Guntner
CO$_2$
unit coolers

Special properties of carbon dioxide CO_2 as a refrigerant

1. General

With the use of CFCs being abandoned, it has been realised that carbon dioxide (CO_2) can be used in its capacity as a "natural" refrigerant in an increasing number of cases and also in a wider range of applications. As a refrigerant belonging to Group L1, Safety Group A1 (as specified in EN 378-1, Appendix E), CO_2 is a nonflammable refrigerant which has no significantly adverse effects on employees. CO_2 is colorless, odorless and tasteless. It is nontoxic and has been approved of for use in foodstuffs and does not need to be declared. CO_2 is an environmentally compatible refrigerant (ODP = 0, GWP = 1). Localized hazards only occur in the case of high concentrations in the breathing air (3-10 %), resulting in irritation of the respiratory system or even unconsciousness due to blocked breathing. In comparison, there are global hazards associated with CFCs and HFCs (GWP >>1). The cost-effectiveness of CO_2 (extremely high volumetric refrigerating capacity) – even though this applies only to the subcritical range (approx. 7 to 9 times higher than R22 and NH_3) – means that, in view of the suitability of CO_2 compared with HFC refrigerants and ammonia, there is a useful alternative for the further increased use of CO_2.

2. Abandoning the use of CFCs

The process to abandon the production and use of ozone-damaging CFCs has been internationally accelerated in view of the dramatic decrease in the concentration of stratospheric ozone above the Antarctic and the proven weakening of the ozone layer in the northern hemisphere. The obvious substitutes were HFC because they are chemically very closely related to CFC. They also have very similar properties in refrigeration systems and thus they helped to minimize development work (table 1). Due to the direct greenhouse warming potential (GWP>0) that is still inherent in

them, these substances continue to be considered to be ecologically harmful. So, parallel to the introduction of HFC, the operation of those refrigeration units was encouraged or re-discovered, that can be used with the so-called "natural" refrigerants like ammonia, hydrocarbons, water,

CO_2 and air (table 2). They are considered to be very important substances for many applications in the refrigeration technology.

Alternative refrigerants	Refrigerant to be replaced
HFC: R134a, R32, R125	R12, R22, R502
HFC mixtures: e.g. R404A, R407C, R 410A, R 507	R12, R22, R502
Hydrocarbons: R170 (ethane) R270 (cyclopropane) R290 (propane) R1270 (propylene) R600 (butane) R600a (isobutane) Isopentane	R13 R12, R502 R12, R502 R12, R502 R114 R12 R11
Ammonia (NH_3) (R717)	R12, R22, R502

Table 1: Alternative refrigerants

Alternative refrigerants	Applications
Ammonia (NH_3)	Large refrigeration systems; t_0 = -45...+10 °C Q_0 = 200...5000 kW, with just <50 kW also being possible
Hydrocarbons, in particular propane (R290) isobutane (R600a)	Small refrigeration systems; t_0 = -50...+10 °C Filling max. 1 kg Domestic refrigerators, beer refrigeration systems, warehouse refrigeration
Water	Large refrigeration systems; $t_0 \geq \pm0$ °C Very low pressures, high volume flows, Aqua turbo systems (ILK); (no knowledge of reliable systems)
Carbon dioxide (CO_2)	Car air-conditioning, air-conditioning a. refrigeration for transportation, commercial and industrial refrigeration t_0 = -50...+10 °C
Air	Aircraft air-conditioning, train air-conditioning; $t_0 \leq$ -40 °C

Table 2: "Natural" alternative refrigerants

3. Carbon dioxide CO_2 as a refrigerant

3.1 The properties of CO_2

CO_2, which has been used as a refrigerant since 1850 (British Patent by Alexander Twining), has a long tradition in refrigeration engineering. Its safe properties (non-flammable, nontoxic and chemically inactive) were the reasons for its widespread use in the following years in the 1920s and early 1930s, in particular as the preferred refrigerant on board ships and in industrial applications. With the introduction of the so-called "safe refrigerant" (CFC in 1928), CO_2 was pushed out of the market. Whereas ammonia (NH_3) continued to be used as the main refrigerant in industrial refrigeration systems (breweries, slaughterhouses, chemical industry, etc.), CO_2 disappeared completely from the market in the 1950s.

The main reasons for its disappearance are several special characteristics such as the extremely high pressure situation caused by the very low critical temperature.

As a result of the move away from CFC refrigerants (see Section 2) and, at the same time, the dramatic restrictions on the filling quantities in ammonia refrigeration systems at the beginning of the 1990s (IV. BImSchV; that is IV. Federal Immission Control Regulation), the refrigerant CO_2 started to make a comeback.

This was due to its advantages regarding environmental compatibility, cost-effectiveness, recycling and disposal and it was considered as a potential alternative for use in refrigeration systems.
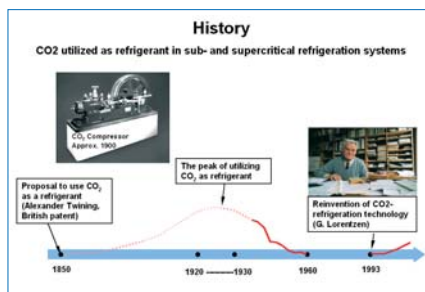

Fig. 1: CO_2; The history of the use of CO_2
in refrigeration systems. (Source: Danfoss)

CO_2 is a "natural refrigerants" (see above). The other natural refrigerants mentioned above have certain disadvantages:
✱ NH_3 is toxic and flammable

CO_2 properties compared with various refrigerants

Refrigerant	CO_2	R134a	R404A	NH_3
Flammable or explosive	NO	NO	NO	(YES)
Toxic	NO	NO	NO	YES
Natural substance	YES	NO	NO	YES
Ozone Depletion Potential (ODP)*	0	0	0	0
Global Warming Potential (GWP)*	1	1,300	3,260	0
Critical point bar °C	73.6 31.1	40.7 101.2	37.3 72	113 132.4
Triple point bar °C	5.18 -56.6	0.004 -103	0.028 -100	0.06 -77.7

Fig. 2: Comparison with other refrigerants

✱ Hydrocarbons are flammable or explosive
✱ H_2O has a limited range of applications (freezing point at 0° C, low saturation pressure at low temperatures, low volumetric refrigerating capacity)

In comparison with them CO_2 is neither toxic nor flammable.

CO_2 also differs from other conventional refrigerants (e.g. HFC) in many aspects and has some very special characteristics. The characteristic properties, which distin-
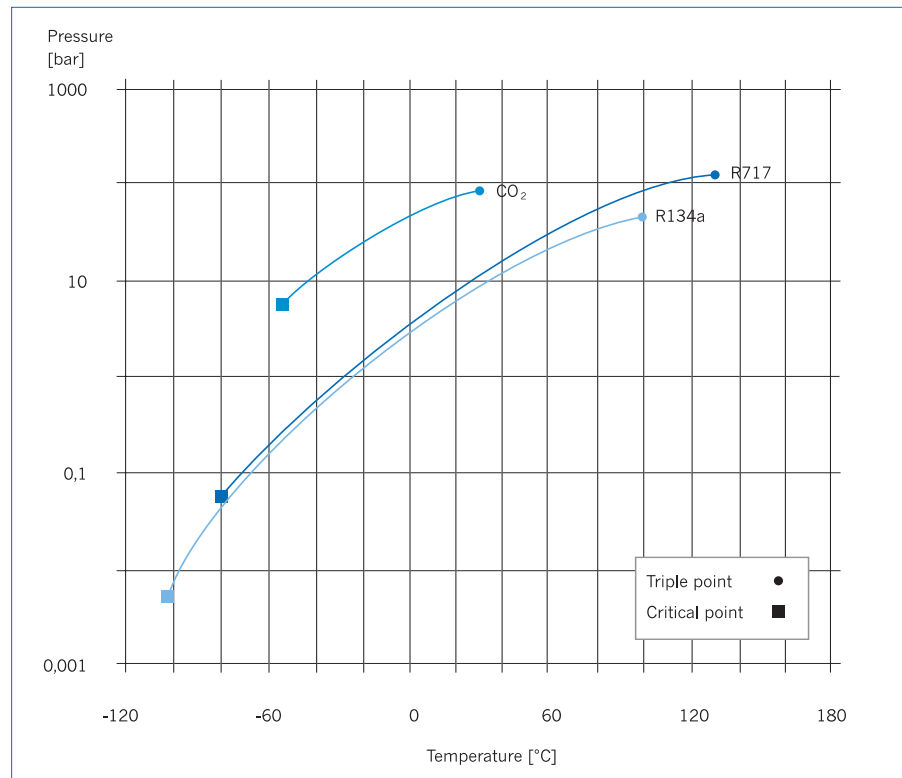

Fig. 3: CO_2: Pressure/temperature diagram

CO$_2$: Important properties

Chemical formula	CO$_2$ (covalent bond O = C = O)
Molar mass	44.011 kg/kmol
Molar standard volume	22.263 m³/kmol
Special gas constant	0.1889kJ/kgK
Normal density	1.977 kg/m³
Density ratio of CO$_2$ to air in normal state	1.529
Critical temperature	+31 °C
Critical pressure	73.83 bar
Critical density	466 kg/m³
Sublimation point	-78.9 °C at 0.981 bar
Triple point	-56.6 °C at 5.18 bar
Decomposition temperature	Above approx. 1200 °C with a decomposition rate of 0.032 % vol.
Color in the gas phase	Colorless
Flammability	Nonflammable, well-known as a fire extinguishing agent
Reaction in normal conditions	Stable compound. Used as an inert gas in normal atmospheric conditions.
Reaction with water	CO$_2$ + H$_2$O = H$_2$CO$_3$ When CO$_2$ gas is dissolved in water, only approx. 0.1 % is present in the acid H$_2$CO$_3$. The pH value of aqueous CO$_2$ solutions at normal pressure is 3.7. The value falls to its limit of 3.3 under pressure. CO$_2$ is therefore very suitable for use as a neutralisation agent for alkaline solutions.
Odor	Odorless
Taste	Neutral
Toxicity	Nontoxic, approved for foodstuffs, no declaration necessary
MAC value	5,000 ml/m³ (ppm) on average during an 8 hour period. A peak concentration of 10,000 ppm (momentary value) is permissible 3 times per shift for a period of 60 minutes.
Medical application	Inhalation of 3-5 % CO$_2$ in respiratory gas
Practical experience with high concentrations of CO$_2$ in the breathing air	Irritation of the respiratory system at 30,000 to 50,000 ppm (3-5 % vol.). Unconsciousness at 70,000 to 100,000 (7-10 %) due to lack of oxygen.

Table 3: CO$_2$: Important properties

Fig. 4: CO$_2$; Phase diagram

Fig. 5.: CO$_2$; log p, h-diagram

guish CO$_2$ from other conventional refrigerants, are the following:

* Higher operating pressure at a given temperature
* Narrow range for the operating temperature
* Triple point at a much higher pressure
* Critical point at a much lower temperature

The technical development since the 1920s has improved the possibility of using CO$_2$ but users still have to be aware of the characteristic properties of CO$_2$ and make the necessary provisions in their CO$_2$ refrigeration systems to avoid the problems involved.

With conventional refrigerants the triple point and the critical point are not usually of any importance but in CO$_2$ refrigeration systems it is essential that attention be paid to them:
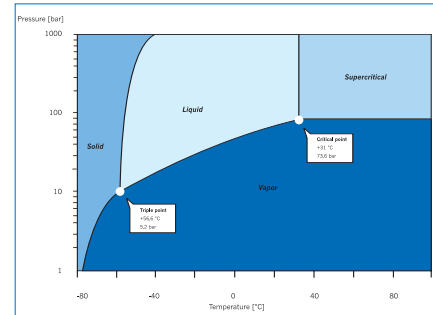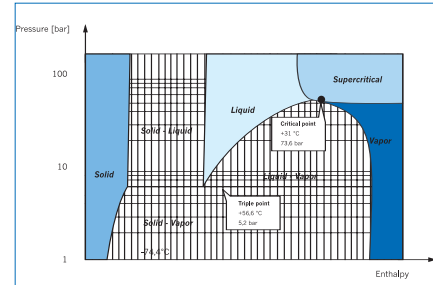
* The triple point is very high at 5.2 bar, much higher than atmospheric pressure. This circumstance causes problems if the necessary safety precautions are not taken.
* The critical point is very low at 31.1 °C and this affects the design of the system.

CO$_2$ occurs in three phases:
* solid
* liquid
* vapor

The CO$_2$ phase diagram shows that there are points at which CO$_2$ can be present in two different phases in an equilibrium:
* solid and vapor
* liquid and vapor
* solid and liquid

At atmospheric pressure, CO$_2$ can only occur as a solid or as a vapor. At this pressure, CO$_2$ cannot be liquid. Below -78.4 °C, CO$_2$ is the solid known as "dry ice". Above this temperature, CO$_2$ sublimes directly from its solid to its vapor phase.

At 5.2 bar, CO$_2$ reaches a characteristic state, its triple point. At this point all three phases (solid, liquid, vapor) exist together in an equilibrium.

CO$_2$ reaches its critical point at +31.1 °C. At this temperature the densities of the liquid and the vapor are equally high. Consequently, the difference between the two

Refrigeration properties									
Refri-gerant	Appli-cation[1]	Evap. temp. at 1.013 bar	Evap. pressure at 0 °C	Condens. pressure at +45 °C	Condens. pressure at +55 °C	Density at +45 °C liquid	Latent heat at boiling point	Volumetric refriger. capacity ±0/±40 °C	Refrigerant group/ safety group
		°C	bar	bar	bar	kg/m³	kJ/kg	kJ/m³	
R744 (CO_2) [2]	D	(-78.4) / -57 [4]	79.4	- -	- -	1,029 [5]	~ 335	4,650 [3]	L1 / A1
[1] D: Deep-freezing (- 50 to -20 °C)				[4] Triple point at 5.2 bar $\hat{=}$ -57 °C					
[2] Operation with cascade refrigeration system only				[5] At –20 °C					
[3] At –50 / -10 °C									

Table 4: CO$_2$; Refrigeration properties

phases disappears and a new phase, the supercritical phase, occurs. (See the log p, h diagram with the solid, liquid, vapor and supercritical phases marked.)

In Germany CO_2 is classified as a refrigerant in Group L1, Safety Group A1 (as specified in EN 378-1) on account of its low risks (nonflammable and without significant harmful effects on humans). (See figure 6 and table 4.)

Nevertheless it must be remembered that CO_2 displaces the air and blocks breathing. If sufficient oxygen is available large quantities of CO_2 have a narcotic effect. In small quantities, CO_2 has a stimulating effect on the respiratory system. Due to the acidic properties of CO_2 certain local irritation can occur, in particular in the nasal mucous membranes, the throat and the eyes, and the person coughs. These symptoms occur if air enriched with CO_2 is inhaled, and they are more pronounced as the CO_2 concentration rises. The data for adults with good health are shown in table 5.

3.2 CO$_2$ as a refrigerant

CO_2 can be used as a refrigerant in various refrigeration systems, both in subcritical and supercritical operation. The critical point and the triple point of each CO_2 refrigeration system must be taken into consideration. The conventional CO_2 refrigerant circuit is subcritical, that is, the circulation process (pressure and temperature) takes place below the critical point and above the triple point.

A single-stage CO_2 refrigeration system is relatively simple, but is at a disadvantage because of the limited range of temperature and the high pressure (it becomes necessary to dissipate the heat from the condenser through an additional refrigeration system).

Transcritical CO_2 refrigeration systems (in which the circulation process takes place both in the subcritical range (evaporation) and in the supercritical range (gas coo-

Gas concentration	Affect on humans
0.04 %	Concentration in the atmospheric air
2 %	Tolerable. Increase in respiration rate by 50 %
3 %	Irritating. Brief exposure is permissible for 10 min. Increase in respiration rate by 100 %
3 – 5 %	Respiratory stimulant. When inhaled for a prolonged period the first signs of poisoning appear.
5 %	Increase in respiration rate by 300 %. Headache and sweating begin after approx. 1 hour
8 %	Limit. Brief exposure is no longer permissible. Signs of suffocation. (Extinguishes open flames)
8 – 10 %	Headache after 10-15 minutes. Dizziness, burning feeling in the eyes, increase in blood pressure, increase in pulse rate, nervousness and nausea
10 – 18 %	After a few minutes cramping similar to an epileptic fit, unconsciousness and shock (e.g. rapid decrease in blood pressure). These symptoms disappear quickly if fresh air is supplied.
18 – 20 %	Symptoms similar to a stroke
30 %	Fatal

Table 5: CO$_2$: Physiological affect on humans

ling)) are at present of interest only as small and commercial refrigeration systems, e.g. car air-conditioning, small heat pumps and supermarket refrigeration systems, but not as industrial refrigeration systems. The operating pressures for subcritical circulation processes lie in the range of 5.7 to 35 bar, which corresponds to the temperature range of –55 to ±0 °C. If the evaporator has to be defrosted with hot gas, operating pressures are approx. 10 bar higher.

The most frequent use of CO_2 in industrial refrigeration systems is in cascade refrigerating systems, because here the pressure of the CO_2 can be limited to values that allow the use of normal, conventional system components such as compressors, control elements and valves.

The most frequently used refrigerant in the high-pressure stages of cascade refrigerating systems with CO_2 is ammonia. In this case, the CO_2 part of the system can be operated in various modes:
✽ CO_2 direct evaporation ("dry" or "DX")
✽ CO_2 in a forced circuit with a pump ("pump system")
✽ CO_2 as the secondary fluid ("brine system")
✽ A combination of the above modes
In the case of direct evaporation, CO_2 is

Fig. 7: Subcritical CO_2 Refrigeration Circuit

Fig. 8: Transcritical CO_2 Refrigeration Circuit

Carbon dioxide, CO_2

Regulation taken from EN 378-1

⇨ **Section 5.4.:**
   Classification of refrigerants:
   Refrigerants are classified in groups with regard to their effects on health and safety.

⇨ **Appendix E: CO_2 is a refrigerant in Group L1, Safety Group A1:**
   Refrigerants that are nonflammable, irrelevant of their concentration in air, are refrigerants with a time-weighted, averaged concentration that have no adverse effects on the majority of staff who are exposed everyday during a normal 8 hour working day and a 40 hour working week to this concentration, which is greater or equal to 400 ml/m³ (400ppm (V/V)).
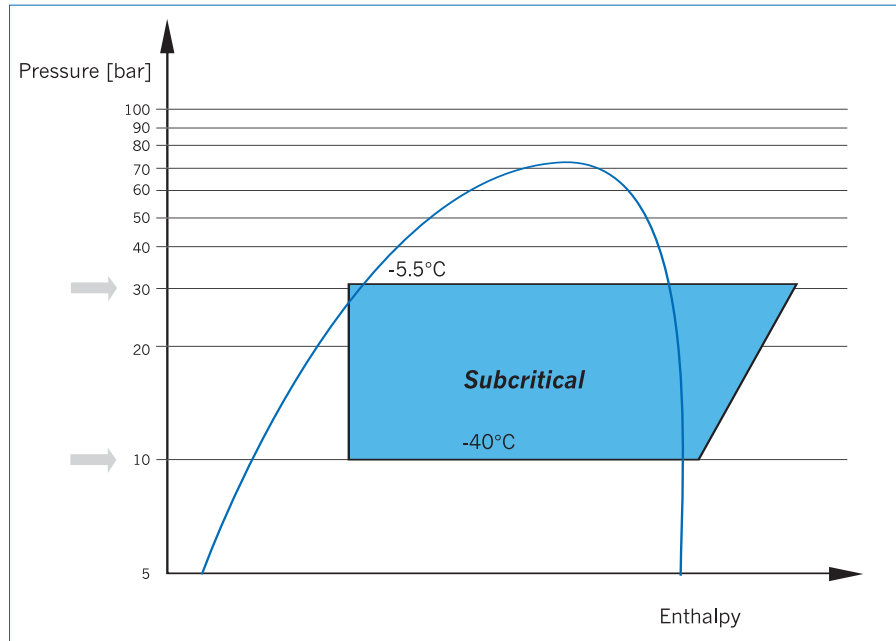
Fig. 6: CO_2; Regulation taken from EN 378-1

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used as a change-of-state refrigerant (liquid-vapor). The vaporous CO_2 is compressed in the compressor, and condensed in the CO_2-NH_3 heat exchanger (evaporator-condenser, cascade heat exchanger). This heat exchanger functions as an evaporator on the NH_3 side. Compared with a conventional NH_3 refrigeration system, the volume of NH_3 in the system can be reduced to about one tenth.
The CO_2 side of the refrigeration system works as a pump circuit when the liquid

CO_2 is pumped into the user heat exchanger (evaporator) and is partly evaporated there, and the mixture of liquid and vapor is returned to the refrigerant separator.
The remainder of the refrigerant circuit is the same as in the case of direct evaporation (figure 8).
In so-called secondary-fluid mode ("brine system", figure 9), the CO_2 that is circulated by the pump is partially evaporated in the user heat exchanger (evaporator), as in
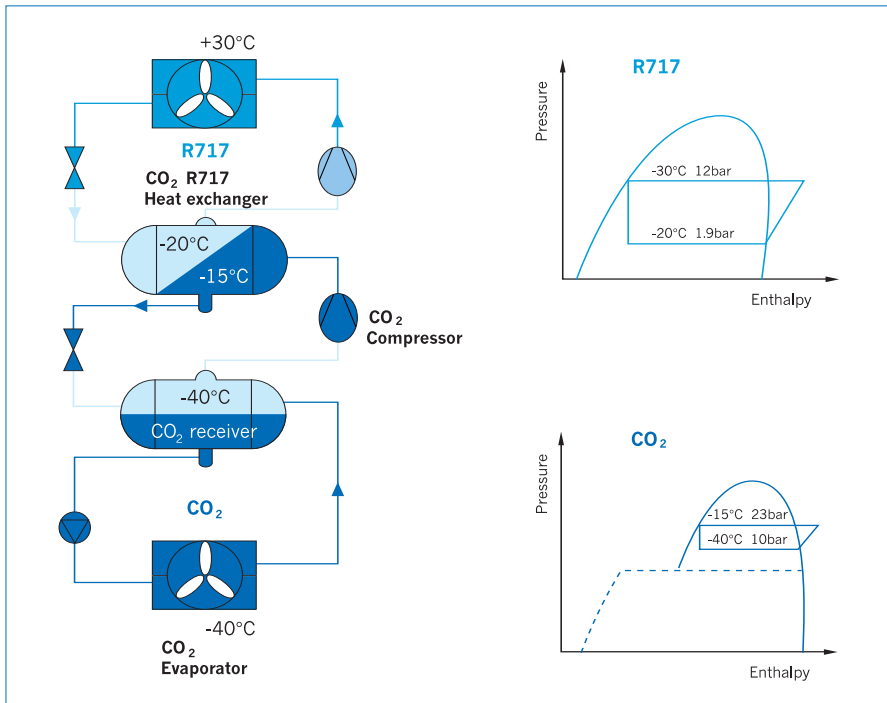
Fig. 9: Principle of CO$_2$; Refrigeration; NH$_3$-CO$_2$ Cascade Refrigerating System
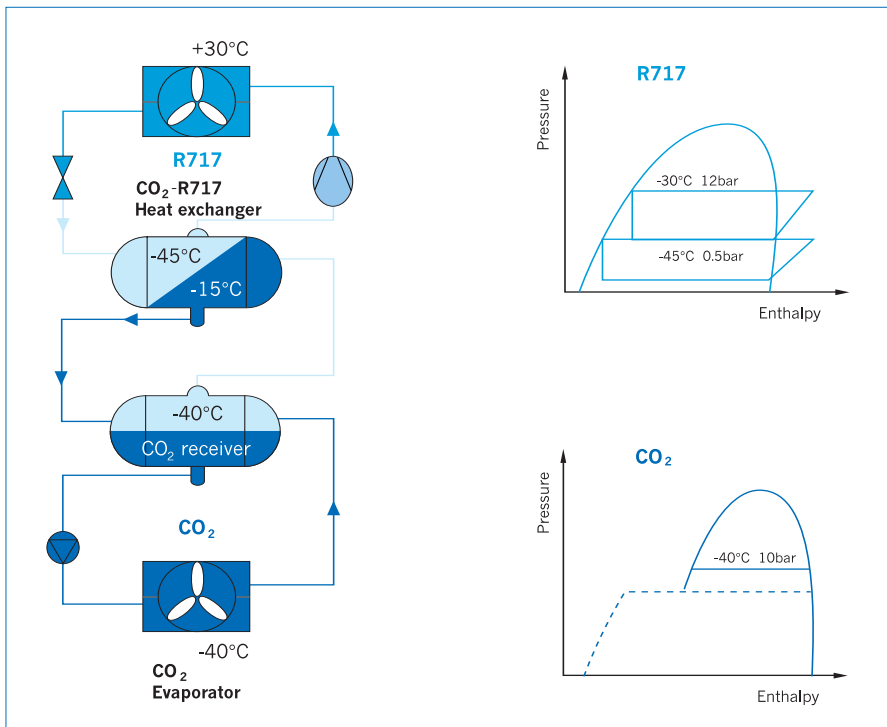

Fig. 10: Principle of CO$_2$ Refrigeration; NH$_3$-CO$_2$ Secondary-Fluid Mode ("Brine System")

pump-forced circulation mode, then led back to the refrigerant separator but not compressed by a CO$_2$ compressor, being circulated in the open refrigerant separator/cascading heat exchanger circuit.
One example of a combination of the above modes would be a combined CO$_2$ refrigeration system with pump-forced circulation and direct evaporation; for instance, a supermarket refrigeration system for two different temperature levels for consumers (evaporators, figure 11).

3.3 Operating Pressure

The following are the main aspects to take into account when defining the operating pressure of a CO$_2$ refrigeration system:

✱ Pressure during operation
The pressure during operation depends on the current version of the system (single-stage, cascade, secondary-fluid) and the load fluctuation

✱ Pressure while the refrigeration system is stopped
The pressure can be very high when the system is stopped, and must be taken into account in the following manner:
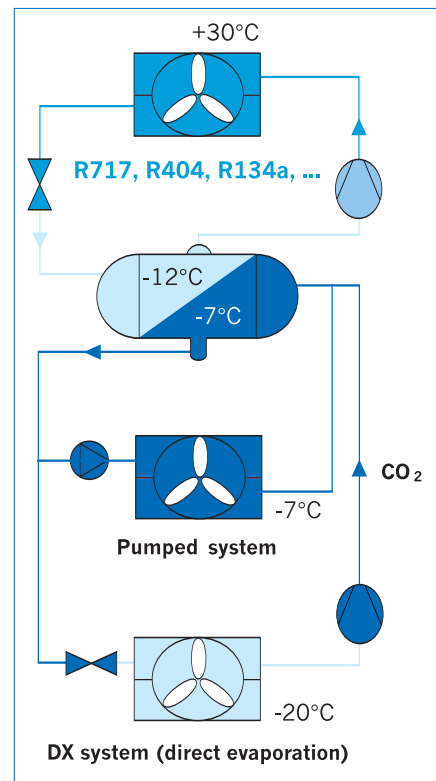If there is no control over the pressure

Fig. 11: Principle of CO$_2$ Refrigeration;
CO$_2$ Cascade Refrigerating System
for Two Temperature Levels

during stoppages when the CO$_2$ refrigeration system is switched off, the pressure in the system rises markedly as a result of heat being introduced from the surrounding air. At a temperature of ±0 °C, pressure is already at 34.9 bar, and at a temperature of +20 °C it is as high as 57.2 bar. In the area of industrial refrigeration applications, the design of a CO$_2$ refrigeration system that would withstand these pressures during stoppages (e.g. saturation pres-

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Practical limit: PS $\geq$ P$_{saturation}$ + 15 %

Design pressure

Pressure peaks
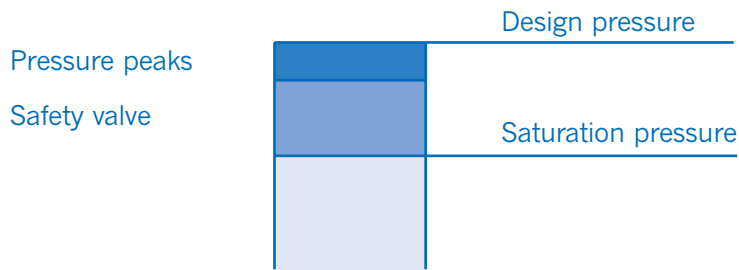
Safety valve

Saturation pressure

Fig. 12: CO$_2$; Practical calculation of the pressure tolerance for the safety valve

sure proportionate to the ambient temperature) would be far too costly.

The following measures for limiting the pressure during stoppages are feasible:

⇨ A separate, small refrigeration system keeps the temperature of the liquid at a level at which the saturation pressure is lower than the operating pressure, i.e. the maximum pressure during a stoppage is limited to an acceptable value (e.g. 30 bar).

⇨ The refrigeration system is configured with a compensating vessel of such a size that it keeps the system pressure below operating pressure.

⇨ The system is designed such (e.g. high pressure safety accumulator) that it can withstand the saturation pressure at operating temperature (see above).

✱ Pressure tolerance of the safety valve
Safety valves require a differential between opening pressure and system pressure, to ensure that the valve is always closed. In actual practice, about 10 % of the system pressure are taken for this tolerance. To compensate for load fluctuations, another 3 % to 5 % should be added (the IIR recommends a total tolerance of 20 %).

✱ Pressure during hot-gas defrosting
There are several ways to defrost a CO$_2$ refrigeration system:
   ⇨ Circulating air
   ⇨ Water
   ⇨ Electric defrosting
   ⇨ A heat-transfer medium in an additional piping system
   ⇨ Hot gas
      Hot gas defrosting is the most effec-

tive, especially at low temperatures, but it also requires the highest pressures. At an operating pressure of 50 bar, it is possible to achieve a defrosting temperature of +9 °C. The saturation pressure at +9 °C is approx. 44 bar. Add to this approx. 10 % for the safety valve and approx. 5 % for fluctuations in the system pressure, and the result is a maximum permitted operating pressure of approx. 50 bar. However, this method of defrosting can only be used in CO$_2$ refrigeration systems that are specifically dimensioned for this range of operating pressure. Any of the above-mentioned methods of defrosting can be used, depending on the concept of the system, the compressor and other system components.

3.4 Reactivity with materials
In contrast to NH$_3$, CO$_2$ is compatible with most generally known metallic materials. There are not even restrictions on its compatibility with copper and brass.
From the point of view of compatibility with plastics, a more differentiated view of CO$_2$ must be taken. In general, CO$_2$ is a very inert, stable material, and has no critical reactions with plastics. The main emphasis in these reactions is on the physical-chemical effects, e.g. porosity, swelling, formation of shrinkage cavities, and internal ruptures. These effects stem mainly from the solubility and diffusion of CO$_2$ in the respective material.
New developments and improvements in these materials have already been suc-

cessful in making versions of already well-known materials that are resistant to CO$_2$, e.g. by sealing them.

3.5 Oil in CO$_2$ refrigeration systems
Both miscible and non-miscible refrigerator oils are used in CO$_2$ refrigerator systems with conventional refrigerant compressors.
In the case of refrigerator oils that are not miscible with CO$_2$ (polyalphaolefin oils, PAO), oil management is rather complicated. The density of the PAO oils is lower than that of liquid CO$_2$. The PAO oil floats on top of the CO$_2$, making oil separation more difficult than, for example, in an NH$_3$ refrigeration system. In order to also reduce the thermal resistivity of the PAO oil in the evaporator tubes (fouling factor), there must be effective separation of the non-miscible oils in the compressor unit to keep the CO$_2$ refrigeration system as free of oil as possible.
Refrigerator oils that are miscible with CO$_2$ (polyolester, POE) allow a much simpler oil management. However, POE oils have a high affinity with water, so that the stability of the POE oil must be assured while changing the oil. When selecting compressors for use with POE oils, the marked reduction in viscosity must be taken into account. Oil-free operation, for instance when CO$_2$ is used as a secondary fluid, is the optimum from an economical point of view. This increases the heat transfer coefficient (U-value) in the evaporators. Nevertheless, this mode of operation is possible only if all valves, control equipment and other system components can work without oil.

3.6 Water in CO$_2$ refrigeration systems
It is indeed generally known that in refrigeration systems reactions with oil, air, water and solid impurities can cause problems. However, these can be controlled nowadays, e.g. by changing oil and using air separators and dryers. In comparison to NH$_3$, CO$_2$ is less susceptible, but the presence of water still causes considerable problems because of the low water solubility.
The permissible water proportion in CO$_2$ refrigeration systems is therefore signifi-

Refrigerator Oils

Oil type	PAO **Poly-alpha-olefin oil** (Synthetic Mineral oil)	POE **Polyol-ester oil** (Ester oil)
Solubility	Low (immiscible)	High (miscible)
Hydrolysis	Low	High affinity to water
Oil separation system	Special demand: High filtration demanded Multistage coalescing filters Active carbon filter	No special requirements (System requirements like HCFC/HFC)
Oil return system	Special demand: Oil drain from low temperature receiver (oil density lower than CO_2-opposite NH_3)	Simple (System requirements like HCFC/HFC)
Challenge	Oil separation and return system Long term oil accumulation in e.g. evaporators	High affinity to water Long term stability of „Clean" refrigerant system required

Table 6: CO_2; Refrigerator Oils

cantly lower than in refrigeration systems with other conventional refrigerants.

If the dew-point is not reached, and the temperature is lower than ± 0 °C, the water freezes out and there are disturbances in system components, e.g. blockage of the control valves in DX mode.

However, the water content in the CO_2 can be easily kept within the permissible limits by installing dryers. Dryers in a refrigeration system are very effective and are normally installed to avoid unnecessary pressure losses in the liquid line.

The most effective method for avoiding inadmissible water content in the CO_2 is knowing how water can get into the system and adopting corresponding countermeasures.

The pressure in the CO_2 refrigeration system is always higher than the atmospheric pressure. Therefore, risks that the water or air can penetrate through leaks do not apply, such as in a NH_3 refrigeration system. Nevertheless, it cannot be ruled out that water penetrates into the system.

How can water therefore penetrate into a CO_2 refrigeration system?
* By diffusion
* In maintenance and repair: CO_2 is handled like one of the so-called "safety refrigerants". There are therefore no

special safety regulations. If a CO_2 refrigeration system is open (e.g. during maintenance or repair), air can penetrate and the air humidity condenses in the pipes and other parts of the system. If this system is then not evacuated correctly, this inadmissible amount of water can remain there.
* Due to incomplete drying during installation and operation
* Refrigerator oil containing water is introduced into the system: during the oil change in the compressor, especially in the case of the strongly hygroscopic CO_2 soluble POE oils.
* CO_2 containing water is introduced into the system: When filling, refilling or changing the CO_2 filling, it must be considered that, on the basis of the tolerance for the permissible water proportion, there are different batches in which a relatively high water proportion can also occur.

3.7 Environmental compatibility and safety

CO_2 is an environmentally neutral substance without ozone depletion potential (ODP) and with only a low, negligible direct global warming potential (GWP = 1; basic
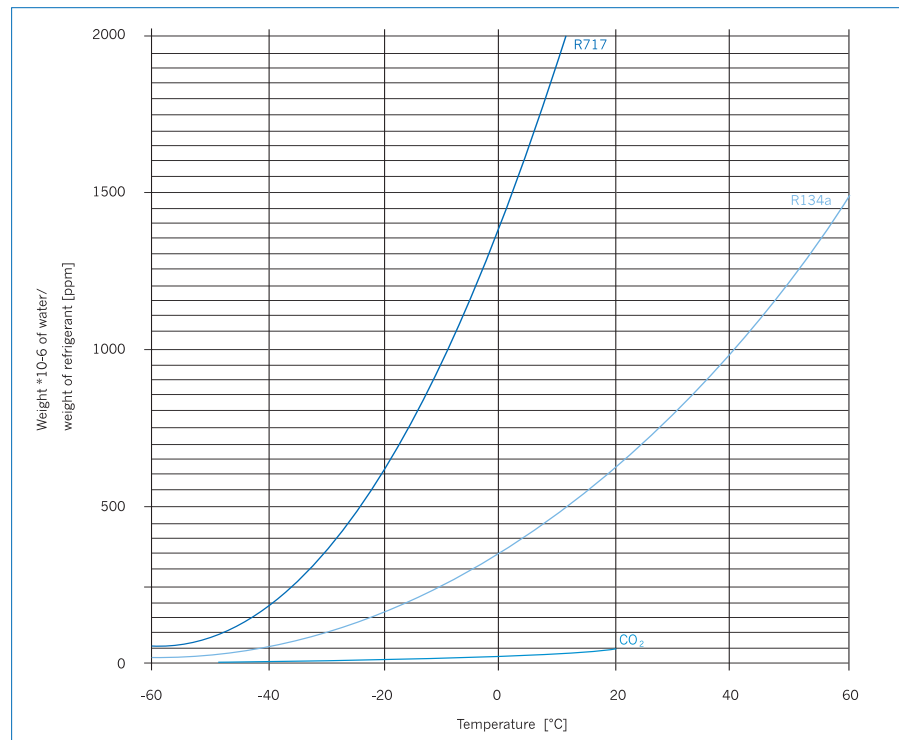
Fig. 13: CO_2; water solubility in the refrigerant vapor of different refrigerants

value).

However, CO_2 possesses compared with the HFC refrigerants a lower limit value in air (see table 5 "CO_2; Physiological affect on humans"), i.e. corresponding safety and monitoring equipment is required in closed rooms.

The global environmental burden is expressed by the TEWI value (Total Equivalent Warming Impact), which takes into account the direct global warming contribution due to CO_2 emissions caused by the refrigerant itself, and the indirect contribution caused by the energy consumption during the operation of the refrigeration system (figure 14).

The CO_2 equivalent burden expressed by TEWI is determined in this case less by the refrigerant (GWP of CO_2 = 1; basic value) than rather by the selected system principle, since the indirect share caused by the drive energy plays the decisive role (according to prevailing scientific opinion approx. 90 %).

The thermodynamic properties of CO_2 (see figure. 15 "CO_2; COP value of different refrigeration systems") facilitate favorable coefficient of performance values (COP values) and thus only low TEWI values.

Recommendations of the Federal Environmental Agency:

✱ Natural refrigerants: air, N_2, CO_2, H_2O, R717 (ammonia)
✱ Combustible refrigerants: R 290 (propane), R600 (butane)

Recommendable with limitations because of the GWP > 0:

✱ HFC refrigerants

CO_2 is an odorless, colorless substance which is classified as non-combustible and non-toxic (EN 378-1; see above), but if these properties appear to be very positive, CO_2 also has its disadvantages.

Due to the fact that CO_2 is odorless, there is no self-warning effect (such as with NH_3), e.g. if a leakage occurs in the CO_2 refrigeration system.

CO_2 is heavier than air, i.e. it collects on the ground. This can result in dangerous situations, especially in low lying rooms or rooms with limited volume. CO_2 can displace the oxygen, thus causing a dangerous condition. The density of CO_2 is around 1.5 times the density of air at ± 0 °C. This risk requires special attention in the design and in the operation of CO_2 refrigeration systems. Leak detection and/or forced ventilation are the primary requirements here.

Compared with NH_3, CO_2 is a safe refrigerant. The TLV (threshold value) is the maximum concentration of CO_2 gas in air which is permitted for an 8-hour shift in a 40-hour week. The TLV of NH_3 is 25 ppm and for CO_2 5000 ppm (0.5 %). Around 0.04 % CO_2 is present in the atmospheric air. At higher concentrations of CO_2 in the air reactions such as described in table 5: CO_2; "Physiological affect on humans" occur.

The danger due to the operating pressure (because of the higher pressure level of CO_2), which can occur when using CO_2, was already referred to in Section 3.3.

Thus minimising the filling quantity of CO_2 becomes a decisive planning and design objective to achieve an effective reduction of the danger potential.

Corresponding to the state of the art (laws, ordinances, administrative regulations, directives, standards, accident prevention regulations, technical rules etc.) CO_2 refrigeration systems are configured, designed, built, installed and operated in such a way that they withstand the mechanical, thermal and chemical loads that can occur in operation and standstill of the system and remain leak-proof (primary protective measures).

Therefore the probability that larger amounts of CO_2 escape from CO_2 refrigeration systems is very low.

Measures for limiting the effects of CO_2 take effect if primary protective measures fail and an unexpected release of larger quantities of CO_2 occurs (secondary protective measures).

Fresh air and oxygen devices come into question as means of protection on escape of CO_2 gas. Symptoms occurring due to CO_2 soon disappear on treatment with fresh air.

Design, organisational and personal protective measures, construction and type of the refrigeration system and of the installation site (e.g. escape openings for natural and forced ventilation for CO_2 gas, safety and monitoring devices) contribute decisively to limiting damage.

Laws, ordinances, regulations, directives, standards and other specified rules of engineering which define the protective goals and state technical, organisational and personal protective measures to prevent hazards and to limit damage in its effect have existed for decades for the protection of persons, the environment and assets.

As representatives of these standards the following are named:

⇨ European Norm EN 378:2000; refrigeration systems and heat pumps; safety engineering and environmentally relevant requirements
⇨ EU Directive 97/23/EG for pressure vessels (pressure vessel directive)

The following measures for preventing damage and accidents are regulated in them:

⇨ Basic requirements, definitions and definitions of terms, classifications and selection criteria
⇨ Design, production, construction

TEWI value

TEWI = Total Equivalent Warming Impact

TEWI = Leakage + recovery losses + energy requirement $\,\hat{=}\,$ CO_2 emission

Leakage + recovery losses $\quad\hat{=}\quad$ Direct burden due to
È   System leaks
È   System disposal

Energy requirement $\qquad\hat{=}\qquad$ Indirect burden due to
È   Energy consumption of the refrigeration system

Fig. 14: CO_2; TEWI value

and equipment, testing, labelling and documentation

⇨ Installation and protection of persons and objects

⇨ Operation, maintenance, repair and recovery

Here it is assumed that the CO_2 refrigeration systems satisfy all other safety engineering requirements according to the state of the art which have as contents the general hazards which can be caused by technical installations (e.g. moving parts, excessive pressure, electrical power, material failure, release of energy etc.).

3.8 Economy

One of the most important reasons why it is possible to advantageously use CO_2 in many areas of commercial and industrial refrigeration is its economy. This is due to the fact that CO_2 can be used very economically if we optimise the thermodynamic and ecological advantages of CO_2 (a very high volumetric refrigerating capacity, low procurement prices, no reclamation and disposal) with the additional costs of a CO_2 refrigeration system (greater apparatus costs, dryness and oil problems).

In CO_2 cascade refrigeration systems it is necessary to use a cascade heat exchanger. Using this heat exchanger results in less economy in the refrigeration system arising from higher costs for this heat exchanger and the need for a temperature difference between the two refrigerants. It is important that this system does not leak: If even minor leaks occur, NH_3 and CO_2 combine to form the notorious gingerbread spice ammonium carbonate and destroy the heat exchanger. The compressor and the other components of a CO_2 refrigeration system must be specially designed because of the high vapor density and pressure level (especially on the suction side). Also, only high-grade dried CO_2 may be used.

Despite this, compressor operation with CO_2 is more efficient (COP value) and the heat exchange is larger.

The total economy of a CO_2-NH_3 cascade refrigeration system is therefore almost comparable with a standard NH_3 refrigeration system.

A very advantageous use from an energy point of view, using CO_2 as a secondary
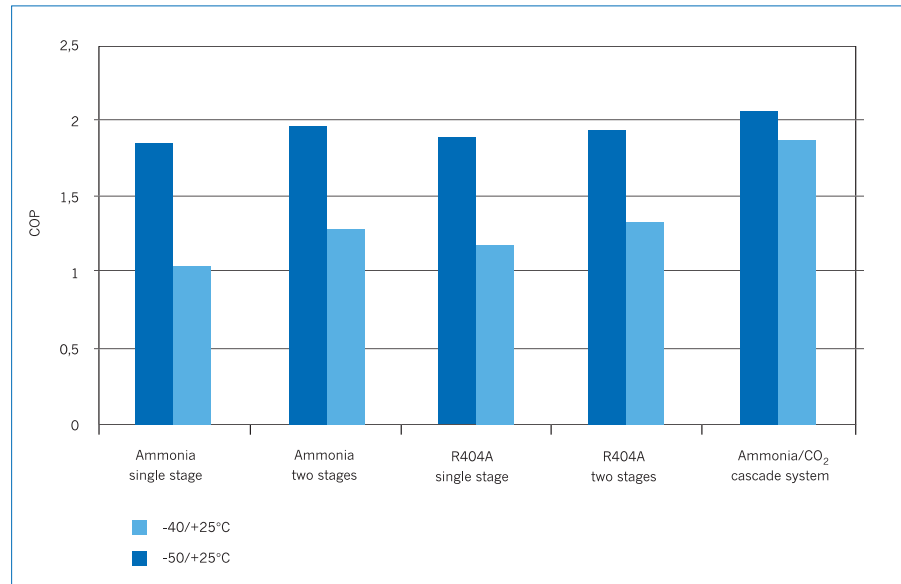

Fig. 15: CO_2; COP value of different refrigeration systems

fluid in a cascade refrigeration system (figure 10), is appropriate for industrial and larger commercial refrigeration systems. The operating method there is not critical, thereby ensuring good economy. In the application range favorable for this (t_0 = -50 to -10 °C), the pressures are still at a level for which already available components can be adapted at a reasonable cost.
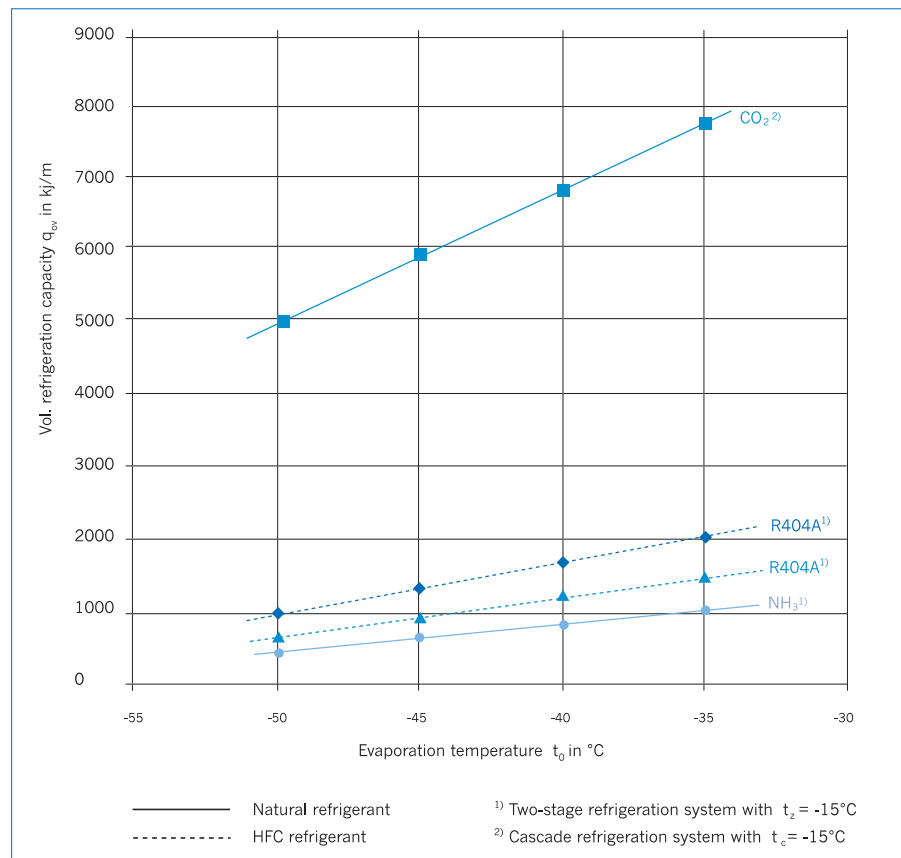

Fig. 16: CO_2; volumetric refrigerating capacity of different refrigerants

CO₂; comparison of refrigerants									
Refri-gerant	Appli-cation[1]	Evap. temp. at 1.013 bar	Evap. pressure at 0 °C	Condens. pressure at +45 °C	Condens. pressure at +55 °C	Density at +45 °C liquid	Latent heat at boiling point	Volumetric refriger. capacity ±0/±40 °C	Refrigerant group/ safety group
HFC or HCFC			°C	bar	bar	bar	kg/m³	kJ/kg	kJ/m³
R22	T, N	-40.8	5.0	17.3	21.8	1,106	234	3,280	L1 / A1
R134a	N, K	-26.1	2.93	11.6	14.9	1,125	217	2,050	L1 / A1
R407C	N, K	-43.6	4.5	19.5	25.6	1,047	246	2,850	L1 / A1
R404A	T, N	-46.7	6.1	20.6	25.8	931	202	3,130	L1 / A1
R410A	ET, T, N	-51.6	8.0	27.3	34.3	946	264	4,710	L1 / A1
R507	T, N	-47	6.3	21.09	26.44	858	200	3,300	L1 / A1
R23 [2]	ET, T	-82	25.0	-	-	-	239	3,160 [3]	L1 / A1
Natural refriger-ant			°C	bar	bar	bar	kg/m³	kJ/kg	kJ/m³
R290 (Propane)	T, N, K	-42	4.7	15.3	19.1	458	423	2,770	L3 / A3
R1270 (Propylene)	T, N, K	-47.9	5.86	18.46	22.9	466	439	3,370	L3 / A3
R600a (Isobutane)	K	-12	1.56	6	7.7	524	367	~ 1,100	L3 / A3
R717 (NH₃)	T, N, K	-33	4.29	17.83	23.11	571	1,368	3,700/472 [3]	L2 / B2
R723 (NH₃/DME)	T, N, K	-36.5	4.7	18.72	24.1	579	991	3,830	L2 / B2
R744 (CO₂) [2]	**T, N, K**	**(-78.4) /**	**79.4**	**- -57 [4]**	**-**	**1,029 [5]**	**~ 335**	**4,650 [3]**	**L1 / A1**
R718 (H₂O)	Aqua Turbo	+100	0,01 [6]			1,000 [7]	2,255	11	L1 / A1

[1]	N:	Normal cooling	(- 20 … +10 °C)		[3]	at –50 / -10 °C		
	T:	Intense refrigeration	(- 50 … +10 °C)		[4]	Triple point at 5.2 bar, -57 °C		
	ET:	Extreme refrigeration	(-100/-60 … °C)		[5]	at –20 °C		
	K:	Climate unit	(-10 … +10 °C)		[6]	at +6.7 °C		
[2]		Operation only with cascade refrigeration system			[7]	at ±0 °C		

Table 7.: CO₂; Comparison of the refrigerants

system shown in figure 9, the principle structure of the secondary circuit with CO_2 largely corresponds to a low-pressure pump circuit of the type often used in NH_3 refrigeration systems, except that the secondary fluid CO_2 is liquefied in a cascade heat exchanger (evaporator-condenser).

The ratio of refrigerating capacity to drive power (COP value, example in figure 15) of a CO_2 system is very favorable and pushes the primary energy requirement to an extremely low level.

The extremely high volumetric refrigerating capacity of CO_2 (latent heat through phase change) leads here to a very low volumetric flow and allows small compressor sizes and therefore small pipe and valve cross sections. Compared with NH_3 and the HFC refrigerants, CO_2 is different in many respects. This fact is emphasized below. Under constant operating conditions, e.g. evaporating temperature, condensing temperature, we obtain a real comparison.
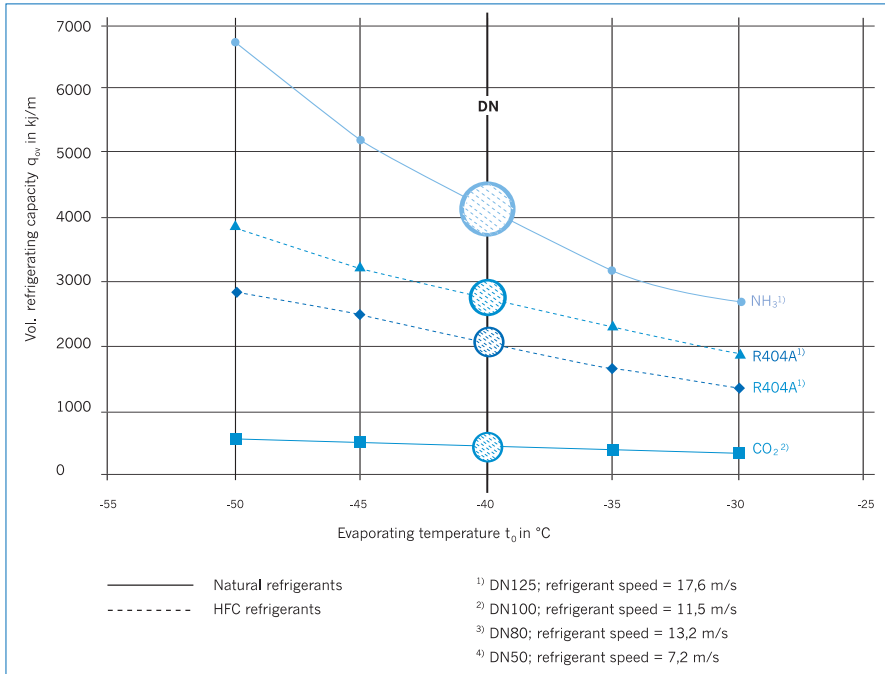
Figure 17

Vol. refrigerating capacity q_{ov} in kj/m (y-axis): 0, 1000, 2000, 3000, 4000, 5000, 6000, 7000

Evaporating temperature t_0 in °C (x-axis): -55, -50, -45, -40, -35, -30, -25

DN

Labels: NH$_3$[1], R404A[1], R404A[1], CO$_2$[2]

Natural refrigerants
- - - - - - HFC refrigerants

[1] DN125; refrigerant speed = 17,6 m/s
[2] DN100; refrigerant speed = 11,5 m/s
[3] DN80; refrigerant speed = 13,2 m/s
[4] DN50; refrigerant speed = 7,2 m/s

Fig. 17: CO_2; volumetric flows and nominal width ranges (return line at t_0 = -40 °C) of various refrigerants in pump operation for $\dot{Q}_0$ = 100 kW
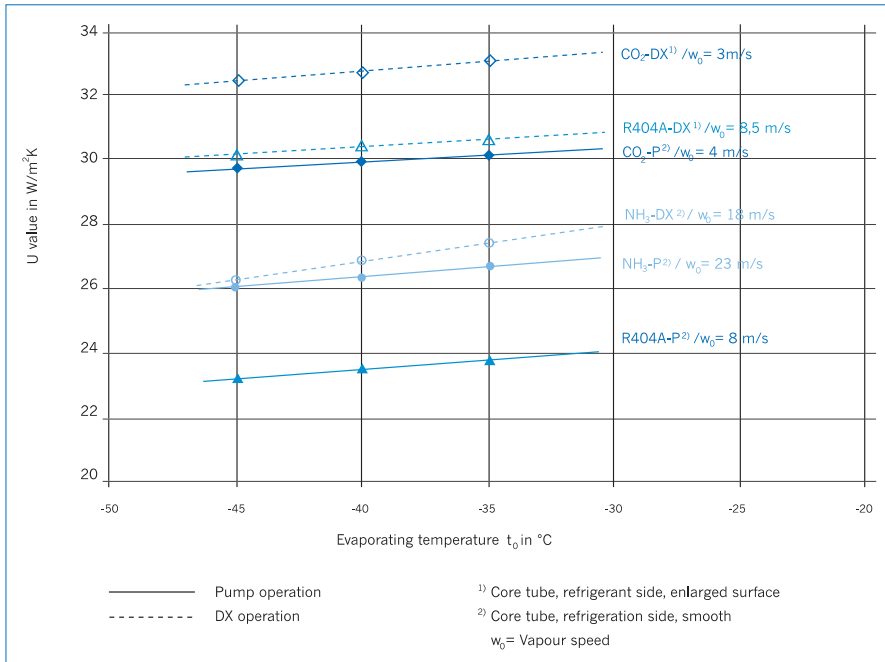
Figure 18

U value in W/m²K (y-axis): 20, 22, 24, 26, 28, 30, 32, 34

Evaporating temperature t_0 in °C (x-axis): -50, -45, -40, -35, -30, -25, -20

Labels:
CO_2-DX[1] /w_0 = 3 m/s
R404A-DX[1] /w_0 = 3,5 m/s
CO_2-P[2] /w_0 = 4 m/s
NH$_3$-DX[2] / w_0 = 18 m/s
NH$_3$-P[2] / w_0 = 23 m/s
R404A-P[2] /w_0 = 8 m/s

Pump operation
- - - - - - DX operation

[1] Core tube, refrigerant side, enlarged surface
[2] Core tube, refrigeration side, smooth
w_0 = Vapour speed

Fig. 18: CO_2; heat transfer coefficient (U value) of various refrigerants

Suction lines for direct expansion (DX operation)

The comparison of suction lines in DX operation shows almost the same results as the comparison in pump operation, for both pressure loss and for the pipeline dimensions.

Liquid lines

In both pump and DX operation, the liquid line cross sections designed for CO_2 will be greater than for NH$_3$, but only slightly greater than for HFC. This is explained by the greater evaporation heat of NH$_3$ compared with CO_2 and HFC.

Due to the relatively low vapor volume and the high volumetric refrigerating capacity of CO_2, the CO_2 refrigeration system is relatively sensitive to capacity fluctuations. It is therefore very important to configure the liquid separator with a sufficient volume to accommodate the vapor volume from the pipelines.

Refrigerant compressor

With the same compressor refrigerating capacity and the same operating conditions, in a CO_2 system we require a smaller refrigerant compressor than in a NH$_3$ or HFC refrigeration system. For a compressor refrigerating capacity of Q_{0V} = 250 kW whereby t_0 / t_C = -40 / -15 °C, the compressor for CO_2 is approximately 8 times smaller than for NH$_3$, and 6 times smaller than for R 404A.

Heat transfer coefficient (U value)

Under the same operating conditions (core tube diameter, tube pattern, fin spacing, face velocity, superheating temperature, condensing temperature) we obtain in a CO_2 refrigeration system higher U values than in a NH$_3$ or HFC refrigeration system, in both DX and pump operation (figure 18).

Discussion of the results:
Return lines in pump operation

The direct comparison in pump operation (figure 17) shows that much smaller tube cross sections are required for the return lines in CO_2 refrigeration systems than is the case with NH$_3$ and HFC refrigerants.

The permissible pressure in CO_2 return lines is approx. 10 times higher than for NH$_3$ and HFC return lines at the same temperature. This is due to the relatively high vapor density of the CO_2. Depending on an optimized pump rate for each refrigerant, the dimensioning results differ only insignificantly.

Boundary Conditions

Condition	des. / dim.	DX operation	Pump operation
Core tube	d in mm	15	15
Tube pattern	s_R in mm	50 x 50	50 x 50
Fin pattern	s_L in mm	10	10
Face velocity	w_1 in m/s	2.1	2.1
Superheating	Δt_{0h} in K	8,0	-
Condensing temperature	t_c in °C	-15	-15

To figure 18: boundary conditions

Fig. 19.a): CO_2; applications; industrial refrigeration

3.9 Applications

From an ecological point of view, CO_2 can be used with a positive overall balance (favorable TEWI value) where a high potential leakage rate can be expected (e.g. car air-conditioning systems) and where flammable refrigerants cannot be accepted. Other potential areas of use can be hot water treatment and service water heating, drying processes in which there is a high temperature gradient on the high-pressure side and industrial and commercial applications as already described in Section 3.8 "Economy".

As with all refrigerants, CO_2 does not fulfil all the requirements for all-round use.

Fig. 19.b): CO_2; applications; heat pump for hot water treatment

Fig. 19.c): CO_2; applications; car air-conditioning systems

Changed system technologies and specially tailored components are required for this (also see Section 3.8 "Economy"). Special demands can be placed on the materials with regard to the technical regulations and standards, particularly for operation in the overcritical and transcritical ranges. Most normal refrigeration machine oils cannot be mixed with CO_2 and require complex measures for oil return, whereas with the mixable and easily soluble POE oils we can expect higher water contents and heavy viscosity losses (see Section 3.5 "Oil in CO_2 refrigeration systems").

4. Summary

CO_2 possesses favorable properties, particularly for use in refrigeration systems with low temperatures, but it is not a complete substitute for NH_3. Most industrial CO_2 refrigeration systems are cascade refrigeration systems with NH_3 on the high-temperature side.

In many respects, CO_2 is a very uncomplicated refrigerant but it is essential to remember that CO_2 possesses a few of its own special characteristics that differ significantly from normal refrigerants. These differences must be known and allowance must be made for them in design, order processing, installation, operation and servicing of the CO_2 refrigeration system so that problems can be avoided or minimized.

Components for industrial CO_2 refrigeration systems with pressures up to 40 bar are available in sufficient dimensions. Almost all manufacturers of refrigeration systems for normal refrigerants also supply components for CO_2 refrigeration systems.

The availability of components for industrial CO_2 refrigeration systems with high pressures is currently still limited, but the availability of precisely these critical components is an important factor for the growth rate of the use of the refrigerant CO_2.

To summarize, the following can be said about the refrigerant CO_2:

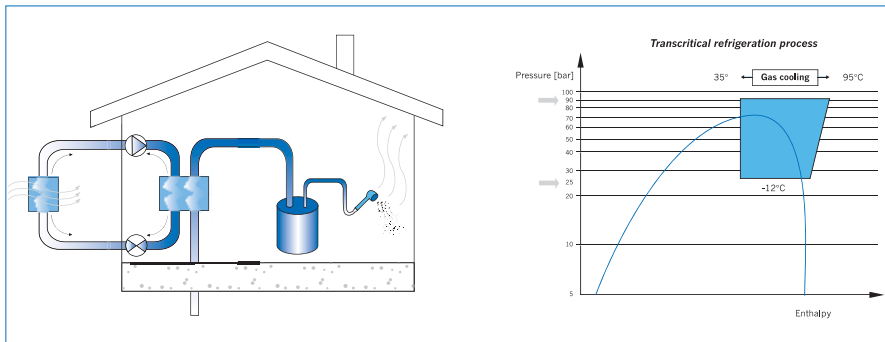✽ CO_2 is a natural, non-toxic and non-flammable substance

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* CO_2 is a chemically inactive refrigerant
* "All" reactions with CO_2 require the presence of water
* The permissible water content in CO_2 refrigeration systems is much lower than in other refrigeration systems
* Maintaining the permissible water content in CO_2 refrigeration systems is an effective method for preventing reactions with CO_2
* Water, air, oxides, oil, impurities and metallic system components are the most important causes of chemical reactions - also in CO_2 refrigeration systems
* Particular attention must be paid to active agent and system component selection due to the high vapor density and pressure level
* Optimum utilisation of the thermodynamic, safety-related and ecological advantages of CO_2 lead to alternative refrigeration system solutions despite additional complexity on the system side

5. References

* Danfoss; CO_2 refrigerant for industrial refrigeration (article)
* Danfoss; Components for CO_2 applications in Industrial Refrigeration (Application guide)
* Danfoss; CO_2-presentation; PART I and PART II
* Bitzer; Kältemittelreport, 12th Edition; A-500-12
* Air Liquid; Eigenschaften der Kohlensäure; CO_2 Kohlendioxid
* R. Plank; Handbuch der Kältetechnik, Bd. IV; „Kältemittel"; Part 2 Section D „Kohlendioxid"
* Güntner-WT-Schulung; Section 2 Kältemittel (intern)
* Güntner-WT-Schulung; Section 3 Ammoniak (intern)

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Design criteria for CO_2 evaporators

Following the replacement of CFC by chlorine-free refrigerants because of their damaging effect on the ozone layer, concerns began to be raised at an early stage about the high risks these new refrigerants posed of exacerbating the greenhouse effect. In recent years attention has therefore increasingly been focused on natural refrigerants. The past year has witnessed a significant increase in the use of CO_2 particularly and since the DKV* conference 2001 in Ulm, if not earlier, it has been recognised that CO_2 has applications, principally in NH_3 cascade operations, that are economically viable now. Plants of this type generally operate at evaporating temperatures of between –40 and –50 °C.

In designing CO_2 evaporators there are some special features which must be borne in mind and these will be examined in greater detail in the following pages.

Classification of CO_2 evaporators

Essentially the main difference between different types of CO_2 evaporators lies in their mode of operation, i.e. direct expansion evaporators, pump-operated evaporators and evaporators for generating process gas. Güntner has been manufacturing the latter for over 20 years and from a thermodynamic viewpoint they may be classified between the two variants pump operation and direct expansion. However, since such evaporators play no part in refrigeration, it is not intended to discuss them any further here.

A further distinguishing feature is the required operating pressure. Provided it has been confirmed at the planning stage that the operating pressure of 32 bar (the current nominal pressure for Güntner evaporators) will not be exceeded, any standard materials can in principle be used. This is of particular importance where the direct expansion system is being used because in that case internally grooved pipes can be used. This variant is the most economical in all cases.
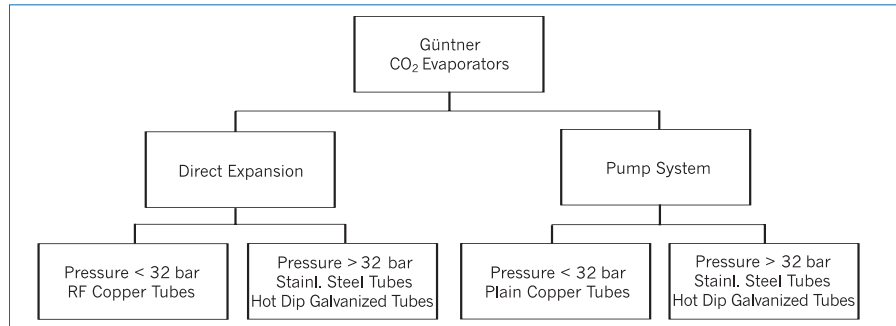

Fig. 1: Overview of CO_2 evasporators

Where hot gas defrosting using CO_2 is being carried out, the permitted pressure is generally between 45 to 50 bar. This is difficult to achieve with copper pipes. Thicker heat exchanger pipes would have to be used in all cases and in any event these are not available with internal grooves. In addition, the joining pipes (header outlets, connecting pipes) would also need thicker walls and would need to be purpose-built. Heat exchangers made of hot dip galvanised or stainless steel where pressures of 50 bar are relatively easy to achieve would be suitable here.

Design of CO_2 evaporators

Design using the Güntner Product Calculator

The CO_2 evaporator designs are carried out in the Güntner design program (Güntner Product Calculator) in the same way as for all other evaporators, i.e., in accordance with strict thermodynamic rules: Firstly, the material data are determined which are stored in the form of temperature-dependent functions. However, any amount of material data can also be entered manually. Mass flow rates, heat transfers, pressure losses, dehumidification and finally refrigerating capacity are then determined using an iterative process.

The stored calculation methods have been progressively refined over the years and adjusted where necessary to take account not only of new information but also of new tube and fin structures. The program is now so accurate that only the calculated data need to be submitted to Eurovent for the certification of a new range of unit types. Up to now all certifications have been granted immediately, i.e. no measured output was more than 8 % below the calculated values.

An additional indication of the program's reliability is the fact that there has hardly

Refrigerant:	CO_2	R22	NH_3
Vapor Pressure [bar]	10	1	0.7
Enthalpy of Evap. [kJ/kg]	322	243	1387
Density of Gas	26.24	4.85	0.64
Gas Volume Flow for 10 kW [m³/h]	6	41	47
dp/dT [bar/K]	0.37	0.05	0.04
Required Distributions (ca. 8m/s)	2	12	12

Table 1: Material properties of CO_2, R22 and NH_3 at - 40 °C

* [Deutsche Kälte- und Klimatechnik Verein: German Association of Refrigeration and Air Conditioning Technology]

No. of Passes	Distributions	Tube Length
4	18	8 m
6	12	12 m
8	9	16 m
12	6	24 m
18	4	32 m
24	3	48 m
36	2	72 m

Table 2: Circuiting variants GHN 051C/210

ever been a complaint where too little refrigerating capacity had been provided without an obvious cause. This also applies to the CO_2 evaporators which have been sold in large numbers. When CO_2 is being used, an additional factor comes into play: The internal heat transfer is so high with CO_2 that its effect on the evaporating capacity is practically negligible. For example, a heat transfer increase (ai) at the refrigerant side from 2000 W/m²K to 3000 W/m²K in a typical industrial air cooler increases the absolute heat transfer value (U) by only about 6 %. The effect on refrigerating capacity is even less since a greater heat transfer value leads to increased air cooling and thus to a lower driving temperature difference. Some of the internal heat transfers occurring when CO_2 is being used are still considerably greater than 3000 W/m²K so that any uncertainty about the calculations in this respect may be ignored.

Nevertheless, in April 2004 a Güntner CO_2 evaporator was gauged at the FKW laboratory [Research Centre for Refrigeration Technology and Heat Pumps] at Hanover. This confirmed that the output calculated by the GPC was accurate to within 1 %. It was also possible to show that large distribution lengths could be controlled by automatic control techniques.

Special Properties of CO_2 / comparison with R22 and NH_3

As is widely known, CO_2's special property as a refrigerant is its high pressure level. On the one hand this property may engender considerable reluctance to use CO_2 as

a refrigerant because it makes CO_2 very difficult to handle, especially at higher temperatures. At low temperatures, however, this property is a decisive advantage. For example, a comparison of the properties of CO_2 with those of R22 and NH_3 at a temperature of -40 °C (table 1) shows that the volumetric refrigerating capacity of CO_2 is about 7 times greater than that of R22 or 8 times greater than that of NH_3. To put this another way the pipe cross-sections required in a CO_2 plant are only about an eighth of those in a comparable NH_3 plant. This leads to smaller pipe cross-sections, smaller compressors and, of course, to less filling quantity as well and ultimately to more economical plants. However, the high pressure and the associated high gas density also lead to the above-mentioned high temperature heat transfer during evaporation.

A further consequence of the high pressure level is the weak temperature/pressure dependency. For example, a pressure change of about 0.37 bar is required to change the steam pressure by 1K at -40 °C. When using R22 that could be achieved with a pressure change of as little as 0.05 bar and with just 0.04 bar using NH_3. Because of this beneficial side effect, pressure losses have only a negligible effect on the evaporating temperature.

However, this property is ultimately the

only thing that makes it possible to use CO_2 with normal evaporator geometries. In order to explain this more clearly it is necessary to illustrate the relationships by means of a specific example. For example, an evaporator with 6 rows of pipes in the air flow direction, 12 rows of vertical pipes and a pipe diameter of 15 mm would give the following theoretically possible circuiting variants (pass numbers):

Assuming a fixed refrigerating capacity of 15 kW, a refrigerant speed of over 7 m/s can be achieved with R22 at -40 °C with a 6-pass circuiting (12 distributions). However, it is only possible to achieve a speed barely exceeding 6 m/s with a 36-pass circuiting (2 distributions) when using CO_2.

This means that when CO_2 is being used the distribution length must be 6 times longer. Assuming on an initial estimate that the pressure loss per metre of pipe is the same for all refrigerants at the same speed, there would be 6 times more pressure loss with CO_2. In fact the actual pressure loss using CO_2 is even a little greater because of the properties of the materials (figure 2).

However, from a thermodynamic viewpoint, it is only the temperature drop caused by the pressure loss that is important since this lowers the driving temperature difference at the evaporator and, as a re-
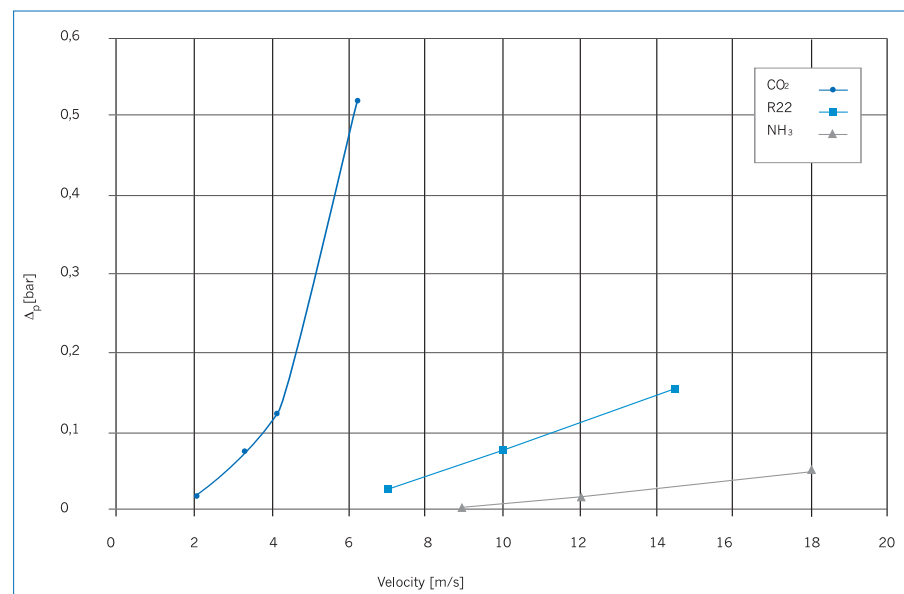

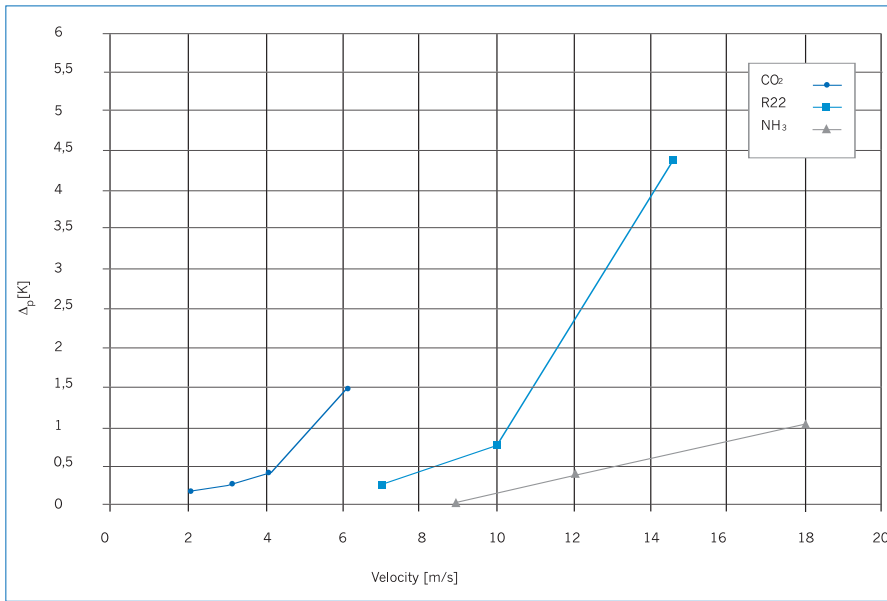*Fig. 2: Pressure loss in relation to refrigerant speed*

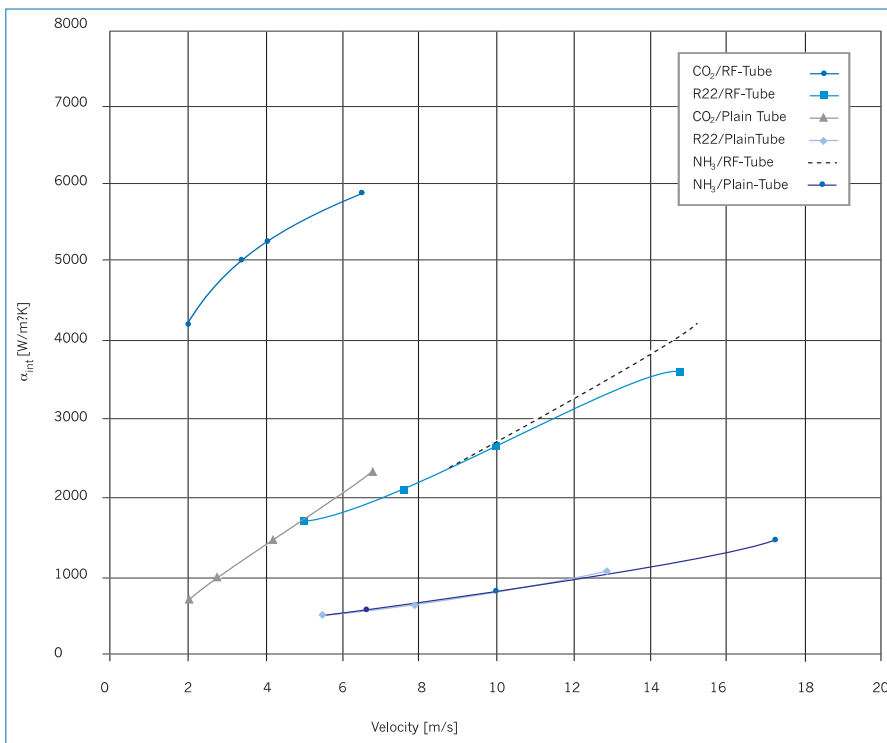Fig. 3: "Temperature drop" in relation to refrigerant speed

Fig. 4: Heat transfer of different refrigerants in relation to velocity

sult, the refrigerating capacity. This is however only slightly greater than with R22 and NH_3 because of the weak temperature pressure/dependency (fig. 3). If this were not the case, other heat exchanger geometries would have to be used for CO_2, i.e. pipes with a considerably smaller diameter would have to be used without increasing the number of pipes.

Design/results of calculations
By way of an example let us assume that a comparison of the three refrigerants, CO_2, R22 and NH_3 was to be carried out using the evaporator described above. The conditions chosen would be T_0 = -40 °C (evaporating end temperature), T_1 = –30 °C (air inlet), RH = 95 % (relative

humidity of air). In order to exclude other influences the same quantity of air was used in all cases.

Similarly, the calculations were based on the use of copper pipe throughout, even though this was purely a theoretical assumption for NH_3. The capacity was set at a constant value. Circuitings appropriate to the respective refrigerant were used. Figure 4 shows the internal heat transfer coefficients in relation to the refrigerant speed using direct expansion. As can be seen, values can be achieved with CO_2 with plain pipe that can only be achieved with internally grooved pipe when using R22. In this comparison it should be noted that the values for NH_3 are purely theoretical since, as previously mentioned, the calculations were based on the one hand on the use of copper pipe and on the other hand on superheating of 5K which is difficult to achieve with NH_3.

However, the fact that the temperature drop caused by pressure loss is more important than the refrigerant speed makes figure 5 more meaningful, showing as it does the internal heat transfers in relation to the "temperature drop" caused by pressure loss.

However, this diagram does not provide much help in creating an optimum design for an evaporator either. What ultimately matters is the product of the total heat transfer value (U) and the average logarithmic temperature difference (ΔT) which is often also expressed as the thermal load. Figure 6 shows this again for the three refrigerants, CO_2, R22 and NH_3 for evaporators with plain tubes and internally grooved tubes. This clearly shows that the optimum speed for CO_2 is markedly less than the speed achievable with R22 and substantially less than the speed achievable with NH_3. This is due on the one hand to the greater pressure loss of CO_2 because of the circuitry and on the other hand to the greater heat transfer. Both result in a situation where, when using CO_2 even at relatively low speeds and on increasing the speed, the negative effect of the pressure loss has a greater effect on capacity than the positive effect caused by the increasing heat transfer. Exactly the opposite situation applies when using NH_3 where good heat transfers can be achieved only
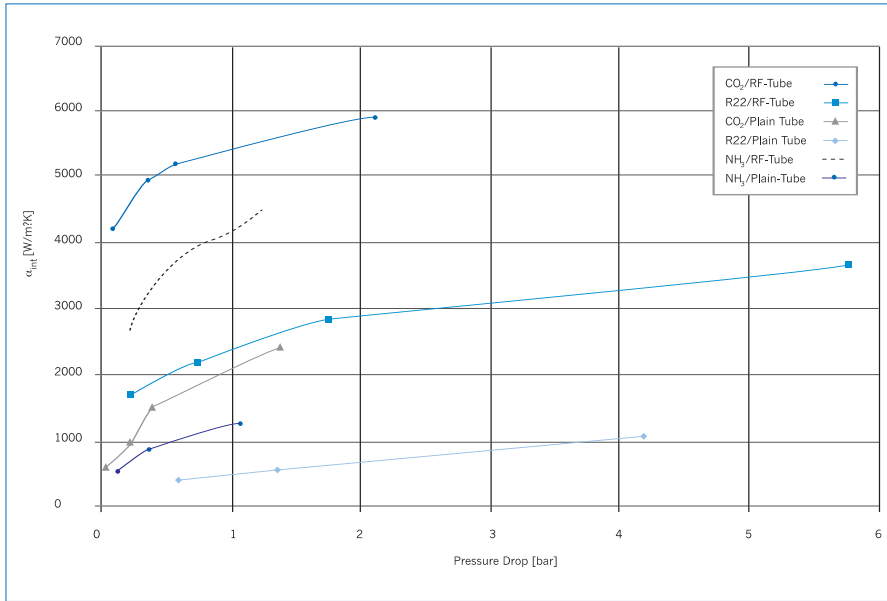
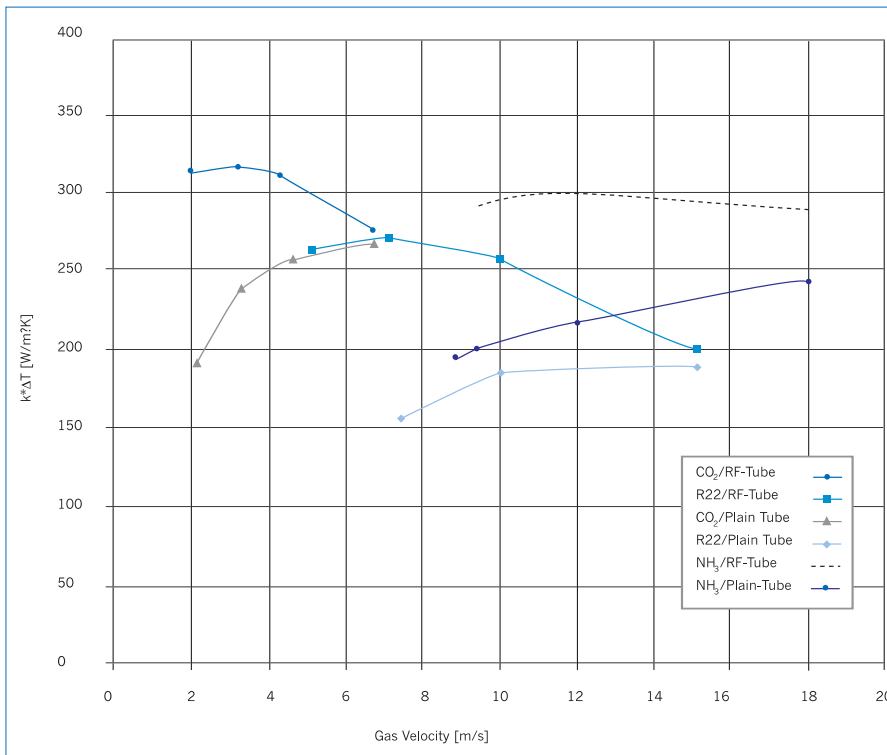Fig. 5: Heat exchange of different refrigerants in relation to pressure loss

Fig. 6: Thermal load in relation to refrigerant speed

re be even greater in a real heat exchanger. Figure 8 again shows the thermal load of the different variants. The values for direct expansion are shown again for comparison purposes. It can be seen that CO_2 excels in pump operation. Again this is because, when CO_2 is being used, the driving temperature difference plays a more important role than the heat transfer which is in any event high. Because of the absence of superheating the average logarithmic temperature difference using pump operation is generally greater than when using direct expansion. The total thermal load during pump operation is therefore greater than when using direct expansion although there is somewhat less heat transfer. When using NH_3 the same effects again result in the opposite outcome: the greater heat transfer with direct expansion outweighs the disadvantage caused by the smaller temperature difference so that direct expansion has an advantage, at least in this theoretical case.

Summary

By using CO_2 greater heat transfer values can be achieved both in pump operation and in direct expansion than with all other current refrigerants. Because of the high pressure level and the resulting weak pressure/temperature dependency, all standard heat exchangers can be used just by altering the circuiting to an increased number of passes or smaller distributions. Should operating pressures of 32 bar be insufficient, either thicker walls or other materials (steel, stainless steel) must be used. In that case the cost advantage achieved by the greater heat transfers is again offset or even reversed. It would therefore be sensible in the long term to introduce a specific fin geometry which has a smaller pipe diameter with the same pipe spacing. In this way the long distribution lengths could on the one hand be dispensed with and on the other hand there would be the advantage that pipes of smaller diameters have greater stability under pressure and that the required wall thicknesses would thus be kept within reasonable bounds. However, it is only worth investing in such a tool if greater quantities of CO_2 evaporators are required.

at very high speeds. However, to make up for that, the pressure losses at these speeds remain very low because of the circuitry. At this point it must be stressed again that the value for NH_3 shown in figure 6 is practically unachievable.

The advantages of using CO_2 in pump-operated evaporators are even more evident. Figure 7 shows a comparison of the internal heat transfers at a pumping rate of 2.5. The internally grooved tube has been dispensed with here since it produces no significant advantages when pump operation is being used.

In this case calculations for NH_3 were also based on an assumption that copper pipe was used. The difference between the other refrigerants and CO_2 would therefo-
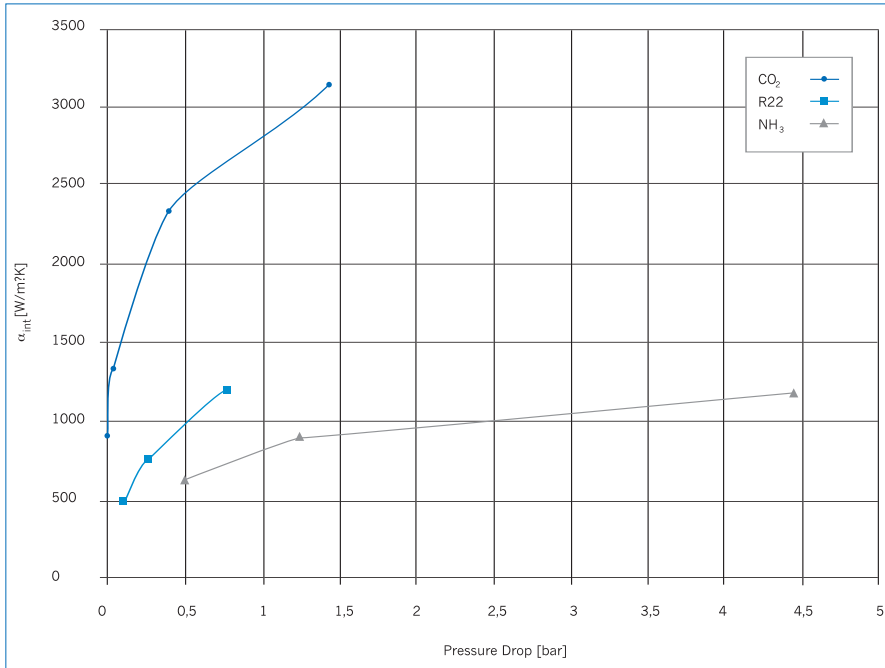
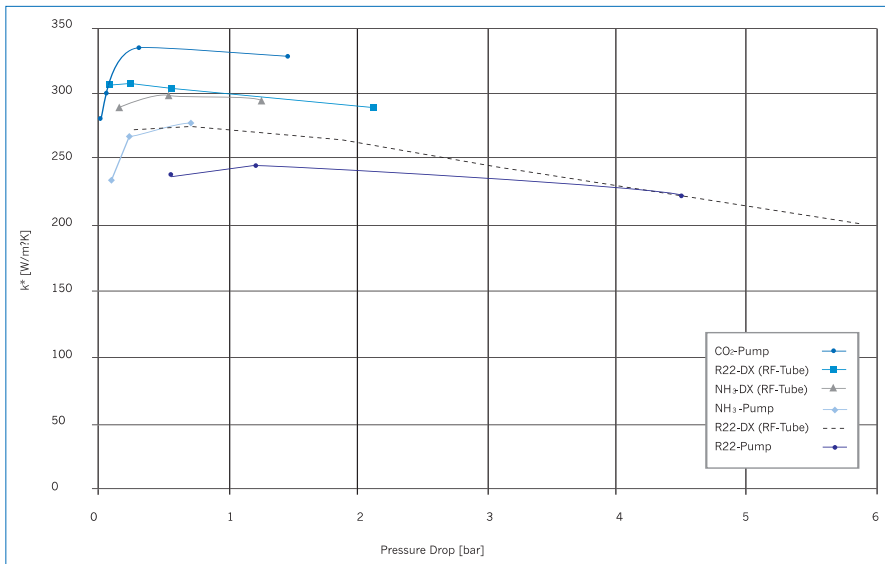*Fig. 7: Internal heat transfer with pump operation*

Fig. 8: Heat load of various refrigerants in different modes of operation

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