The evaporator will provide more capacity as its surface area is used totally for cooling and not partially to superheat the refrigerant. Therefore a circulation system is more efficient than a corresponding DX system.

The line between the condenser inlet and the receiver is intended for pressure equalisation to ensure that the condensing liquid from the condenser can run to the receiver without problems.

In pump circulation systems it is important to keep the pump running, i.e. that the pump operation is not unintentionally interrupted. Therefore pump control is important to ensure that the pump has the correct pressure difference, that a constant supply of liquid is ensured and that the condition of the pump is not compromised. This subject is discussed in Section 8.

In circulation systems there is no superheating which can be used as a control variable for a thermostatically controlled expansion valve operation.

Expansion Valve 1 is usually controlled by the level in the liquid separator or sometimes by the level in the receiver/condenser. This is also called liquid level control, which is discussed in Section 4.

If the evaporators are of a fin and tube design and used with air and if the evaporation temperature is below 0°C, a layer of frost/ice builds up on the evaporator surface which originates from the water/moisture present in the air. This layer must be removed regularly as otherwise it will restrict the evaporator airflow and reduce the evaporator capacity.

Possible defrosting methods are hot gas, electrical heat, air and water. In figure 17.1.2 hot gas is used for defrosting. Part of the hot gas from the compressor is led to the evaporator for defrosting.

The hot gas warms up the evaporator and melts the ice layer on the evaporator and simultaneously the hot gas condenses and becomes high-pressure liquid. Using an overflow valve, this high-pressure liquid can be returned to the liquid separator in the suction pipe.

Hot gas defrosting can only be used in systems that contain at least three parallel evaporators.

During defrosting, at least two of the evaporators (by capacity) must be cooling and a maximum of one evaporator should be defrosting – otherwise there is insufficient hot-gas available for the defrosting process.

The method for switching between refrigeration and defrosting cycles is discussed in the section on evaporator control (Section 5).

Not all valves are shown.
Not to be used for construction purposes.

Two-stage system

A typical two-stage system is shown in fig 16.1.3. Part of the liquid refrigerant from the receiver first expands into the intermediate pressure, and evaporates to cool the other part of liquid refrigerant in the intermediate cooler.

The intermediate-pressure vapour is then directed into the discharge line of the low-stage pressure, cools the low-stage discharge vapour, and enters the high-stage compressor.

The power used to compress this part of vapour from the suction pressure into the intermediate pressure is saved and the discharge temperature of the high-stage compressor is lower.

So the two-stage system is especially suitable for low-temperature refrigeration system, for the high efficiency and low discharge temperature.

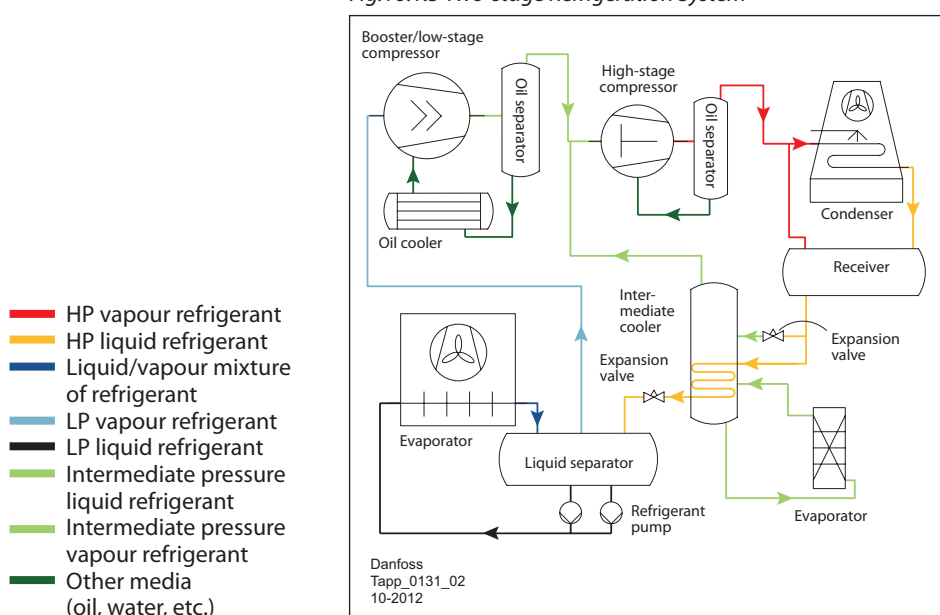
The intermediate cooler could also supply refrigerant to intermediate-temperature evaporators. In fig. 16.1.3, the intermediate supply refrigerant to the plate type evaporator by gravity circulation.

Compared with pump circulation, gravity circulation is driven by the thermosyphon effect in the evaporator, instead of the pump. Natural circulation is simpler and more reliable (on pump failure), but the heat transfer is generally not as good as the pump circulation.

Two-stage system could be theoretically effective. However, it difficult to find a kind of refrigerant that is suited for both the high temperature and the low temperature in low-temperature refrigeration systems.

At high temperatures, the refrigerant pressure will be very high, posing high requirement on the compressor. At low temperatures, the refrigerant pressure may be vacuum, which leads to more leakage of air into the system (the air in the system will reduce heat transfer of the condenser, see Section 9.3). Therefore, cascade system may be a better choice for low refrigeration system.

Fig. 16.1.3 Two-stage Refrigeration System



Not all valves are shown.
 Not to be used for construction purposes.

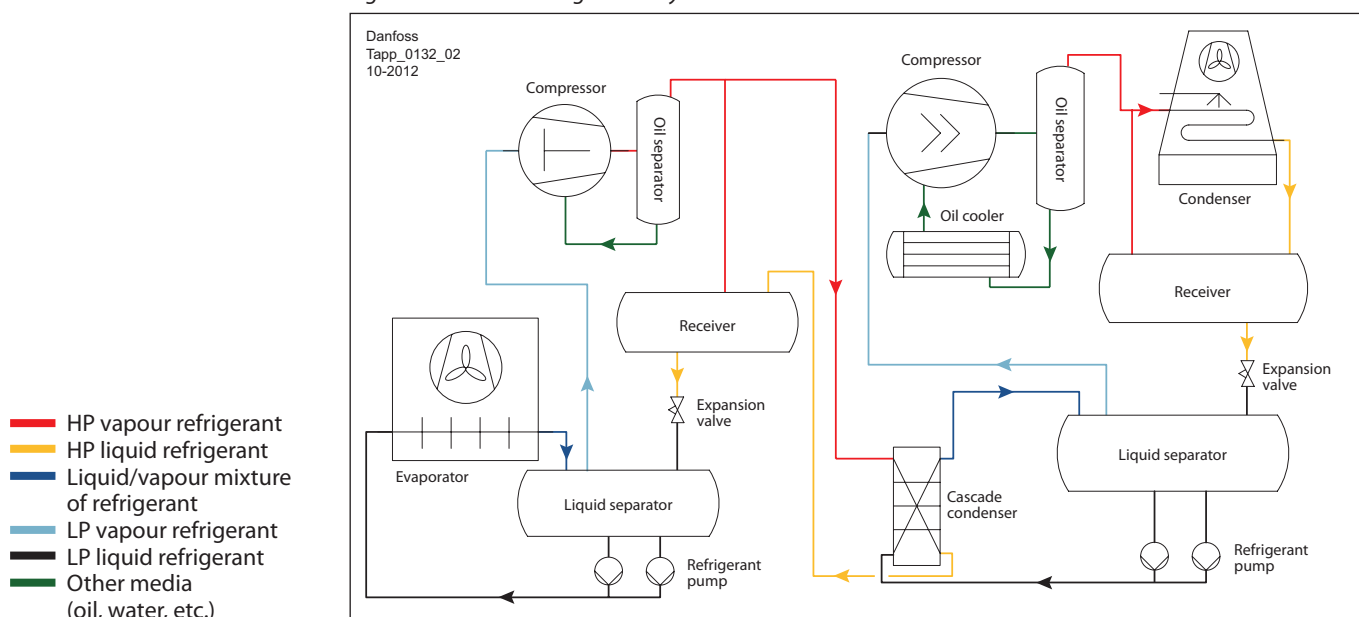
Cascade system

A cascade system consists of two separate refrigeration circuits, as shown in fig. 16.1.4. A cascade condenser interconnects the two circuits by acting as both the condenser of the high temperature circuit and the evaporator of the low temperature circuit.

The refrigerant for the two circuits could be different, and optimized for each circuit. For example, the refrigerant could be NH₃ for the high temperature circuit and CO₂ for the low temperature circuit.

This CO₂/NH₃ system needs less charge of ammonia and proves to be more efficient in low temperature refrigeration than a similar two-stage ammonia system.

Fig.16.1.4 Cascade Refrigeration System



17. ON/OFF and modulating controls

Detailed below is the basic theory for ON/OFF and modulating control. The intention is to provide a basic understanding of control theory

and the technical terms used. Furthermore some practical advice will also be given.

Abbreviations and definitions

P	Proportional
I	Integration
D	Derivative
PB	Proportional Band [%] in a P, PI or PID controller. Number in percent, that Process variable (PV), has to change, in order for the controller to change the output (y) from 0 to 100 %
K_p	Amplification factor in a P, PI or PID controller
T_i	Integration time [s] in a PI or PID controller
T_d	Differential time [s] in a PID controller
PID	A typical controller that includes both P, I and D functions
SP	Set point
PV	Process Variable (the controlled parameter: temperature, pressure, liquid level, etc)
offset (x)	Difference between Set point (SP) and Process Variable (PV)
y	Calculated output of a controller.
dead time	If Process Variable (PV) measurement is physically mounted thus the signal is always has a time delay, compared to if Process Variable (PV) measurement was installed locally without time delay.

References

- [1] Reguleringsteknik, Thomas Heilmann / L. Alfred Hansen

17.1 ON/OFF control

In some cases a control application in practice can be achieved with ON/OFF control. This means that the regulating device (valve, thermostat) only has two positions: contacts closed or open. This control principle is called ON/OFF control. Historically ON/OFF was employed widely within refrigeration, particularly in refrigerators equipped with thermostats.

However ON/OFF principles can also be used in advanced systems where PID principles are used. E.g. is an ON/OFF valve (i.e. Danfoss type AKV/A) used to control superheat with PID available parameters on the dedicated electronic controller. (Danfoss type EKC 315A)

An ON/OFF controller will only react within some given limit values, like e.g. Max and Min. Outside these limit values an ON/OFF controller can not carry out any action.

Normally ON/OFF is used because:

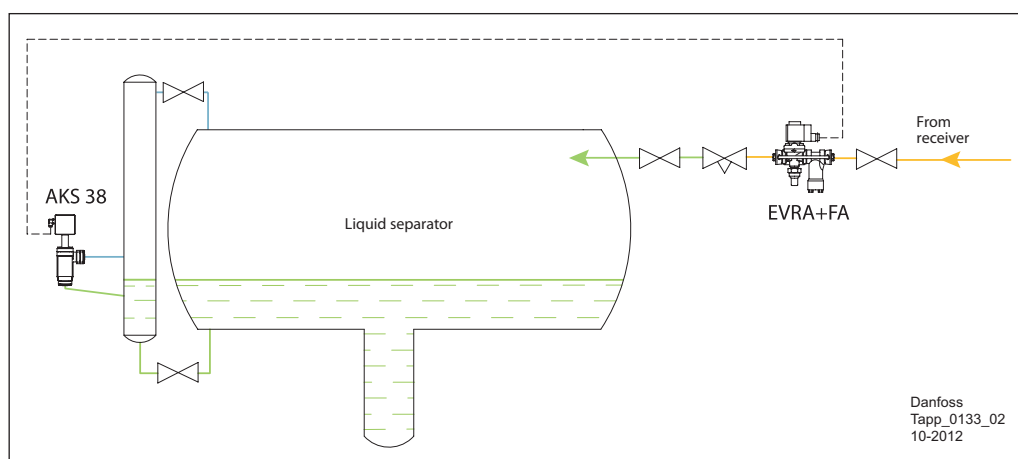
- Low price, less complicated system, no feedback loop.
- It can be accepted that PV varies a little from SP, along with that the ON/OFF device is operating.
- The process has so big capacity that the ON/OFF operation does not have any influence on PV
- In systems with dead time, ON/OFF control can be advantageous

In ON/OFF systems you will have a feed back, as for modulating systems, but, characteristic of ON/OFF systems is that PV varies and the system is not able to eliminate any offset.

Application example 17.1.1 ON/OFF control

To control liquid level between a minimum and a maximum level an ON/OFF device can be used like Danfoss type AKS 38. AKS 38 is a float switch that can control the switching of ON/OFF solenoid valves.

- HP liquid refrigerant
- LP vapour refrigerant
- LP liquid refrigerant



Not all valves are shown.
Not to be used for construction purposes.

17.2 Modulating control

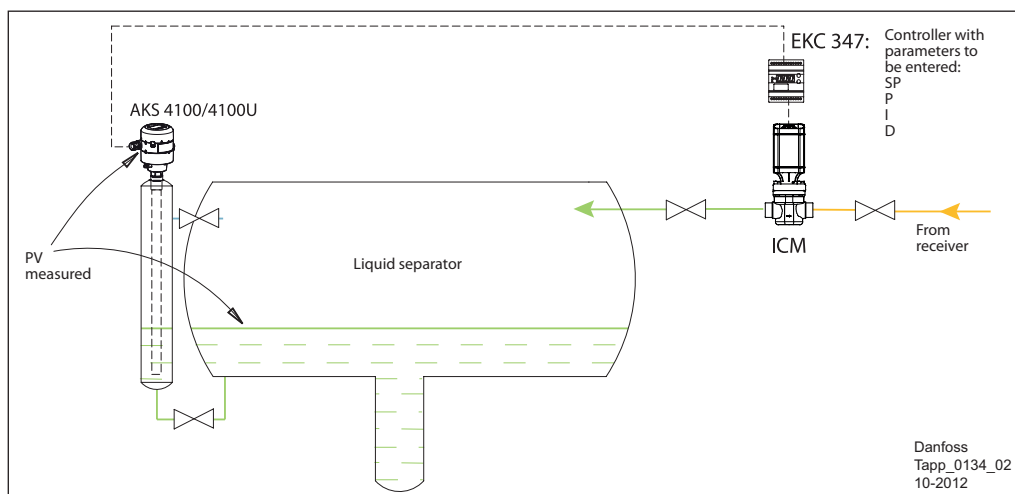
The main difference between modulating controls and ON/OFF systems is that modulating systems will constantly react when there is a change of PV.

like P, I and D. This gives a high degree of flexibility which again is very useful because the controller can then be adjusted to suit different applications.

Furthermore electronic controller provide the flexibility to change different control parameters,

Application example 17.2.1 Modulating control

HP liquid refrigerant
LP liquid refrigerant



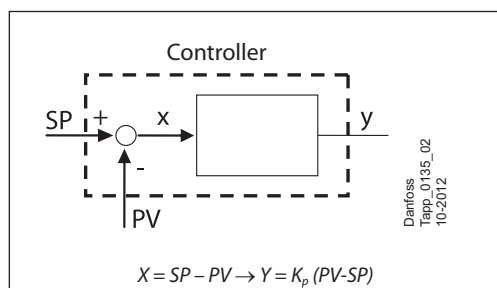
Basic P, I and D principles

Generally, in most common controllers there is the facility to adjust parameters for P, PI, or PID settings

- In a P controller it is possible to adjust: PB or K_p ;
- In a PI controller it is possible to adjust: PB or K_p and T_i ;
- In a PID controller it is possible to adjust: PB or K_p and T_i and T_d .

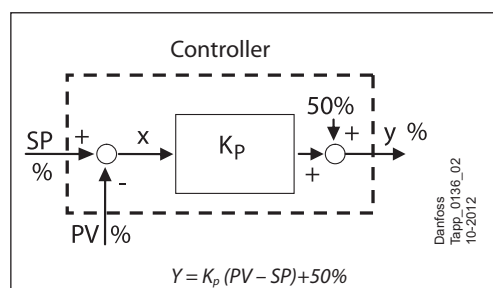
P-controller

In every controller a P component exists. In a P-controller there is a linear relation between input and output.



Practical P-controllers are designed so when $SP=PV$ the controller must give an output that corresponds to the normal load of the system.

Normally this means that the output will be 50 % of max output. E.g. a motorized valve will over time run in 50 % opening degree in order to maintain SP.



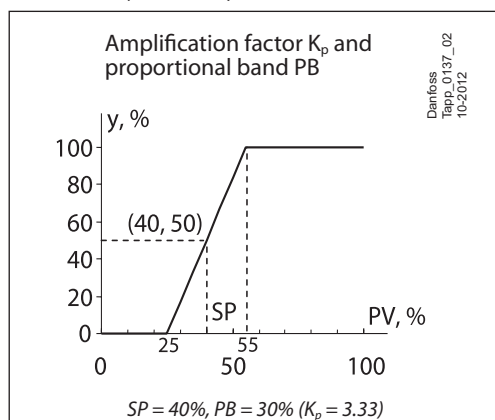
Some controllers do not use PB, but K_p . The relation between PB and K_p is:
 $PB[\%] = 100/K_p$

Please observe that PB can be bigger than 100%, corresponding to that K_p is less than 1.

Not all valves are shown.
Not to be used for construction purposes.

17.2 Modulating control (continued)

P-controller (continued)



When PV = SP = 40% the regulator gives an output (y) of 50%. (This means that the valve has an opening degree of 50%).

If PV increases to 46%, there is a deviation between PV and SP of 6%. As K_p is assumed to be 3.33, a deviation of 6% means that output increases by $6\% \times 3.33 = 20\%$, i.e. if PV rises to 46%, the output increases to $50\% + 20\% = 70\%$.

The deviation of the 6% is a deviation that a P regulator cannot overcome. The resulting deviation stems from the basic function of a P regulator.

In order to achieve a minimum deviation it is important that the regulation device (the valve) is shaped so that the output (y) from the regulator can control the process so that it is equal to the standard average load. Then the deviation will always be as small as possible and will in time approach zero.

P-controller adjustment characteristics

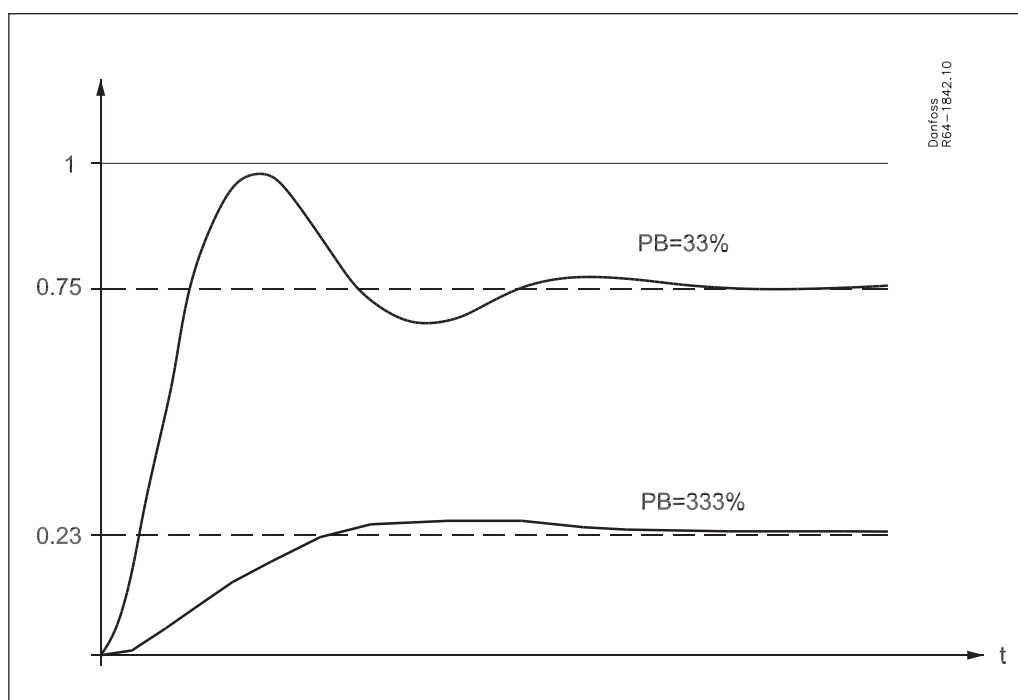
P is the primary control component. In most cases, P will create a permanent offset that can be insignificant small, but also unacceptable big. However a P control is better than none (no feedback, no closed loop).

Change of PB has two important effects:

- Smaller PB (bigger amplification) gives less offset, i.e. better effect against load changes, but also increased tendency to fluctuations.
- Bigger P-band (smaller amplification) gives more offset, but less tendency to fluctuations.
- Smaller PB means that theoretically the control is approaching ON/OFF operation.

Below drawing is of universal validity for straight forward P controlled loop.

It shows the different responses by a loop having PB = 33% and PB = 333% when the P controlled loop is influenced by SP is changed by +1 unit.



17.2 Modulating control (continued)

I-controller

The most important characteristic for an I-controller is that it eliminates offset, and that is why it is used. I-controller continues to change its output as long as offset exists. However the ability to fully remove offset is linked to that it in practice, is proportioned correctly.

I-controller's good property to remove offset has also a negative action: It will increase the tendency to fluctuations in a control loop.

Basically the tendency to fluctuations is worse for an I-controller than a P-controller.

The ability to counteract on load changes is slower for an I-controller than a P-controller.

PI controller

The combination of advantages and disadvantages for both P and I makes it advantageous to combine P and I into a PI-controller.

In a PI controller it would be possible to adjust: PB and T_i. T_i is normally entered in seconds or minutes.

When T_i has to be entered, it has to be compromise between stability and elimination of offset.

Decreased T_i (bigger integration influence) means faster elimination of offset, but also increased tendency to fluctuations.

D-controller

The most important characteristic for a D-controller (derivative) is that it can react on changes. This also means that if a constant offset is present, a D-controller will not be able to do any action to remove the offset. D-component makes the system fast respond on load changes.

D effect improves stability and makes the system faster. It does not have any significance for offset, but it works to make tendency to fluctuations smaller. D reacts on changes in the error and the loop reacts faster against load changes than without D. The fast reaction on changes means a damping of all fluctuations.

In controllers with D influence the T_d can be adjusted. T_d is normally entered in seconds or minutes.

It has to be observed not to make T_d too big, as then the influence, when e.g. changing SP, will be too dramatic. During start-up of plants it may be advantageous simply to remove the D influence. (T_d=0)

The above means that a D-controller will never be used alone. Its typical use is in combination as PD or PID with its ability to damp fluctuations.

PID-controller

The combination of all three components into a PID controller has become of general use.

The general guidelines / properties for a PID controller are:

- Decreased PB improves offset (less offset), but the stability is worse;
- I component eliminates offset. Bigger I (less T_i) makes faster elimination of offset.

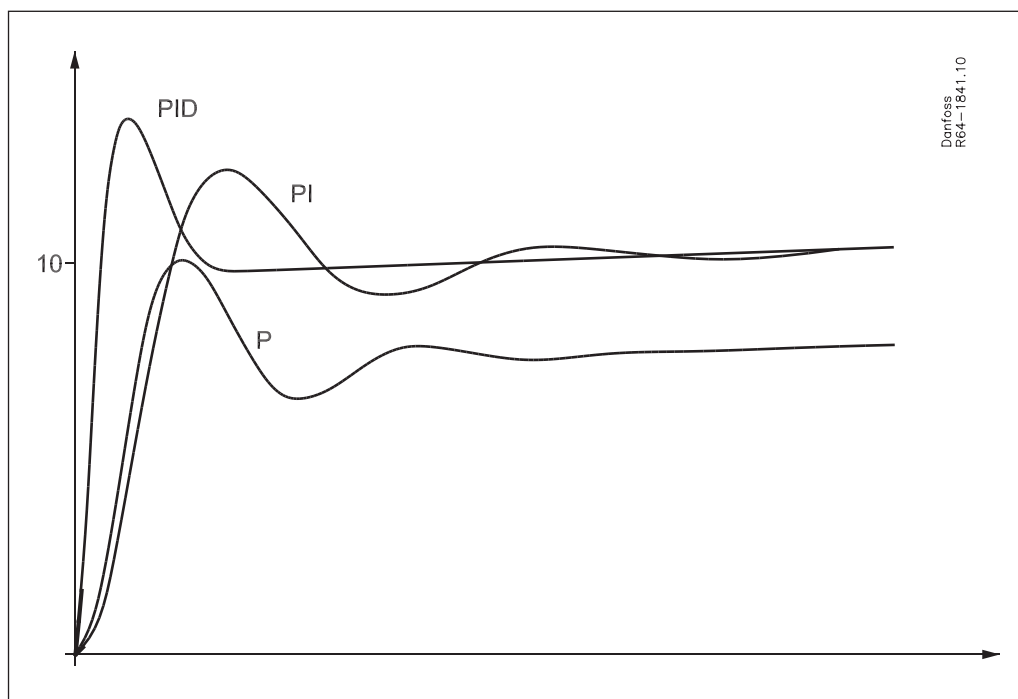
- I component increases the tendency to fluctuations.
- D component damps the tendency to fluctuations and makes the control faster. Bigger D (bigger T_d) the stronger influence on above, however until a specific limit. A too big T_d will mean that it reacts too strong on sudden changes, and the control loop becomes unstable.

17.2 Modulating control (continued)

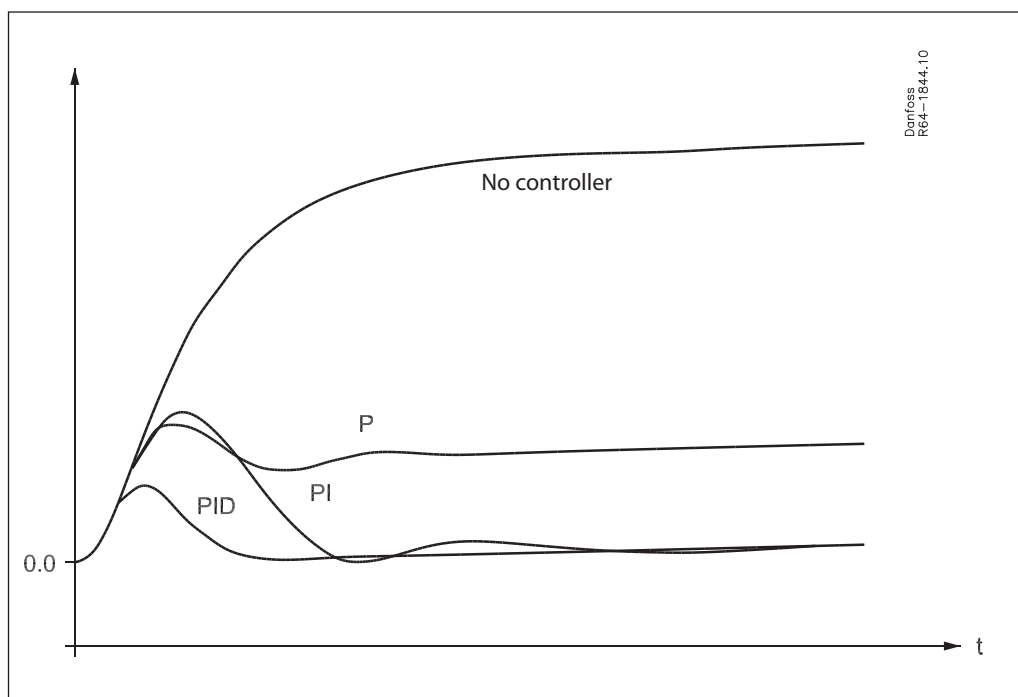
Typical PID transient state curves 1: optimal PID settings

The settings:

	PB	T _i	T _d
P	66.7 %	-	-
PI	100 %	60 s	-
PID	41.7 %	40 s	12 s



Above displays the different controls principles, when is influenced by SP is changed by +1 unit.



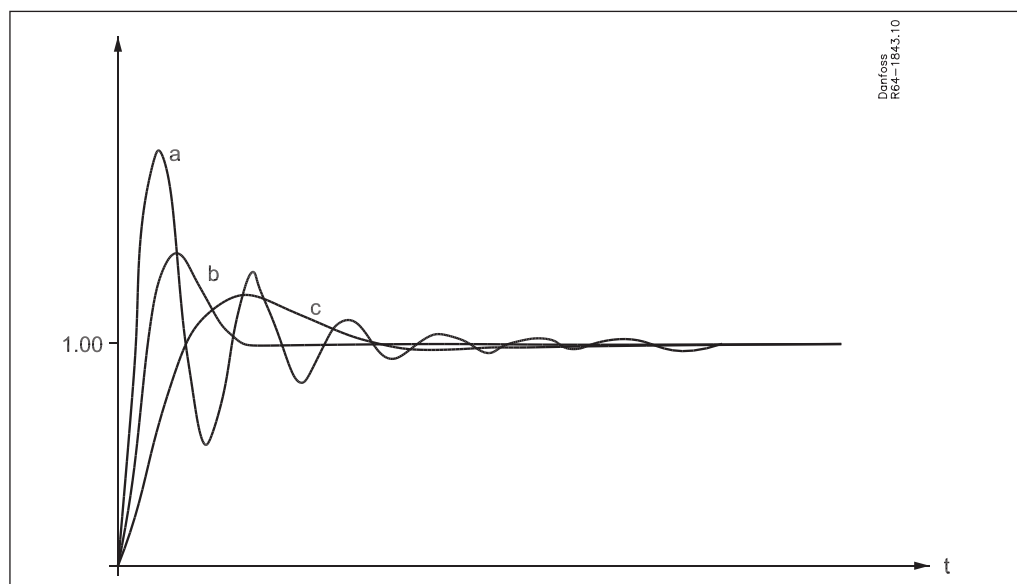
Same settings as above. Exposed to a load change of 1.

17.2 Modulating control (continued)

Typical PID transient state curves 2: change of PB

The settings:

	PB	T _i	T _d
PID-a	25.0 %	40 s	12 s
PID-b	41.7 %	40 s	12 s
PID-c	83.3 %	40 s	12 s



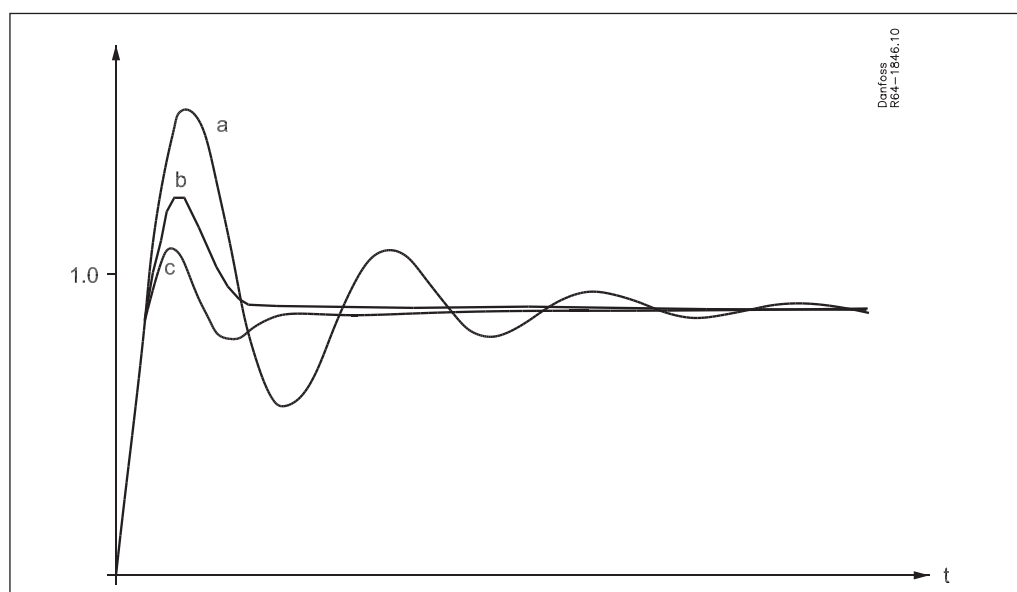
Above shows variation of PB for PID control when is influenced by SP is changed by +1 unit. From above it is clear when PB is too small the systems

becomes more unstable (oscillatory). When PB is too big it becomes too slow.

Typical PID transient state curves 3: change of T_i

The settings:

	PB	T _i	T _d
PID-a	41.7 %	20 s	12 s
PID-b	41.7 %	40 s	12 s
PID-c	41.7 %	120 s	12 s



Above shows variation of T_i for PID control when is influenced by SP is changed by +1 unit. From above it is clear when T_i is too small the systems

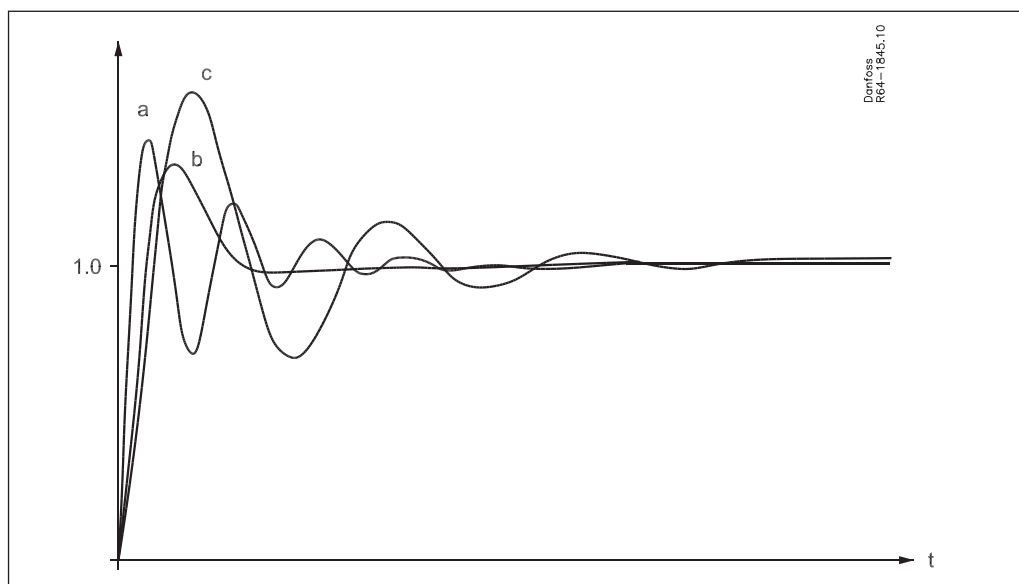
becomes more unstable (oscillatory). When T_i is too big it takes a very long time to eliminate the last offset.

17.2
Modulating control
(continued)

Typical PID transient state curves 4: change of T_d

The settings:

	PB	T_i	T_d
PID-a	41.7 %	40 s	24 s
PID-b	41.7 %	40 s	12 s
PID-c	41.7 %	40 s	6 s



Above shows variation of T_d for PID control when is influenced by SP is changed by +1 unit. From above it is clear when T_d is either too small or too

big compared to the optimal ($T_d=12$) the systems become more unstable (oscillatory).

Reference Documents -
Alphabetical overview

Type	Title	Technical leaflet / Manual	Product instruction	
AKD 102	Variable speed drive	PD.R1.B	MG11L	
AKS 21	Temperature sensor	RK0YG	RI14D	
AKS 32R	Pressure transmitter		PI.SB0.A	
AKS 33	Pressure transmitter	RD5GH	PI.SB0.A	
AKS 38	Float switch	PD.GD0.A	PI.GD0.A	
AKS 4100/4100U	Liquid level sensor	PD.SC0.C	PI.SC0.D	PI.SC0.E
AKVA	Electrically operated expansion valve	PD.VA1.B	PI.VA1.C	PI.VA1.B
AMV 20	Three point controlled actuator	ED95N	EI96A	
BSV	Safety relief valve	PD.IC0.A	PI.IC0.A	
CVC-XP	Pilot valves for servo operated main valve	PD.HN0.A	PI.HN0.A	
CVC-LP	Pilot valves for servo operated main valve	PD.HN0.A	PI.HN0.M	
CVP	Pilot valves for servo operated main valve	PD.HN0.A	PI.HN0.C	
CVPP	Pilot valves for servo operated main valve	PD.HN0.A	PI.HN0.C	
CVQ	Pilot valves for servo operated main valve	PD.HN0.A	PI.VH1.A	
DCR	Filter drier	PD.EJ0.A	PI.EJ0.B	
DSV	Double stop valve (for safety valve)	PD.IE0.A	PI.IE0.A	RI.7D.A
EKC 202	Controller for temperature control	RS8DZ	RI8JV	
EKC 315A	Controller for control of industrial evaporator	RS8CS		
EKC 331	Capacity controller	RS8AG	RI8BE	
EKC 347	Liquid level controller	PS.G00.A	PI.RP0.A	
EKC 361	Controller for control of media temp.	RS8AE	RI8BF	
EVM	Pilot valves for servo operated main valve	PD.HN0.A	PI.HN0.N	
EVRA / EVRAT	Solenoid valve	PD.BM0.B	PI.BN0.L	
FA	Strainer	PD.FM0.A	PI.FM0.A	
FIA	Filter	PD.FN1.A	PI.FN1.A	
GD	Gas detection sensor	PD.S00.A	PI.S00.A	
GPLX	Gas powered stop valve	PD.B00.A	PI.B00.A	
HE	Heat exchanger	PD.FD0.A	PI.FD0.A	
ICF	Control solution	PD.FT1.A	PI.FT0.C	
ICM / ICAD	Motor operated valve	PD.HT0.B	PI.HT0.A	PI.HT0.B
ICS	Servo operated valve	PD.HS2.A	PI.HS0.A	PI.HS0.B
KDC	Compressor discharge valve	PD.FQ0.A	PI.FQ0.A	
LLG	Liquid level glass	PD.GG0.A	PI.GG0.A	
MLI	Sight glass	PD.GH0.A	PI.GH0.A	
MP 55 A	Differential pressure control	PD.CG0.B	PI.CG0.E	
NRVA	Check valve for ammonia	PD.FK0.A	PI.FK0.A	
OFV	Overflow valve	PD.HQ0.A	PI.HX0.B	
ORV	Oil regulating valve	PD.HP0.B	PI.HP0.A	
PMFL / PMFH	Modulating liquid level regulator	PD.GE0.C	PI.GE0.D	PI.GE0.A
ICLX	Solenoid valve, two-step on/off	PD.HS1.A	PI.HS1.A/B	
POV	Pilot operated internal safety valve	PD.ID0.A	PI.ID0.A	
QDV	Quick oil drain valve	PD.KL0.A	PI.KL0.A	
REG-SA/SB	Hand regulating valve	PD.KM1.A	PI.KM1.A	
RT 107	Differential thermostat	PD.CB0.A		
RT 1A	Pressure control, differential pressure control	PD.CB0.A	RI5BC	
RT 260A	Pressure control, differential pressure control	PD.CB0.A	RI5BB	
RT 5A	Pressure control, differential pressure control	PD.CB0.A	RI5BC	
SCA-X	Stop check valve / check valve	PD.FL1.A	PI.FL1.A	
SFA	Safety relief valve	PD.IF0.A	PI.IB0.A	
SGR	Sight glass	PD.EK0.A	PI.EK0.A	
SNV	Stop needle valve	PD.KB0.A	PI.KB0.A	
SV 1-3	Modulating liquid level regulator	PD.GE0.B	PI.GE0.C	
SV 4-6		PD.GE0.D	PI.GE0.B	
SVA-S/L	Stop valve	PD.KD1.A	PI.KD1.A	
TEA	Thermostatic expansion valve	PD.AJ0.A	PI.AJ0.A	
TEAT		PD.AU0.A	PI.AU0.A	
VM 2	Pressure balanced valve	ED97K	VIHBC	
WVS	Water valve	PD.DA0.A	PI.DA0.A	

To download the latest version of the literature please visit the Danfoss website.

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