Because of the high-volume flows, most manufacturers use turbo machines as compressor for their refrigeration systems with water as the refrigerant[2)]. These are usually designed as radial turbo-compressors. An Israeli company is still the manufacturer with the highest number of large vapour compression refrigeration systems using water as the refrigerant. Corresponding installations can be found in Denmark, Germany, Japan, Switzerland, and South Africa. Two German companies also have some systems in operation, for example, at the University of Essen, at an automobile manufacturer in Dresden, at Daimler in Sindelfingen, and at the Zwickau University of Applied Sciences.

Water has a great future as a refrigerant – especially for high-temperature heat pumps. The higher the application temperature, the higher the vapour pressure of the water, and the disadvantages mentioned above have less impact.

1.5 Literature

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2) Oil-free screw compressors derived from vacuum pumps are under development.

2 Thermodynamic properties of natural refrigerants

Michael Kauffeld

When selecting the refrigerant, several properties, sometimes contradictory, have to be taken into account. These can be divided into three main groups:

- thermodynamic properties for the design and operation of the plant and its energy efficiency;
- physical, chemical and physiological properties that guide the selection of materials and processes to achieve the safety of equipment and personnel;
- environmental properties like ozone depletion or global warming potential, contribution to ground smog and impact on water, flora and fauna.

In addition, local, national and international requirements, such as laws, regulations and standards, as well as specific customer requirements must be taken into account. Last but not least, the operating and investment costs are of paramount importance when choosing the system.

To assess the properties of a refrigerant, the following criteria among others must be examined and evaluated:

- vapour pressure level / saturation temperatures
- critical temperature and pressure
- molar mass, heat of vaporisation, specific heat capacity, viscosity
- volumetric cooling capacity

The size of compressor required depends on the volume of gas to be passed in order to generate the required cooling capacity. This depends on the suction-side saturation pressure and the latent heat of the refrigerant. Volume flow and suction pressure are roughly inversely proportional. Selecting a refrigerant with a higher suction pressure at the desired cooling temperature means that a lower volume flow is required, since the suction gas density is higher.

At the same time, the critical temperature of the refrigerant must be taken into account. For the conventional vapour compression cycle, the condensation temperature ought to be well below the critical temperature. This ensures that the latent heat of condensation on the high-pressure side almost corresponds to the latent heat of evaporation at suction pressure so that the proportion of flash gas at the expansion valve is reduced to a minimum.

The molar mass not only determines the gas density of the refrigerant and thus the displacement volume of the compressor, but also the speed of sound of the gas which limits the flow velocities that may not be exceeded in pipes, valves and at the compressor connections with reasonable pressure losses. Lightweight molecules like ammonia can be subjected to higher gas velocities without incurring substantial pressure drop.

In the following section, the thermodynamic properties of the natural refrigerants are compared.

Figure 2.1 compares the specific, mass-related latent heat of evaporation of different natural refrigerants. The enthalpy change of water is twice that of ammonia and six times that of propane. This reduces the mass flow required for a given cooling duty.

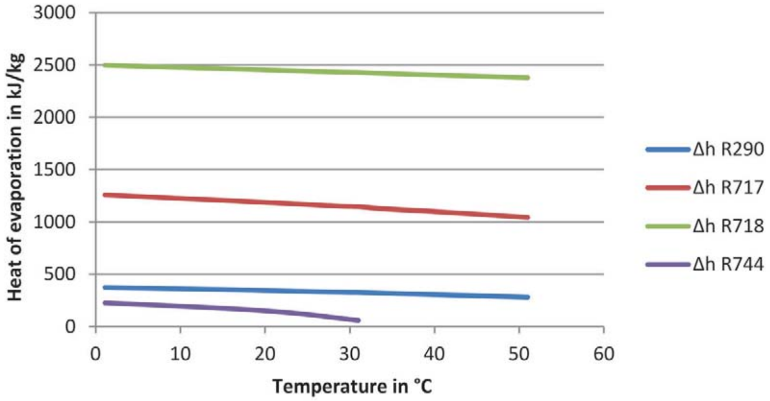

Figure 2.1 Latent heat of evaporation vs. temperature. Please note that the critical point of CO_2 is at 31 °C (87.8 °F) and that no phase change occurs and hence heat of evaporation does not exist for CO_2 above that temperature

Since the compressor size is more dependent on the volume flow rate of the refrigerant than the mass flow rate, the specific volume of the refrigerants at the compressor inlet, i.e. as first approximation the specific volume of the saturated vapour, is also important (see Figure 2.2).

Using the specific volume and the latent heat of evaporation, the specific volume flow per kW cooling capacity can be determined (see Figure 2.3).

The higher-than-average specific volume of water, compared to the other refrigerants, results from the low vapour pressure of water at the temperatures prevailing in refrigeration systems. This is significantly lower for water than for all other substances, see Figure 2.4. With similar low vapour pressures, the other refrigerants would also require correspondingly high-volume flows. However, these low pressures do not occur at the temperatures found in refrigeration systems with refrigerants other than water.

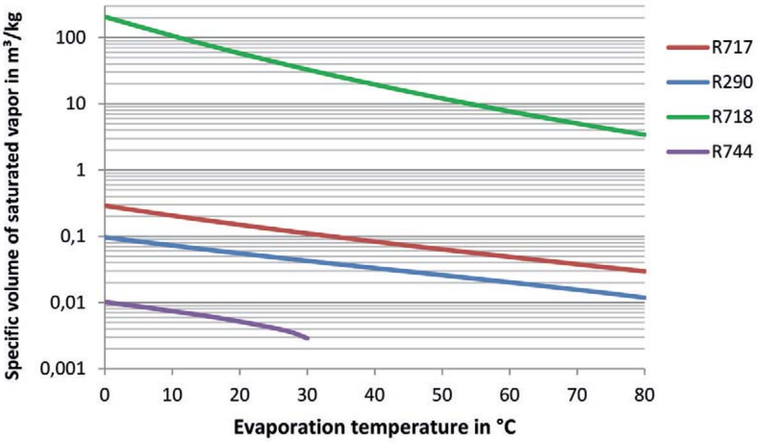

Figure 2.2 Specific volume of saturated vapour, i.e. on the dew point lines, for ammonia (R717), propane (R290), CO_2 (R744) and water (R718). Please note that the critical point of CO_2 is at 31 °C (87.8 °F) and no saturated CO_2 vapour exists above that temperature

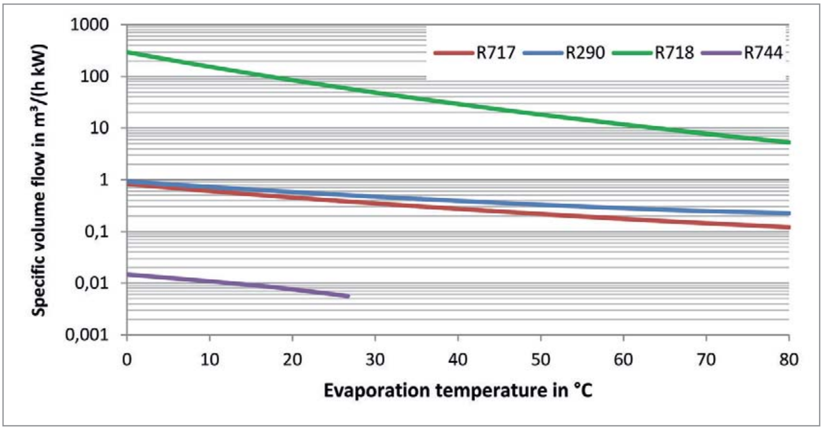

Figure 2.3 Specific volume flow per kW cooling capacity for ammonia (R717), propane (R290), CO_2 (R744) and water (R718). Please note that the critical point of CO_2 is at 31 °C (87.8 °F) and no evaporation of CO_2 occurs above that temperature.

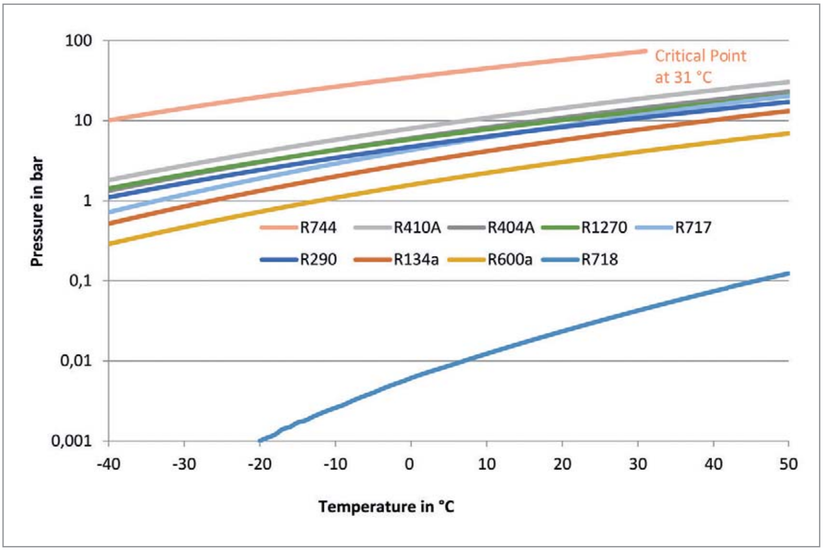

Figure 2.4 Pressure-Temperature relationship of various refrigerants

However, the decisive factor for any refrigeration application is not the pressure of the refrigerant, but the temperature required in the evaporator. Hence, for a given operating temperature, CO_2 with its comparatively high pressures leads to very compact refrigeration systems, while water with its very low pressures requires very large compressor, component and pipe dimensions.

For the design of evaporators, the pressure imposed on the refrigerant is important. This pressure may result from a column of liquid at the evaporator inlet, or a pump may produce it. Either way the result is that the boiling of the refrigerant is suppressed until either the pressure has dropped to the saturation point or the temperature has risen. This static head is, in effect, subcooling the liquid and making it more difficult to evaporate. Due to the low vapour pressures and the relatively high density of liquid water, the effects of a static head or liquid column on the boiling point of water as a refrigerant are significantly greater than with other refrigerants, as shown in Figure 2.5.

4.3 Plant engineering aspects

4.3.1 Plant systems and types

4.3.1.1 Flooded and dry expansion systems / liquid separators

Ammonia refrigeration systems are mostly designed and built as flooded systems. Dry-expansion systems are mainly used when the refrigerant charge has to be kept low. At the same time, the system structure for dry-expansion systems is simpler, and it corresponds more closely to the known technology, which is also used for synthetic refrigerants. Special care has to be taken to keep the superheat after the evaporator(s) of DX ammonia systems as low as possible due to the high isentropic coefficient (index of compression) of ammonia vapour, see chapter 4.2.1. With the right plant design and control strategy, DX ammonia systems can be built very energy efficient. Another argument for using dry expansion is the much easier oil return to the compressor. Most ammonia DX systems use lubricants which are miscible with ammonia, but with the correct plant design also non-miscible oils can be used and are already being used in ammonia DX systems. Oils miscible with ammonia that are suitable for this purpose are significantly more expensive and must be used with special care due to their distinct hygroscopic properties.

Typically, a higher efficiency (COP/ EER) of the refrigeration systems can be achieved by the use of the flooded systems. Oil recirculation is more complex and the refrigerant charge is higher, but both oil recirculation and the reduction of the refrigerant charge can easily be made in a simple, manageable way. For example, the refrigerant charge of the flooded systems can be greatly reduced with the help of optimised heat exchanger designs and liquid separation systems.

For comparison, Figure 4.10 shows the simplified flow diagrams of the two systems.

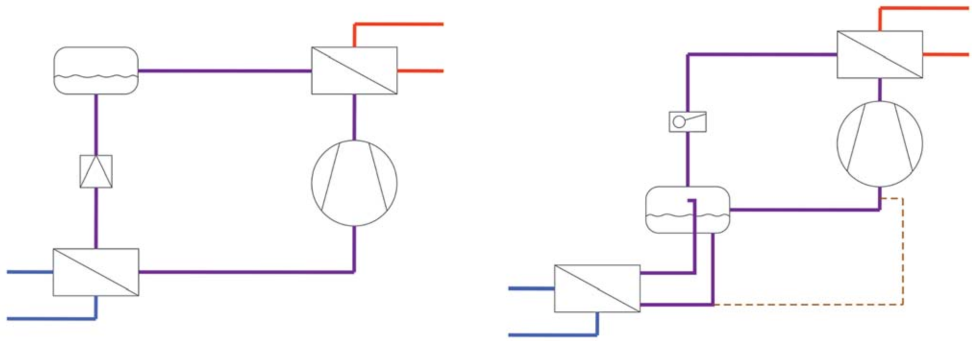

Figure 4.10 Dry expansion (left) and flooded evaporation (right)

4.3.1.2 Liquid separator

In flooded systems, the refrigerant liquefied in the condenser is expanded in the liquid separator using a high- or low-pressure float valve. The liquid separator protects the compressor by separating the liquid and gaseous phases coming back from the evaporator. Due to its density, the liquid falls to the bottom of the separator, and the gas can be extracted by the compressor.

When designing the liquid separator, it should be sufficiently large to accommodate the total volume of the system (receiver function) and also account for potential operating conditions which may lie outside the design conditions. This includes, for example, the start-up situation of

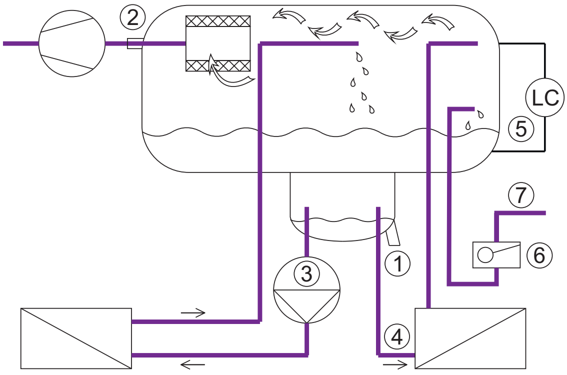

Figure 4.11
Schematic for liquid separator (also known as surge drum)

a refrigeration system with a warm secondary refrigerant (brine). Due to the higher temperature difference between the ammonia evaporation and the secondary refrigerant temperature at start-up, the performance of the evaporator is vastly increased. This leads to higher refrigerant volume flows on the return side of the evaporator. As a result, there may be liquid carry-over at the compressor inlet if the liquid separator is not large enough for these higher volume flows.

Figure 4.11 shows a schematic of a liquid separator. The left-side evaporator works with a refrigerant pump (3). The right-side evaporator (4) operates by gravity circulation. For illustration purposes, Figure 4.12 shows a compact liquid separator assembly with a built-in plate heat exchanger (evaporator).

Figure 4.12
Liquid separator (also known as surge drum) (© EM Polar)

4.3.1.3 Refrigerant pump circulation

When several evaporators are connected to an ammonia refrigeration system and are distributed over a large area (e.g., in a cold store), the system is usually designed for refrigerant pump operation, but some manufacturers also apply direct expansion for multi-evaporator systems. The evaporators are connected to a pipe network. This network consists of a supply and a return line and is connected to the low pressure (LP) liquid separator. With the help of one or more refrigerant pumps, the liquid refrigerant is continuously pumped through the pipes and is thus available to the evaporators. A simplified flow diagram is shown in Figure 4.13. The evaporators are fitted with an electrically controlled solenoid valve on the incoming liquid side.

The solenoid valve opens when refrigeration is required, and the liquid refrigerant flows through the evaporator. The flow rate is often higher than the amount of refrigerant actually required. In practice, a multiplication factor of the mass flow between three and five times has proven itself and is called circulation rate. In this case, one speaks of a flooded evaporation system. The circulation rate should, on the other hand, not be too high, because it will unnecessarily increase the energy needed for the pump(s) decreasing the overall ammonia plant energy efficiency. Due to the very high evaporation enthalpy of ammonia, and despite the increased circulation rates, only relatively small refrigerant components are needed. This reduces investment costs for pumps, pipes, fittings etc., while at the same time, the operating costs of the circulation pumps are cheaper than for systems with synthetic refrigerants.

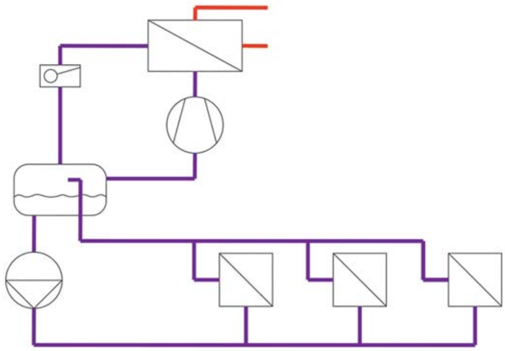

Figure 4.13 Pump recirculation

4.3.1.4 Two-stage systems

The use of two-stage systems is particularly useful when large temperature lifts (and therefore large pressure differences) have to be overcome.

To limit the compression ratio and the hot gas temperature, a two-stage compression process should be selected at low evaporation temperatures or at high external pressure conditions. This results in better system conditions and, above all, better performance figures. For the design and construction of the two-stage system, a distinction must be made between systems with reciprocating piston compressors and systems with screw compressors. In the case of reciprocating compressors, it is important that the hot gas (shown in Figure 4.14) is cooled after the first compression stage, and only then is it led into the second compression stage. Cooling takes place by injecting liquid refrigerant into a space between the first and second compression stages. It is important to ensure that no condensation can form in this space. One way to reduce the risk of condensation is to use an additional heat exchanger. This variation (Figure 4.15) is often used in systems with synthetic refrigerants, but is not common in ammonia systems. In systems that are designed with screw compressors, the intermediate cooling of the gas is less critical because the gas temperature is cooled by the oil injection into the compressor. However, it is recommended that any flash gas due to the first expansion stage be drawn off through the high-pressure (HP) compressor.

5.3 R744 refrigeration system characteristics and their effect on design and operation

Note: All pressure data refer to the absolute pressure in bar, unless otherwise stated. All pressure-enthalpy diagrams refer to logarithmic pressure scale.

5.3.1 Introduction and definitions

CO_2 can be distinguished from other fluids in many areas. This is of great importance for design, operation and maintenance. The main differences are:

- low critical temperature (31,1 °C / 87.9 °F)
- triple point pressure is above atmospheric pressure (5,18 bar / 75.1 psi)
- generally higher pressures than other refrigerants at similar temperatures
- particularly favourable heat transfer properties
- high thermal expansion coefficient in the liquid phase

The pressure-enthalpy diagram is often used as the basis for process analysis in the fields of refrigeration technology. The pressure-enthalpy diagram in Figure 5.1 shows the physical states of CO_2. These are dependent on pressure and temperature.

CO_2 phase diagrams for various pressure ranges are available.

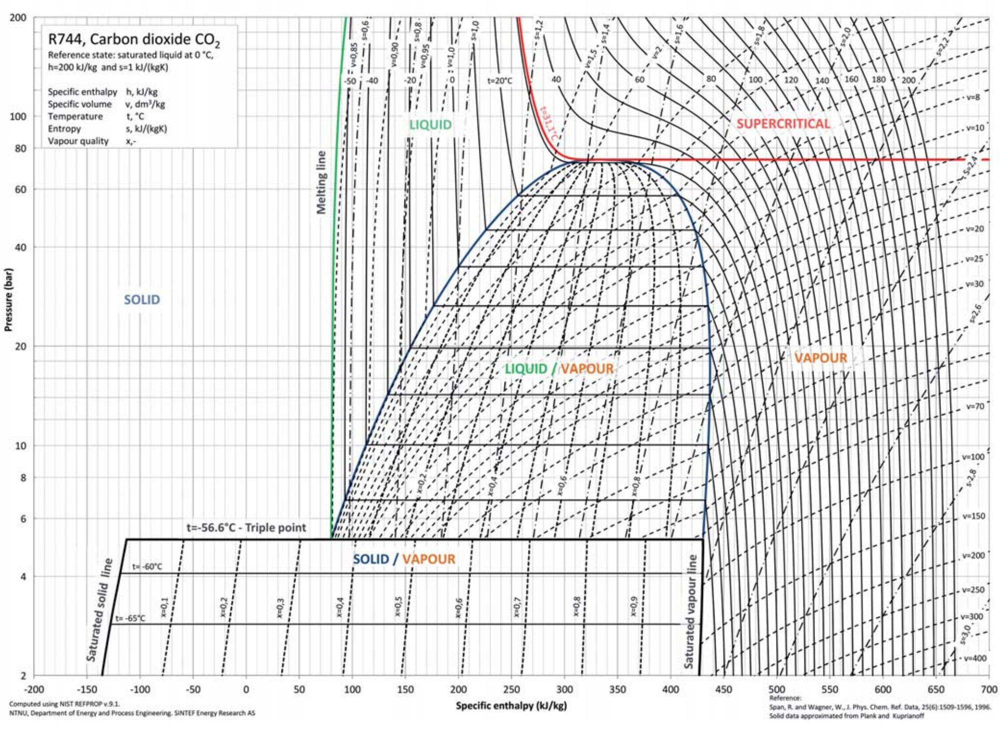

Figure 5.1 Logarithmic pressure-enthalpy diagram for CO_2 incl. phase regions

The boundary lines for saturated vapour and liquid meet at the critical point. The critical point is a combination of pressure (critical pressure p_{cond}) and temperature (critical temperature t_{cond}), where the liquid and vapour phases meet. CO_2 has a critical pressure of 73,8 bar (1070.4 psi) and a critical temperature of 31,1 °C (87.98 °F).

At subcritical pressure, the liquid and vapour phases, and possibly also the solid phase, occur either simultaneously or separately. There is only one phase in the supercritical area.

If the pressure is above the critical pressure and at the same time, the temperature is above the critical temperature, the CO_2 is in the *supercritical* area. In Figure 5.1 the supercritical area shows as the marked area to the right of the 31.1 °C isotherm and above the critical pressure of 73,8 bar (1070.4 psi). In the supercritical area, CO_2 is a gas with a very high density. It behaves more and more like a liquid when the fluid temperature drops.

The *triple point* of a substance is the only possible state in which all three states – solid, liquid and vapour – are in equilibrium. At lower pressure levels than the triple point, solid and vapour are again in equilibrium; above the triple point, there is liquid and vapour. The triple point of CO_2 is at a pressure of 5,18 bar (75.1 psi) and a temperature of −56,6 °C (−70 °F). Solid CO_2 is also called "dry ice" because it sublimates, it changes directly from solid to gas, at atmospheric pressure.

In the pressure-enthalpy diagram the triple point appears as a line that describes a different distribution of the three phases (= −56,6 °C / −69.9 °F) line at 5,18 bar (75.1 psi) in the diagram). The triple point with its potential solid content usually plays a subordinate role in refrigeration technology since this area below the triple point is often avoided.

The two-phase area of vapour and solid is below the triple point line. As in the two-phase area (liquid/vapour), the vapour fraction is determined or specified as an "x" value (x = vapour fraction). The saturation temperature of the solid CO_2 depends on the corresponding saturation pressure and the sublimation line (bottom left in the diagram). At atmospheric pressure, the saturation temperature for solid CO_2 is at −78,5 °C (−109 °F), which corresponds to the sublimation temperature of dry ice.

5.3.2 Importance of the relatively low critical temperature

5.3.2.1 Transcritical process

Unlike conventional refrigerants, the critical temperature is rather low at 31,1 °C (87.9 °F), which is a common temperature for condensing the refrigerant when the heat is rejected to air or water. This difference is essential for the design and the operation of R744 systems that transfer heat in this temperature range (or higher) on the high-pressure side.

In the supercritical area, there is no longer any condensation when the heat is rejected. Therefore, refrigeration systems and heat pumps with CO_2 in the conventional (subcritical) process cannot have higher condensing temperatures than ~30 °C (86 °F). In practice, an even lower condensing temperature of around 27 °C (81 °F) has actually to be considered as the upper limit for the conventional subcritical vapour compression cycle with CO_2.

However, the refrigeration or heat pump process can also be maintained if the heat sink temperature rises above 31 °C (88 °F). Then the heat is rejected on the high-pressure side of the system in the supercritical area. A transcritical cycle occurs, as shown in the pressure-enthalpy

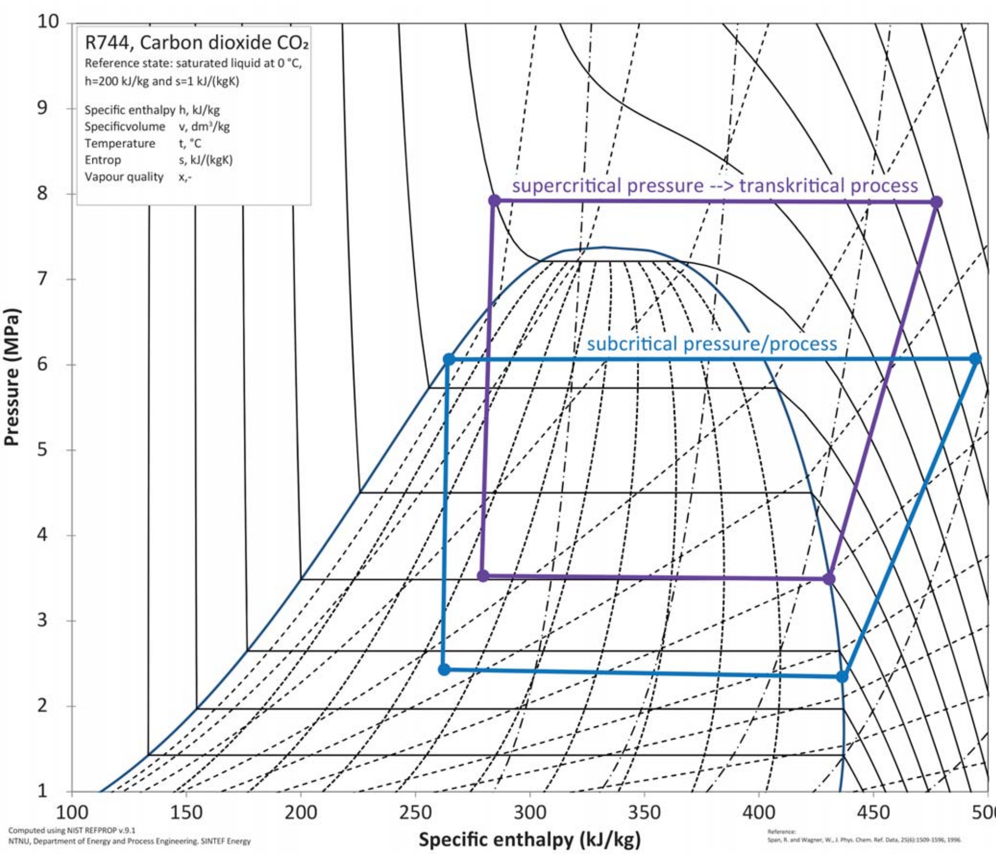

Figure 5.2 Subcritical and transcritical CO_2 processes within the pressure-enthalpy diagram

diagram in Figure 5.2. Instead of condensation, not only is the usual superheated part rejected in the so-called gas cooler, but the entire heat load is transferred at gliding temperatures on the refrigerant side. The heat absorption in the evaporator takes place as with conventional systems in the two-phase area. In efficient and well-designed systems, however, superheating in the evaporator is deliberately avoided since the ratio $\Delta t_{evap}/\Delta p_{evap}$ is close to 1 K/bar (1 K/14.5 psi); i.e., in order to achieve a 1 K difference in the saturation temperature in the evaporator, the saturation pressure must be lowered by 1 bar (14.5 psi). The transcritical process is described in more detail in Chapters 5.4 to 5.6.

If the external conditions allow a conventional subcritical process, this usually is the most energy-efficient variant. However, if a very high temperature lift is desired on the heat sink fluid side, the gliding temperatures in the gas cooler of the transcritical process are advantageous. These make it possible to use the heat more effectively than in a conventional condenser. It is thus possible to heat water (or other fluids) in a gas cooler from, for example, 10 °C (50 °F) to relatively high temperatures (of max. 90 °C / 194 °F). This makes heat recovery with R744 refrigeration systems both very rewarding and flexible with regard to the application of waste heat utilisation.

If the temperature level of the heat sink medium varies greatly, as can be the case, for example, with outside air, the lowest energy consumption of the refrigeration system is achieved by actively controlling the high-pressure level according to the heat sink temperature, see Chapter 5.4.5. The refrigeration system then alternates between transcritical and subcritical operation. In the case of active use of waste heat, i.e. most waste heat is used for heating purposes, the R744 system could operate permanently in transcritical mode.

5.3.2.2 Lower effective coefficient of performance

A simple R744 refrigerant circuit with an air-cooled condenser works – in moderate climates – on the high-pressure side close to the critical pressure. This causes larger losses during the throttling through the expansion valve. (A lot of liquid evaporates when throttling; see Figure 5.2.) The same challenge applies to the transcritical process at elevated heat sink temperatures.

These additional internal losses mean that the theoretical coefficient of performance (COP) of the reference process for R744 refrigeration systems is lower than for conventional refrigerants, as shown in Figure 5.3.

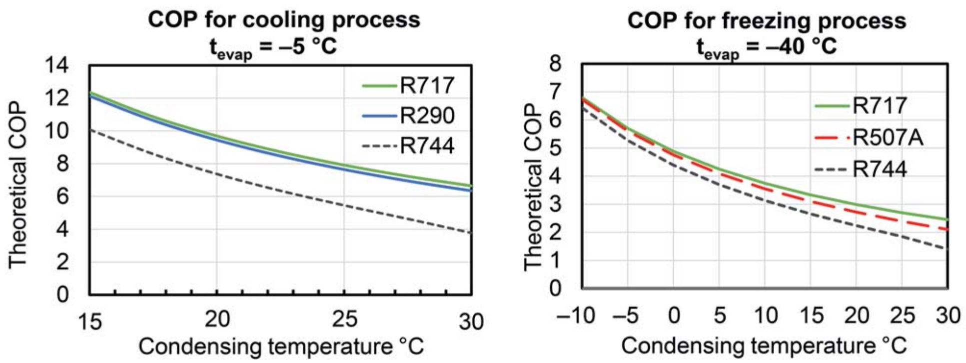

Figure 5.3 Comparative theoretical performance COP for medium temperature –5 °C/–23 °F (left) and low temperature –40 °C/–40 °F (right)

Figure 5.3 shows the theoretical performance figures for a medium- and low-temperature process with R744, R507 or R290, and R717 at an evaporation temperature of –40 °C (–40 °F) or –5 °C (23 °F) with increasing condenser temperatures. (In practice, these graphs and comparisons look different for several reasons, as will be explained in the following chapters.)

5.3.2.3 Other conditions

A low critical temperature can also be of practical importance in other contexts. Some of the following examples are used to determine the pressure in a system, part of a system or a component exposed to high ambient temperatures (above the critical temperature). They also show how to calculate the amount of refrigerant discharged if the safety relief valve opens.

The specific heat capacity of CO_2 changes in the supercritical range. Near the critical point, it varies very strongly with temperature changes. For illustrative purposes, consider the distance between the isotherms in this area in the pressure-enthalpy diagram in Figure 5.1. This relationship is important for the design and operation of heat exchangers for heat rejection in this temperature range; see also section 5.4.

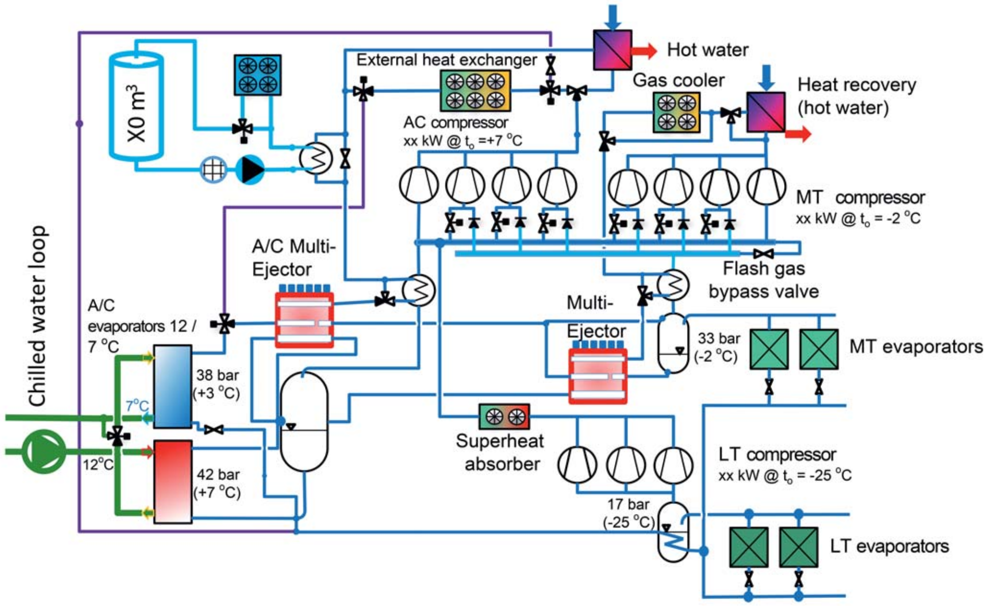

Figure 5.55 Example of a future integrated circuit for supermarkets

Supermarket cooling, future options for integrated systems

Modern supermarket systems are considered to be integrated systems in the sense that both the hot and the cold sides of the refrigeration system are utilised and that all the essential thermal systems of the building are connected to the central refrigeration / heat pump unit. In the previous sections, some examples were shown of how air conditioning can be provided, even on hot summer days. However, integrated solutions are constantly being developed.

The development of such systems generally focuses on reducing their unit costs while maintaining equal or even improving the energy efficiency. Nowadays, end users have the option to implement standardised, energy-efficient CO_2 refrigeration systems with integrated solutions for air conditioning and for heat pump functions from different manufacturers. The energy requirement of appropriately modified, integrated CO_2 refrigeration systems will be reduced by up to 30 %, compared to standard booster systems with separate HFC air conditioning systems. Figure 5.55 shows a circuit of an advanced CO_2 refrigeration system.

Compressor allocation:

The allocation of compressors to the respective suction group is to be flexible with respect to demand and to reduce the number of installed compressors for the future. The different compressors can be assigned either to the medium-temperature cooling suction manifold or to the parallel compressors, as required. The central operational control determines which compressors maintain the pressure level in the low-pressure receiver (MT standard cooling) and in the intermediate-pressure receiver (flash gas, refrigerant supplied by the ejector from the low-pressure receiver of the MT side and air conditioning) most effectively at the corresponding target pressure level.

Integration of the air conditioning:

Chilled water, for the air conditioning of the building, can be provided in a very energy-efficient manner with an ejector-supported refrigeration unit. The chiller evaporator consists of two heat exchangers that are connected in series on the water side. The evaporator, which is connected to the return water line from the air conditioning system, is gravity fed with refrigerant from the intermediate-pressure receiver. In the example shown, the saturation temperature in the intermediate-pressure receiver will be 7 °C/42 bar (45 °F/609.2 psi) when the water returns from the air conditioning system at 12 °C (54 °F). Depending on the cooling demand of the building, the supply temperature of the water can (and should) be adjusted. If lower temperatures are required in the warm season due to an increased cooling demand, the water can be further cooled using the second evaporator. The evaporated refrigerant from the second evaporator is then fully drawn in by the ejector. This ensures a pressure increase of up to 4 bar, if appropriately designed. In this case, the evaporation temperature in the second heat exchanger is lowered to 3 °C/38 bar (37 °F/ psi). Therefore, a chilled-water supply temperature of 7 °C (45 °F) is possible with a suction pressure of 42 bar (corresponding to +7 °C (45 °F) saturation temperature of the CO_2) at the compressor.

External cold absorber:

In climate zones with very high ambient temperatures during the warmest hours of the day and reduced temperatures during the night, it may be economical to include an existing external cold thermal storage unit. An example of such a cold storage device could be the water contained in the building's fire extinguishing water tank. This water can be used as an additional heat sink during the warmest time of day. As a result, the refrigerant temperature downstream of the gas cooler can be further reduced to values below the ambient temperature. During the night, the absorbed heat in the water stored in the fire extinguishing water tank can dissipate into the surroundings via heat exchangers. If available, existing heat exchangers from the previous refrigerating system, which is supposed to be replaced, could be applied for the purpose of night time heat rejection.

Supermarkets with fresh food counters and limited low-temperature demands

In climates such as those found in Central and Northern Europe, the use of ejectors to increase efficiency during the relatively few operation hours at high outside temperatures is often not economically viable. For certain applications, the ejector technology can nevertheless be used advantageously. For example, in markets with open meat and fish counters, it is often difficult to work with the same evaporation temperatures as are required for the closed food counters in the same market. Figure 5.56 shows how these open-display counters can be specifically operated with slightly reduced evaporator temperatures by using an ejector (EJ_{MT_Minus}) without the entire MT suction group's having to work with reduced suction pressures. This leads to a significant energy saving for the market operator, which amortises the additional effort (separate suction line and ejector) in a short time.

In some market configurations, most of the medium- and low-temperature counters are self-contained units that either reject the waste heat to a water cycle or directly into the market's indoor air. In future, instead of using LT compressors for a limited number of evaporators, the integration of these individual LT evaporators in a booster system could be achieved with the

6.4 Safety of refrigeration systems with hydrocarbons

A safety concept considers safety, health and environmental hazards during the life cycle of a hydrocarbon refrigeration system, e.g. production, transport, operation, service/ maintenance, disposal/ recycling, etc. Regarding flammability, there are regulations and responsibilities for manufacturers, planners, users, operators, etc. and specific requirements apply to smaller and larger systems for system engineering, design, production as well as safety devices and measures. There are training courses where expert knowledge of safe use of flammable refrigerants may be acquired by technical personnel.

A thorough and well-regulated process of hazard identification, risk analysis and mitigation yields quick identification of weaknesses and reaction before a serious damage incident occurs.

The manufacturer is responsible for the operational safety comprising inherently safe construction, suitable safety devices and other protective measures. Relevant regulative requirements concerning product safety must be complied with during production and manufacturing of a refrigeration system. Once a refrigeration unit is assembled, its conformity to product standards, i.e. technical, safety, etc., must be declared; rules and regulations are established at international / national levels, and local requirements may apply.

For operators of larger systems, there are further obligations with regard to occupational, health and environmental safety on site, as well as system maintenance and documentation.

Limitation of the extent of damage

Flammable gases form explosive atmospheres with air. If a hazardous area is of non-negligible extent, where concentration throughout the space exceeds the LFL it is appropriate to include construction features to relieve an explosion, in the unlikely event of ignition, to minimise the possibility of structural damage. This is usually achieved by the use of a frangible part of the ceiling or the wall, providing relief to the outside, corresponding to at least 20 % of one wall area. (Although it should be evaluated on a case-by-case basis.)

Risk assessment of refrigeration systems

Risk assessment requires an iterative approach and includes:

- identification of hazards and unintended use of the refrigeration system during all life cycle phases
- analysis of risks arising from the identified hazards
- implementation and documentation of constructive measures to eliminate hazards and of safeguards to reduce or minimise the risks of remaining hazards

Risk assessment of larger systems must also consider that dangerous situations arise when persons are on site who lack knowledge of the intended use and operation of refrigeration systems.

Depending on a hazard situation and probability of its occurrence, accumulation of several hazards must be considered as well.

Regardless of size, all hydrocarbon systems should be subject to risk assessment

Of course, risk assessment procedure is required for all systems, regardless of the refrigerant type (flammable or non-flammable, etc.). Numerous different techniques are available for risk assessment – the most appropriate for the situation should be selected.

Safety class A3

Hydrocarbon refrigerants belong to the safety class A3. The designation describes the:

- toxicity class A (low toxicity or no significant adverse health effects on humans at concentrations below 400 ppm)
- flammability class 3 (higher flammability or LFL of the refrigerant-air mixture at less than 3,5 % v/v)

The banned chlorofluorocarbons belong to safety class A1. Hydrochlorofluorocarbons, fluorocarbons, and hydrofluoroolefins (unsaturated fluorocarbons) belong to safety classes A1, A2, A2L and A3. Hydrocarbons are lower toxicity but higher flammability, so it is flammability that dictates their application in terms of charge limits.

Table 6.12 Selected safety data of some hydrocarbons according to EN 378–1, Table E.1

Refrigerant no	Chemical name	Safety classification	ATEL/ ODL in kg/m^3	LFL in kg/m^3	Normal boiling point in °C	Normal vapour density in kg/m^3	Auto-ignition temperature in °C
170	ethane	A3	0,0086	0,038	–89	1,23	515
290	propane	A3	0,09	0,038	–42	1,80	470
1270	propene	A3	0,0017	0,046	–48	1,72	455
600a	isobutane	A3	0,059	0,043	–12	2,38	460

Table 6.13 Selected safety data of some hydrocarbons according to EN 378–1, Table E.1

Refrigerant no	Chemical name	Safety classification	ATEL/ ODL in lb/ft^3	LFL in lb/ft^3	Normal boiling point in °F	Normal vapour density in lb/ft^3	Auto-ignition temperature in °F
170	ethane	A3	0.00054	0.00237	192.2	0.07679	959
290	propane	A3	0.00562	0.00237	–43.6	0.11237	878
1270	propene	A3	0.00011	0.00287	–54.4	0.10738	851
600a	isobutane	A3	0.00368	0.00268	10.4	0.14858	860

6.4.7 System tightness

Refrigeration systems should be as tight as possible, irrespective of the refrigerant used; for safety, environmental and cost reasons. For hydrocarbons, the emphasis is for minimisation of flammability risk. Components, joints and fittings should as far as possible be brazed and comply with ISO 14903. Other aspects associated with the construction of the system, such as vibration (from compressor and other moving machinery), external corrosion, fretting, external impacts, etc. should also be considered. Minimising leakage is addressed in other safety standards such as EN 378, ISO 5149 and EN 1127-1.

6.4.8 Charge minimisation

When using flammable refrigerants, an effective measure to reduce flammability risk is to minimise the refrigerant charge. It is desirable to minimise the system charge as much as possible, some options to this end are presented.

In order to yield the lowest possible refrigerant charge, systems are optimised for least charge per kW capacity. Refrigerant circuits are charged precisely to suit given operating points so excess charge is avoided. Dry expansion systems yield lower charges; flooded systems usually demand much greater charge. Components with small internal volumes, e.g., air-cooled condenser with microchannels, are used to reduce charge. Some applications like heat recovery employing a desuperheater require a receiver to account for all operating points.

To further minimise refrigerant charge per circuit, it is possible to increase the number of circuits at a given capacity. For example, it may be possible to design a unit with multiple refrigerant circuits with smaller capacity, each requiring a smaller charge, instead of just one circuit with large capacity and large charge.

6.4.9 Determination of limited refrigerant charge

Determination of charge limits is handled differently across the various safety standards. EN/IEC 60335-2-40 and EN/IEC 60335-2-89 apply charge limits according to the design of the equipment. Charge limits in EN 378 differ according to the design of the equipment, but also the type of occupancy within which the equipment is located; these are described by the installation area “access category” and system “location classification”.

Installation site access category of the refrigeration system depends on the potential movement and competence of the persons allowed in the location of refrigerant-carrying equipment. Table 6.14 summarises those access categories.

Table 6.14 Categories of access areas of installation sites of refrigeration systems in accordance with EN 378–1:

Access category	Explanation
a) General access area	The following persons may be present: • sleeping people • people with limited mobility • unknown number of people • people without knowledge of the hazard Examples: hospital, prison, theatre, train station, hotel, residential apartment
b) Monitored access area	The following persons may be present: • limited number of people • at least some people are familiar with the safety arrangements of the facility Examples: office or business premises, manufacturing and workplaces
c) Access area for authorised persons only	Restricted access where the following persons may be present: • all are familiar with the safety precautions of the facility • materials or goods are manufactured, processed or stored there Examples: refineries, cold stores, dairies

The location classification refers to the arrangement of the refrigerant-carrying parts as shown in Table 6.15.

Table 6.15 Location classification in accordance with EN 378

Location classification	Explanation
Class I: Mechanical devices in occupied areas	Class II does not apply, and refrigeration equipment is located in the occupied area.
Class II: Compressors in the machine room or outdoors	Class III does not apply, and refrigerant-containing parts of the refrigeration system are arranged as follows: • all compressors and pressure vessels within the machine room (according to EN 378–3) or outdoors • Pipes and valves possibly in public areas
Class III: Machinery room or outdoors	All refrigerant-carrying parts arranged in the machinery room (according to EN 378–3) or outdoors.
Class IV: Ventilated housing	All refrigerant-containing parts inside ventilated housing (according to EN 378–2 and –3).

In general, the maximum allowable charge of a refrigeration system using any refrigerant is the lower of the two limits determined by toxicity and flammability. For flammable refrigerants, there are then two limits:

- The allowable charge limit, which is usually based on the room size · LFL · a factor smaller than 1, thereby ensuring the entire room volume cannot become flammable.
- An upper charge limit, or "cap" limit (m_1, m_2, m_3), which constrains the charge below a prescribed quantity, is assumed to reflect a threshold of releasable energy (from combustion). This upper charge limit applies regardless of the room size.

Depending on the location of a system's refrigerant-carrying parts and compressor, on its application and on the access category of the location, the maximum allowable charge is determined.

Note that the charge limits are based on individual system (circuit) charges, rather than the total charge. In other words, maximum allowable amount of refrigerant would be based on the largest system (circuit) charge of a multi-circuit unit and not the summation of all the circuits.

Table 6.16 Maximum allowable refrigerant charge for human comfort cooling refrigeration systems with flammability class 3 refrigerants according to EN 378–1, Table C.2

Category of access area	Classification of location			
	I	II	III	IV
a	m_1 or m_{max} and maximum m_2 or 1,5 kg (3.3 lb)		5 kg (11 lb)	m_3
b			10 kg (22 lb)	
c			no limit on the refrigerant charge	

7.4 Application in the low power range

7.4.1 Compact water chillers

The majority of the projects shown in Chapter 7.2.3 are in the cooling capacity range over 500 kW (2.8 TR) and at chilled-water temperatures below 10 °C (50 °F). Exceptions are Kawasaki and Sasakura. In addition to the cooling capacity, systems from Efficient Energy GmbH also differ from the other manufacturers in the temperature range of the chilled water generated since they specialise in chilled-water temperatures above 16 °C (60.8 °F). This made it possible to offer a compact system that closes the gap between classic chilled-water generation and cooling tower applications. It can therefore be assumed that the relevance of R718 in the field of vapour compression refrigeration systems will continue to increase and will establish itself as a serious refrigerant in the field of vapour compression refrigeration machines.

The chiller essentially consists of two vessels, two plate heat exchangers, two hydraulic shunts and the connecting pipes (Figure 7.8). The two vessels in the system work according to the principle described in Chapter 7.3 with external heat exchangers, whereby the inlet of the evaporator of the second vessel is connected to the outlet of the condenser of the first vessel and is therefore connected as a cascade to the first stage. The two plate heat exchangers represent the delimitation of the vacuum system and the interface to the secondary side. The two hydraulic shunts integrated in the hydraulic connection lines can be used to switch between different operating

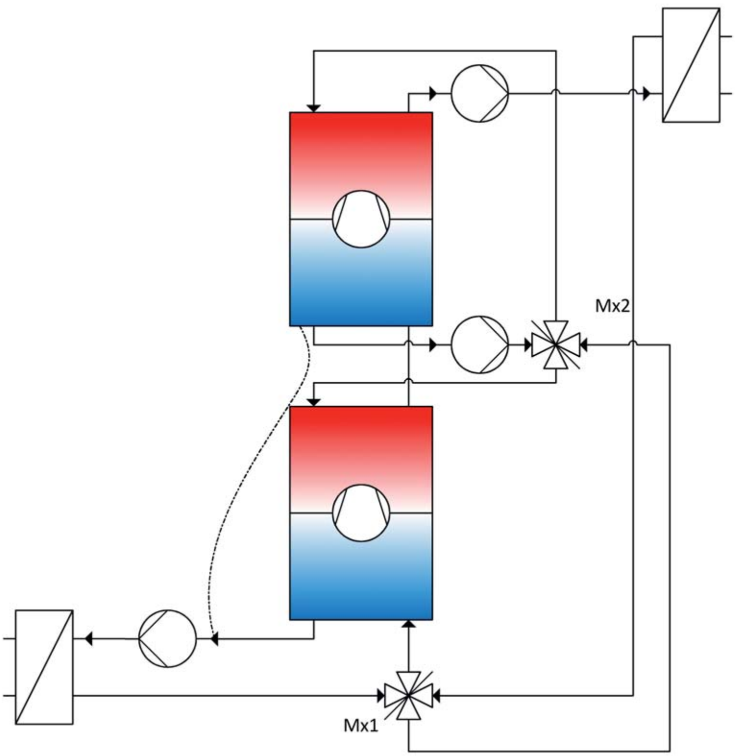

Figure 7.8
Schematic structure of the chiller from the manufacturer Efficient Energy

states. This happens automatically depending on the cooling water temperature in relation to the desired chilled-water temperature and ranges from pure free cooling operation to single-stage and two-stage operation at high cooling water temperatures.

7.4.2 Practical examples

Cooling server rooms [7-18]

The task was to cool a highly efficient data centre with a maximum cold air temperature of 25 °C. The special thing about this application was the slowly growing demand for cooling power due to the gradual expansion of the servers. The challenge was the initially low cooling capacity of around 8 to 10 kW (2.2 to 2.8 TR), which corresponds to approximately 10 to 15 % of the installed cooling capacity.

As a solution, the server racks were set up in a warm aisle enclosure in the server room. The cooling air supply is provided to the servers by side coolers.

After commissioning in summer 2016, it quickly became clear that operation was highly efficient even at this low partial load. This is due to the infinitely variable turbo compressors in connection with the integrated free cooling.

In spite of the initially low cooling capacity required, this installation was able to achieve a coefficient of performance of more than 18 based on measurement data over a period of 18 months.

Cooling of building [7-19]

In this application, a R718 chiller was combined with a photovoltaic system for the energy supply.

Water as a refrigerant has the advantage that no separate machine room is required for installation. This enabled the chiller to be installed in an adjoining room to the staircase. To compensate for fluctuations in electricity generation, a 28 kWh (95600 Btu) electricity storage system was installed, which can cover a complete daily cycle in summer. The combination of electrical and thermal storage (chilled water hydraulics) in connection with the part-load efficient turbo compressor covers the cooling during the day, including pre- and post-cooling of the building structure. At favourable outside temperatures, partial or complete free cooling is carried out at night, which significantly extends the useful life of the battery storage.

By using ceiling cassettes, the optimum chilled-water inlet temperature of 16 °C (60.8 °F) could be set for this system. In summer, there is a cooling load of 25 kW (7.1 TR). The gradual increase in the utilisation of the office space initially led to a load factor of the refrigeration system of only 40 to 60 %. Using the pre- and post-cooling of the building structure results in an annual performance factor (averaged EER) for the refrigeration system of over 12. This is unique in combination with a vapour compression refrigeration system for cooling office space with the possibility of unrestricted indoor installation of the refrigeration system.

Process cooling [7-20]

As an example of process cooling, the cooling of an extruder for the production of a continuous product made of plastic, such as a pipe, is presented. Because of the processing of various prod-