Refrigeration technology

Building Technologies

SIEMENS

Contents

1. Basic principles of thermodynamics

1.1 The origins of refrigeration

It is in the nature of things that human beings have been preoccupied with the subject of cooling since prehistoric times, and we know of many different possibilities of cooling. Liquids (such as wine) were cooled in clay jars wrapped with wet cloths or in canteens (heat extraction via evaporation of water).

The initial considerations on the topic of refrigeration known to us originate from the year 1834, when **Jacob Perkins** patented a vapor compression machine operating with a closed cycle and ethyl ether.

Some 40 years later in 1876, **Carl Linde** was the first to use ammonia as a refrigerant in a vapor compression machine with a piston compressor.

The first domestic refrigerators appeared in 1910, and in 1930 the refrigerants R11, R12, R13, R22, R113 and R114 were developed (see chapter 2). The demand for refrigeration originated in the food supply industry. Large-scale systems for breweries, slaughterhouses, cold storage houses and ice factories as well as refrigeration ships were built for this industry.

Demand increased greatly after the First World War, and a distinction came to be made between the following three fields:

- Industrial refrigeration
- Commercial refrigeration
- Domestic refrigeration

Situation today:

In the **food industry,** the use of refrigeration is the best and healthiest method of keeping foodstuff fresh for long periods and across different climatic zones, thus safeguarding supply.

The use of refrigeration in **mechanical engineering** allows for faster and cheaper production.

In air conditioning, not only heating energy is required in winter, but also refrigeration energy is required for cooling and dehumidification of the air during the summer period.

In earlier times, piped water or well water was frequently used for this purpose. However this usage is not recommendable from an economical or an energetic point of view.

The use of refrigeration is a major factor with regard to our well-being at the workplace and in occupied spaces.

In refrigeration today, mainly the compression refrigeration machine is used to produce the necessary refrigeration capacity.

- Cold stores
- Heat pumps
- Heat recovery in air conditioning

Energy or heat recovery is currently a very important topic in the refrigeration field. Due to the energy crisis in the 1970s and '80s, refrigeration enjoyed a boom with the increasing use of heat pumps.

The theory of refrigeration includes all fields of scientific enquiry (mathematics, technical science, physics, chemistry, etc.). There is hardly another industry that is as diverse as refrigeration.

It is important to be familiar with the fundamentals of refrigeration in order to be able to support the ventilation, air conditioning and refrigeration systems contractor or the consulting engineer with adequate basic knowledge for control implementation. Naturally, the relevance for the practical application of automatic control in refrigeration is the primary focus of this seminar.

1.2 The thermoelectric process

Structure of matter

From the field of thermo-hydrodynamics we are familiar with solid, liquid and gaseous substances. The reason can be found in the structure and properties of the molecules.

In **metals,** for example, the molecular structure is highly coherent (large cohesive forces); the molecules form a solid lattice (Fig. 1-1, left).

In **liquids,** these forces are not as great, so the substance is not solid (Fig. 1-1, middle).

In contrast, **gaseous substances** have only very loose cohesion, which is why they are highly volatile (Fig. 1-1, right).

Solid	Liquid	Gaseous
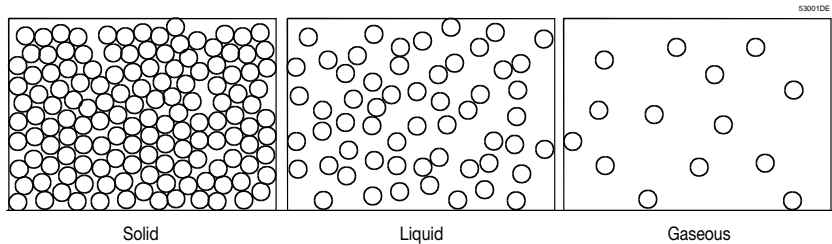
Fig. 1-1   Lattice structure of molecules
Solid: the molecules are firmly connected
Liquid: the molecules are loosely connected
Gaseous: the molecules have lost their cohesion and can move freely

1.2.1 Temperature-enthalpy diagram

The change of state can be easily illustrated using the temperature-enthalpy diagram. The starting point of the example shown in Fig. 1-2 is one kilogram of water at atmospheric pressure and 0 °C:
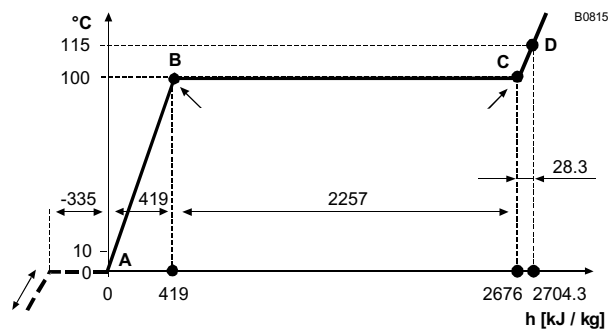
Fig. 1-2   Temperature-enthalpy diagram (t, h diagram) for water
A – B   Sensible heat (heating of the liquid)
B – C   Latent heat (vaporization, transition from liquid ⟹ gas)
A – B   Sensible heat (superheating of the gas)

As shown in Fig. 1-2, the unit "kJ/kg" is used for enthalpy h. This indicates that it must refer to the amount of heat contained in one kilogram of mass (in this case, water).

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Since normally only changes of enthalpy are of interest, i.e. enthalpy differences, the origin of the enthalpy scale can be defined anywhere. In Fig. 1-2 and in the commonly used water vapor tables, the origin is identical to the freezing point of water, i.e. 0 °C. This means, for example, that the latent heat of fusion described under "1.2.3 Melting process" is not included in the enthalpy values indicated.

Line A – B portrays the sensible heat that is required to heat the water from 0 °C to 100 °C. At point B, an enthalpy of 419 kJ/kg can be read off on the enthalpy scale. Therefore, the heat supplied in order to raise the water temperature to the boiling point is still present as the enthalpy of the water at 100 °C.

Line B – C portrays the vaporization process. Latent heat is continuously supplied along this line until the kilogram of water has been completely transformed into steam at point C. The enthalpy of this dry, saturated steam is now 419 + 2257 = 2676 kJ/kg, i.e. the sum of the sensible and latent heat. If the steam is superheated to 115 °C (point D), the enthalpy at point D is 2676 + 28.3 = 2704.3 kJ/kg.

In the refrigeration process, however, only the change of state of the refrigerant from liquid to vapor and from vapor to liquid is of interest. As described in the training module B01HV "Basic physical principles", every boiling point corresponds to a specific pressure (Fig. 1-3). High boiling points require high pressures, low boiling points low pressures.
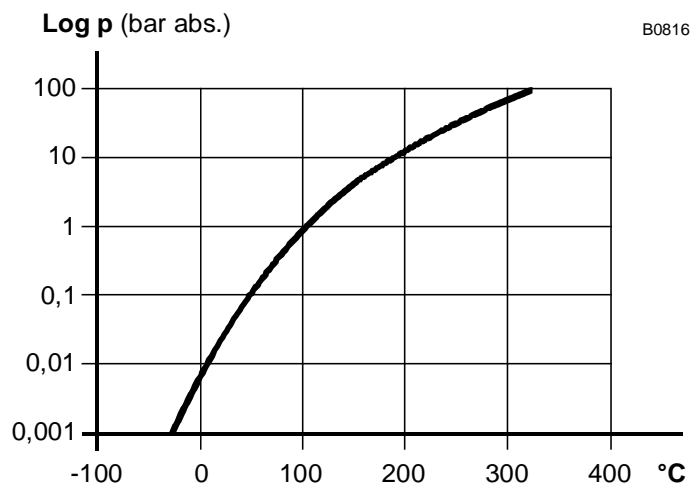
Fig. 1-3   Boiling points of water

1.2.2 Heat flow   Heat is a form of energy that, on its own, can only pass from a higher temperature substance to a lower temperature substance (Fig. 1-4). That means:

Heat only flows in one direction on its own.

and

a heat flow requires a temperature differential.
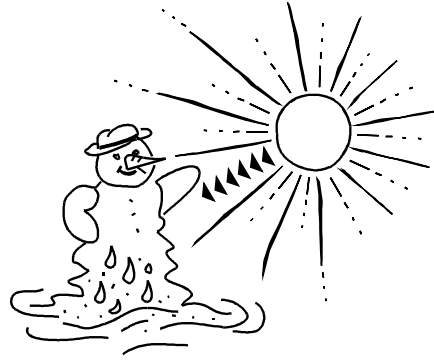
Fig. 1-4   Heat flow

1.2.3 Melting process

It is a known fact that solid bodies to which heat is supplied can assume the liquid state. The heat that causes a solid body to melt is referred to as the latent heat of fusion. In order, for example, to melt 1 kg of ice at 0° C, i.e. to liquefy it to water at 0° C, a quantity of heat of 335 kJ (Fig. 1-5) is required. Note that the temperature does not change due to the supply of heat. The melting process takes place at constant temperature.

Therefore, the heat that has brought about a change of state is referred to as latent heat.
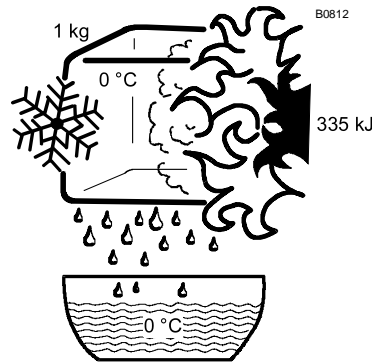
Fig. 1-5   Latent heat of fusion of ice

1.2.4 Vaporization process

If heat is supplied to the water at 0 °C, the water's temperature rises continuously until it eventually reaches its boiling point. Unlike the latent heat that brought about a change of state, this heat is measurable, i.e. sensible, which is why it is referred to as sensible heat.

The quantity of heat required to raise the temperature by 1 K is 4.19 kJ. Therefore, a quantity of sensible heat of 419 kJ must be supplied in order to raise the temperature of the water to 100 °C. Assuming standard pressure (atmospheric pressure at sea level, i.e. an absolute pressure of 1.013 bar), this temperature is the boiling point of the water. It is the point at which the water begins to evaporate (Fig. 1-6).

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100 °C

2676 kJ

100 °C   1 kg   419 kJ

2257 kJ
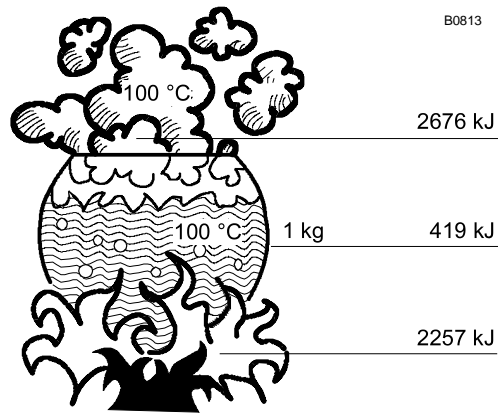
Fig. 1-6   Vaporization process and enthalpy of water

The vapor that is produced has a temperature of 100 °C and is referred to in technical terms as dry, saturated steam. The conversion of water to steam is again a change of state, which is caused by the uninterrupted supply of heat.

In order to completely vaporize the kilogram of water at 100 °C, i.e. to convert it into one kilogram of steam at 100 °C, the latent heat quantity of 2,257 kJ must be supplied. This latent heat quantity is the latent heat of vaporization.

If the 419 kJ required to heat the water from 0 to 100 °C is added to the latent heat of vaporization of 2,257 kJ, the resultant 2,676 kJ is the heat content or enthalpy of 1 kg of steam with respect to 1 kg of water at 0 °C (Fig. 1-2).

1.2.5 Superheating

If the dry, saturated steam at 100 °C now passes through an open coiled pipe, for example, through which additional heat is supplied to it, the steam is superheated (Fig. 1-7). The superheat is again sensible heat, i.e. the temperature of the steam rises.

In order, for example, to raise the temperature of the kilogram of steam produced by the vaporization process from 100 °C to 115 °C, a quantity of sensible heat of 28.3 kJ is required. This is the result of the following formula:

$$\dot{Q} = \dot{m} \cdot c \cdot (t-t_s)$$

$$= 1 \cdot 1.88 \cdot (115 - 100)$$

$$= 28.3 \ (kJ)$$

c_p Average specific heat of the superheated steam (kJ/kg K)
$\dot{m}$ Mass (kg)
t Temperature of the superheated steam (°C)
t_s Boiling temperature of the steam (°C)
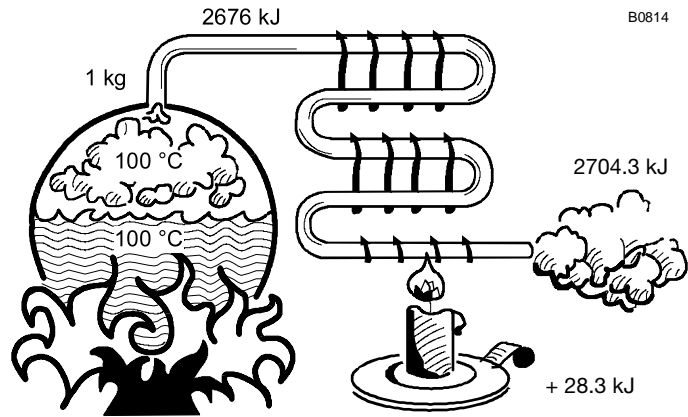
Fig. 1-7   Superheated steam and enthalpy of water

1.2.6 Liquefaction process (condensation)

The change of state from liquid to gas is a reversible process, i.e. the vapor can also be converted back into a liquid. This process, which is referred to as liquefaction, takes place if the same amount of heat is extracted from the vapor as was supplied to it during the vaporization process.
Fig. 1-8 illustrates the liquefaction process.
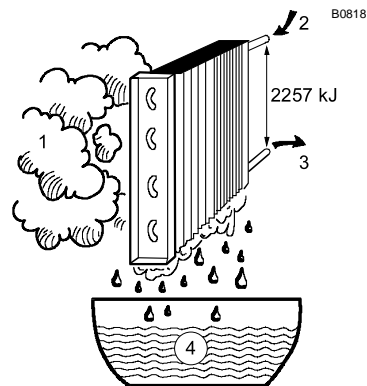
Fig. 1-8   Liquefaction process (condensation)
      1 Steam, 100 °C
      2 Cooling water, cold
      3 Cooling water, warm
      4 Water, ≤ 100 °C

The change of state from liquid to vapor and from vapor to liquid is of the utmost importance for the mechanical refrigeration or heat pump cycle for the following reason:
The change of state requires the transfer of a relatively large quantity of heat per kilogram of substance (refrigerant). As will be shown, it is precisely this heat transfer in the refrigerant cycle (see Fig. 1-2, C – B) that produces the desired effect (cooling in refrigeration machines, heating in heat pumps).

1.2.7 Pressure-enthalpy diagram

In refrigeration / heat pump engineering, the pressure-enthalpy diagram (Fig. 1-9) is used in preference to the temperature-enthalpy diagram shown in Fig. 1-2. This diagram does not show enthalpy as a function of temperature at standard pressure; instead it can be read off for various pressures and the corresponding temperatures. Above all, however, the diagram makes it especially easy to present and calculate so-called cyclic processes, as shown in the following.
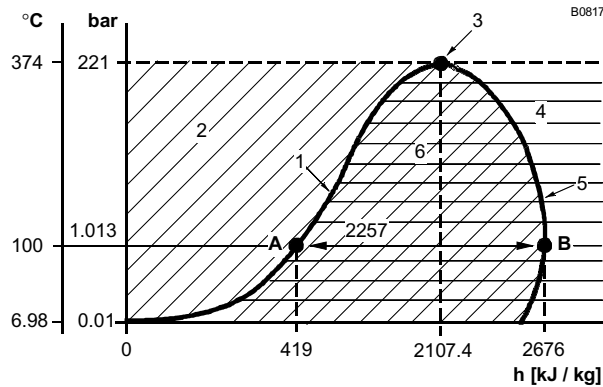
Fig. 1-9   Pressure-enthalpy diagram for water
    1   Liquid line (beginning of vaporization)
    2   Subcooling zone (liquid)
    3   Critical point
    4   Superheating zone (vapor)
    5   Saturated steam line
    6   Wet steam zone (liquid/vapor)
    A – B   Latent heat of vaporization (2,257 kJ/kg)

The line rising from the origin to the critical point (1) is the line of saturated liquid, whereas the continuation of the same line (5) from the critical point (the point at which there is no longer a difference between liquid and vapor) down to the enthalpy scale determines the state of saturated steam.
If a horizontal line is drawn through the two lines at a given pressure, the enthalpy of the saturated liquid can be read off at point A, and the enthalpy of the saturated steam can be read off at point B. The difference between the values at A and B corresponds to the latent heat of vaporization.
The diagram shows that the latent heat of vaporization decreases continuously with rising pressure and temperature until the state is finally reached at the critical point where there is no longer any latent heat of vaporization present.
In the case of water vapor, the critical pressure is 221.2 bar, and the critical temperature is 374.1 °C.

1.2.8 Refrigerants

The working medium circulating in a refrigeration machine or heat pump is referred to as the refrigerant. The basic principles of the refrigeration process have so far only been explained on the example of water, and water genuinely possesses many of the properties required of a refrigerant. Water is cheap, plentiful, nontoxic, non-flammable and has a relatively large latent heat of vaporization/lique-faction. It is, therefore, not surprising that water is used as the refrigerant in the steam jet and absorption type refrigeration machines or heat pumps described in the following.
Water is not suitable as a refrigerant in the compression cycle, because the pressures and temperatures at which changes of state occur are unfavorable. Instead, refrigerants are used in this process that are more volatile than water, i.e. substances that evaporate at relatively low temperatures and correspondingly high pressures.

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CFCs in particular are depleting the ozone layer because their high chemical stability, allows them to remain in the atmosphere for a long time, so theoretically, the entire quantity that is released diffuses to the stratosphere and decays there. The chlorine released in the process destroys the ozone layer that protects against harmful UV radiation. Therefore, only chlorine-free refrigerants are permitted today. Additional information: see chapter 2.

1.2.9 Summary

Substances with low boiling points **(refrigerants** in the refrigeration field) are referred to as gases, because they evaporate immediately at normal temperatures and pressures (our ambient conditions). This already points towards a critical situation with regard to the transport, storage and use of refrigerants.
In the refrigeration cycle, the refrigerant takes on the following states:

- Liquid
- Gaseous

and transfers:

- Latent heat
- Sensible heat

In a thermodynamic cycle, heat is extracted from the medium to be cooled via **vaporization of a refrigerant.**
This process occurs either **directly** via the refrigeration machine or **indirectly** via an intermediate cold-water circulation.
A cooling system is a machine that extracts heat from a substance or a room. The refrigeration capacity is the ability to dissipate a given quantity of heat in a given period of time.

$$\dot{Q}_o = \dot{m} \cdot c \cdot \Delta t$$

$\dot{Q}_o$ = refrigeration capacity [kJ/h or kW]
$\dot{m}$ = mass flow [kg/s]
c = specific heat [kJ/kgK]
Δt = temperature differential [K]

The specific heat c is the energy (in kJ) required to heat 1 kg of a given substance by 1 K. The specific heat depends on the state of aggregation. The following table shows some values for substances typically used in refrigeration:

Substance	c_{solid}	c_{liquid}	c_{gas}
Water	2.03 kJ/kgK	4.18 kJ/kgK	2.05 kJ/kgK
Ammonia NH_3		4.44 kJ/kgK	2.18 kJ/kgK
Refrigerant R22		1.09 kJ/kgK	0.16 kJ/kgK

A distinction is made between sensible heat and latent (nonsensible) heat.

Sensible heat:
Heat with which a change in energy content always involves a temperature change.

Latent heat:
Heat that must be absorbed or dissipated in order to bring about a change of state. It does not give rise to a temperature change. Instead, the energy is required to change the molecular structure of the substance. Latent heat depends greatly on temperature and pressure.

The heat content is the enthalpy h.

- The enthalpy of the liquid is denoted by h'
- The enthalpy of the vapor is denoted by h''
- The enthalpy required for vaporization is denoted by r
- The unit of measurement is kJ/kg in each case

Different substances have different temperature behaviors, i.e. the temperature values of the melting and boiling points differ.

Melting and boiling points depend on pressure.

E.g. water at 1 bar: Melting point = 0 °C and boiling point = 100 °C.

This dependency can be seen on the pressure-enthalpy diagram. Transferring the values from the water vapor table to an h-log P diagram provides a graphical presentation of the relationships.
The refrigerants used in refrigeration can achieve boiling points as low as –120 °C.

In the air conditioning field, boiling points of around + 2 to –20 ºC are used.

2. Refrigerants

2.1 Properties, history

Excerpt from a patent specification for refrigerants from the year 1834:

Patent specification

… " What is claimed herein is a patent for a device that enables a volatile medium to be used to cool and freeze another medium but that at the same time allows the volatile medium to be condensed and reintroduced into the cycle without losses."…

In principle, any substance can be used as a refrigerant, because only the following is important:

"The boiling and liquefaction points must be at technically achievable pressures".

However, the demands on an ideal refrigerant are so great that only very few substances are genuinely suitable.

Physical properties
- Change of state from liquid to gas at "low" pressures
- Large latent heat of vaporization

Chemical properties
- Resistant
- Nonexplosive
- Noncombustible
- Nontoxic
- Noncaustic
- Noncorrosive
- Easy to determine by smell or simple to measure in low concentrations
- Chemically stable, i.e. must not decay to its atomic components even in extreme situations

From a physical and chemical point of view, water would be an ideal refrigerant. However, its use is inconceivable for vaporization temperatures (t_o) above + 4 °C or below 0 °C.
Three main refrigerants have been known since the beginnings of refrigeration:
SO_2 sulphur dioxide
CH_3Cl chloromethane
NH_3 ammonia

These refrigerants fulfil the main criteria expressed in the patent. However, they leave much to be desired with regard to aggressiveness, combustibility and hygiene; for example:

Toxicity:
SO_2 0.5 – 1 vol. %, deadly in 5 minutes
NH_3 0.5 – 1 vol. %, deadly in 60 minutes
CH_3Cl 2 – 2.5 vol. %, permanent damage in 120 minutes

Due to the CFC ban, the refrigerant NH_3 (ammonia) is today again increasingly used in air conditioning in spite of the toxic hazard, although certain safety measures must be implemented.

2.2 Refrigerant naming   A refrigerant should have as low as possible a boiling point at at mospheric pressure, it should have a low vapor volume, and its condensation pressure should be easy to manage by technical means. Additionally, it must not attack the materials and lubricants of the refrigeration system, and it should be as nontoxic, noncombustible and nonexplosive as possible.

In order to provide these properties, manufacturers such as DuPont have developed refrigerants by derivation from methane CH_4 and ethane C_2H_6.

The derivative molecules contain the following substances (with the chemical symbols):

Carbon   "C"
Fluorine   "F"
Chlorine   "Cl"
Hydrogen   "H"

Names

Name	Vendor	Location
Freon	DuPont	USA
Frigen	Höchst	Frankfurt, Germany
Arcton	ICI	United Kingdom
Kaltron	Kali Chemi	Hannover, Germany

The most common refrigerants are referred to by an abbreviation with the letter **R** and a **number.** The names are the company names for these derivatives.
Occasionally, the word "refrigerant" and/or the trade name (e.g. **F**reon) is used instead of the letter **R.**

R and a maximum 3-digit number
Number of hundreds   +1   = number of C atoms (carbon)
Number of tens   −1   = number of H atoms (hydrogen)
Number of units   = number of F atoms (fluorine)

Whereby the zero in the hundreds position is omitted, so methane derivatives (e.g. R22) have a two-digit number. The hydrogen (H) atoms are replaced by the halogens Cl (chlorine), F (fluorine), Br (bromine).

Examples: R114   chemical formula $C_2Cl_2F_4$ tetrafluorodichloroethane CFC
                    $\Rightarrow$ 4 fluorine atoms
            −1 $\Rightarrow$ no hydrogen atoms and, therefore, 2 chlorine atoms
            +1 $\Rightarrow$ 2 carbon atoms

R134a   chemical formula $C_2H_2F_4$ tetrafluoroethane HFC
                    $\Rightarrow$ 4 fluorine atoms
            −1 $\Rightarrow$ 2 hydrogen atoms
            +1 $\Rightarrow$ 2 carbon atoms

R22   chemical formula $CHClF_2$ difluoromonochloromethane HCFC
                    $\Rightarrow$ 2 fluorine atoms
            −1 $\Rightarrow$ 1 hydrogen atom and, therefore, 1 chlorine atom
            +1 $\Rightarrow$ 1 carbon atom

The demands on a refrigerant could be extended to include any number of requirements, such as heat transmission or fluid flow properties.

2.3 Physical properties

Oil as a lubricant in the refrigeration cycle

In compression refrigeration systems, a suitable oil must be used for compressor lubrication. The specific properties of the various refrigerants require the use of appropriate machine oils.

The oil is required for compressor lubrication and is (unintentionally) entrained in the refrigeration cycle. Due to the fact that chemical reactions, such as corrosion and acid formation, occur at high temperatures, there is an additional hazard because the compression temperatures are usually above 100 °C.

For this reason, refrigeration machine oil must meet certain requirements.

Three groups of oils have proved themselves in the refrigeration field:

- Mineral oils   e.g. for R12, R12B1
- Semisynthetic oils   e.g. for R22, R23
- Synthetic oils   e.g. for R134a, R23

Neither mineral oils nor the synthetic alkyl benzenes used until now are suitable for use with the new, chlorine-free refrigerants. Instead, only synthetic ester-based oils, also known as ester oils, should be used.

Miscibility

The miscibility of oil and refrigerant has both advantages and disadvantages:

Advantages:
- System components can be well lubricated
- The oil can be returned to the compressor from the system relatively well

Disadvantages:
- Dilution and heating of the oil in the compressor
- Change of flowability on the evaporator side (cold) and, therefore, problems with oil return. Oil residues in the evaporator give rise to poor heat transmission and to control problems

Behavior in case of system leaks

There are many factors that determine the behavior of a refrigerant in case of leaks. Pressure, viscosity and density are only a few of them.
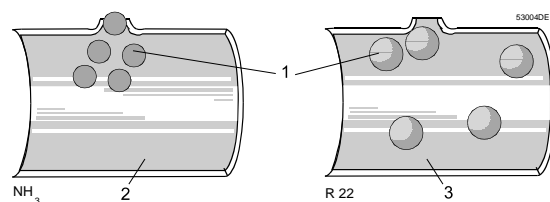
Fig. 2-1   Different behavior of refrigerants in case of leaks

1 Molecule
2 Molar mass = 17.03 g/mol
3 Molar mass = 86.48 g/mol

The greater the molar mass in kg/kmol (formerly molecular weight M), the less the tendency of the refrigerant to escape in case of leaks.

E.g.   R717   (NH_3, ammonia)   17.03 kg/kmol
       R407C   (23 % R32, 25 % R125, 52 % R134a)   86.20 kg/kmol
       R134a   ($C_2H_2F_4$)   102.03 kg/kmol

Odor A slight odor can be an advantage in some refrigerants, because it allows any leaks to be noticed and corrected immediately before the entire contents are lost or damage occurs to system components; examples:
- The odor of R22 is   "slightly ethereal"
- ammonia   "pungent odor"

Fig. 2-2   Odor

Toxicity Refrigerants that are harmful to human beings are generally referred to as "toxic refrigerants". All refrigerants can naturally pose an asphyxiation hazard if they occur in such large amounts that the oxygen in the air is displaced; however, some are already harmful in very small amounts. The degree of harmfulness depends on the concentration, refrigerant type and duration of exposure to the refrigerant.

Fig. 2-3   Toxicity, R-717 ammonia (NH_3)

Combustibility Refrigerants differ greatly with regard to combustibility.
Some, such as R-170 (ethane) or R290 (propane), burn so easily and completely that they are used as fuels. Other refrigerants are non-combustible, e.g. R22 and R134a.

Fig. 2-4   Combustibility; R-170 ethane, R-178 water

2.4 Currently used refrigerants
2.4.1 Ban on CFCs and HCFCs

Effect on the ozone layer
The emission of CFCs into the atmosphere demonstrably causes environmental damage. Although the amount still used as refrigerants in a closed cycle only accounts for 10% of the total CFC production (the major part is used in aerosol cans and insulating foams as well as in solvents), environmental protection is still an important issue.

Leakage
Approximately 3,000 t p.a. in Europe; 10 % is lost during transport, dispensing and filling.

The environmental damage caused by CFC (chlorofluorocarbon) refrigerants is mainly due to their ozone depleting and global warming potential (greenhouse effect).

CFCs in particular are ozone destroyers, because their high chemical stability allows them to remain in the atmosphere for a long time, so theoretically, the entire quantity that is released diffuses to the stratosphere and decays there. The chlorine released in the process depletes the ozone layer that protects against harmful UV radiation.

This process has a lasting detrimental effect on the sensitive balance between oxygen and ozone. The long dwell time and the associated depot effect in the atmosphere are also responsible for the high global warming potential of such compounds.

	Components	Life	ODP	GWP
	CFC 11	60 years	1	1
	CFC 12	120 years	1	3.1
CFCs	CFC 113	90 years	0.8	1.4
	CFC 114	200 years	0.7	4
	CFC 115	400 years	0.4	7
	HCFC 22	15 years	0.05	0.34
	HCFC 123	2 years	0.02	0.02
HCFCs	HCFC 124	7 years	0.02	0.1
	MP 39 (R401)		0.03	0.21
	HP 80 (R402)		0.02	0.63
	HFC 134a	16 years	0	0.26
	HFC 152a	2 years	0	0.03
	HFC 32	7 years	0	0.12
HFCs	HFC 125	41 years	0	0.84
	HFC 23	310 years	0	6
	HP 62		0	0.94
	AC 9000		0	0.28

Table 2-1 Environmental data of various refrigerants

ODP: Ozone depletion potential
GWP: Global warming potential, greenhouse effect with respect to CFC 11 = 1
CFC: Chlorofluorocarbons, fully halogenated (no water atom in the molecule)
          These are: R11, R12, R13, R113, R114, R115 (R500 and R502 are
                    mixtures containing R12 and R115 respectively)
HCFC: Hydrochlorofluorocarbons, partially halogenated (one or more water atoms in
          the molecule)
          These are: R22, R123
HFC: Hydrofluorocarbons, partially halogenated (molecule contains not only fluorine
          but also water atoms).
          These are: R134a, R227

HCFCs and **HFCs** are partially halogenated compounds with considerably shorter atmospheric dwell times. The ozone depletion potential of HCFCs is only a fraction of that of CFCs.
HFCs, such as R134a, are chlorine-free and have zero ozone depletion potential.
The global warming effect of both types of compounds is also low.

Preventive measures   CFC refrigerants have been no longer available in Europe since January 1, 1995 and globally since January 1, 1996.

The HCFC refrigerant R22 has been banned for use in new systems in Europe since January 1, 2000 at the latest. The global use of R22 and the use of other HCFC refrigerants are scheduled to be phased out in stages, i.e. 35 % reduction from January 1, 2004 and 50 % reduction by 2020.

2.4.2 Substitutes for CFC and HCFC refrigerants   For typical R12 applications, the environmentally compatible alternative R134a has proved itself in practice.

2.4.2.1 R134a as an alternative to R12

Tempe-rature	Pressure	Specific volume of the liquid	Specific volume of the vapor	Enthalpy of the liquid	Enthalpy of the vapor	Latent heat of vapo-rization
t	p	v'	v"	h'	h"	r
°C	bar	l/kg	l/kg	kJ/kg	kJ/kg	kJ/kg
– 50	0.299	0.692	596.88	137.72	366.32	228.60
– 45	0.396	0.699	459.14	143.48	369.55	226.08
– 40	0.516	0.706	357.66	149.34	372.78	223.44
– 35	0.666	0.713	281.87	155.32	375.99	220.67
– 30	0.848	0.720	224.55	161.40	379.18	217.78
– 25	1.067	0.728	180.67	167.59	382.34	214.75
– 20	1.330	0.736	146.71	173.88	385.48	211.59
– 15	1.642	0.744	120.15	180.28	388.57	208.29
– 10	2.008	0.753	99.17	186.76	391.62	204.85
– 5	2.435	0.762	82.45	193.34	394.62	201.28
0	2.929	0.772	69.01	200.00	397.56	197.56
5	3.497	0.782	58.11	206.74	400.44	193.70
10	4.146	0.793	49.22	213.57	403.26	189.69
15	4.883	0.804	41.89	220.46	406.00	185.54
20	5.716	0.816	35.83	227.44	408.66	181.23
25	6.651	0.828	30.77	234.48	411.24	176.76
30	7.698	0.842	26.52	241.61	413.71	172.11
35	8.865	0.856	22.94	248.81	416.08	167.27
40	10.160	0.871	19.89	256.11	418.33	162.23
45	11.592	0.888	17.29	263.50	420.45	156.94
50	13.171	0.907	15.05	271.02	422.41	151.39
55	14.907	0.927	13.12	278.69	424.19	145.51
60	16.811	0.949	11.44	286.53	425.76	139.24
65	18.894	0.974	9.97	294.59	427.09	132.49
70	21.170	1.003	8.68	302.95	428.10	125.15
75	23.651	1.036	7.53	311.68	428.71	117.03
80	26.353	1.076	6.50	320.93	428.81	107.87
85	29.292	1.127	5.56	330.91	428.17	97.26
90	32.487	1.194	4.68	342.02	426.40	84.38
95	35.958	1.298	3.83	355.20	422.55	67.36
100	39.728	1.544	2.80	374.97	411.79	36.83

Table 2-2 Vapor table of R134a
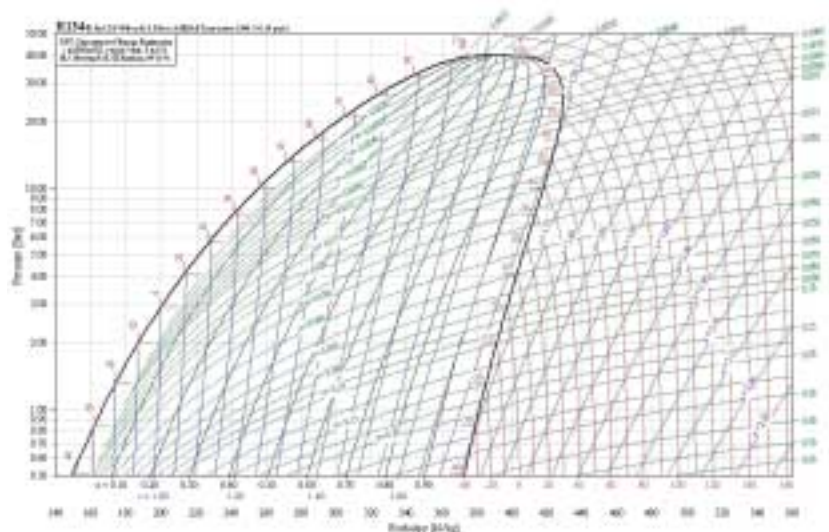
Fig. 2-5   h-log P diagram for R134a

2.4.2.2 R407C and R290 as alternatives to R22 and R502 respectively

For typical R22 applications as well as for R502 applications where R22 has until now been used as a substitute, zeotropic 2- or 3-component mixtures are primarily used, whereby as small as possible a temperature glide (see zeotropes) is targeted.

Azeotropes

These mixtures have a new, common boiling point that is generally in between the boiling points of the individual components.

Zeotropes

Mixtures of two or more components. The unique feature of zeotropes is that each component retains its own boiling point, so evaporation and condensation occur at a gliding temperature. The temperature difference between the start and end of the respective process is referred to as the "temperature glide". This means that different components of the mixture escape in case of leaks and that such refrigerant mixtures have to be charged in liquid form.

A frequently used substitute for R22 is R407C, which is a 3-component mixture with the following composition:

– 23 % R32
– 25 % R125
– 52 % R134a

The various manufacturers market R407C under different names, some of which no longer have a direct reference to the refrigerant name:

DuPont	AC 9000
Hoechst	Reclin 407C (formerly HX3)
ICI	KLEA 66

When using R407C, note that the filling quantity is only about 90 % of that of R22 due to its different density. The coefficient of performance is reduced by approximately 3 – 5 %.

The refrigerant R290 (propane) is used in heat pump systems in particular. Due to its combustibility and the risk of explosions, however, it is only used up to a given heat pump size.

3. The h-log p diagram

3.1 General The thermodynamic processes of refrigeration are quite complex. They can only be calculated with considerable effort in terms of formulae and tables.

Vapor tables Vapor tables show the specific variables of the medium, such as specific volume v, density ρ, enthalpy h and entropy s for the boiling and saturated medium as a function of boiling temperature t or boiling pressure p in tabular form.

Example (see table 3-1):

At a boiling temperature t = 0 °C, the refrigerant is subject to a pressure p of 2.929 bar. This is the boiling pressure for the corresponding temperature t. Its volume in the boiling state is 0.772 l/kg (dm³/kg) of refrigerant. The density ρ, which is the reciprocal of the specific volume (ρ = 1/v), is 1.29 kg/l (kg/dm³) at this point. The relative enthalpy h' was defined at 200 kJ/kg (t = 0 °C is the reference point of this refrigerant for the h-log p diagram under discussion).
The relative heat content of the saturated refrigerant vapor at the same pressure is given as enthalpy h". It is 397.56 kJ/kg for the selected example. The difference h" – h' determines the absolute latent heat of vaporization or condensation r at a given pressure p.
In general, the values with one dash, e.g. v', h', s', etc., refer to the boiling state of the medium, and the values with two dashes (v", h", r", etc.) refer to the saturated gaseous state.

Tempe-rature	Pressure	Specific volume of the liquid	Specific volume of the vapor Dampfes	Enthalpy of the liquid	Enthalpy of the vapor	Latent heat of the vapo-rization
t, ϑ	p	v'	v"	h'	h"	r
°C	bar	l/kg	l/kg	kJ/kg	kJ/kg	kJ/kg
– 50	0.299	0.692	596.88	137.72	366.32	228.60
– 45	0.396	0.699	459.14	143.48	369.55	226.08
– 40	0.516	0.706	357.66	149.34	372.78	223.44
– 35	0.666	0.713	281.87	155.32	375.99	220.67
– 30	0.848	0.720	224.55	161.40	379.18	217.78
– 25	1.067	0.728	180.67	167.59	382.34	214.75
– 20	1.330	0.736	146.71	173.88	385.48	211.59
– 15	1.642	0.744	120.15	180.28	388.57	208.29
– 10	2.008	0.753	99.17	186.76	391.62	204.85
– 5	2.435	0.762	82.45	193.34	394.62	201.28
0	2.929	0.772	69.01	200.00	397.56	197.56
5	3.497	0.782	58.11	206.74	400.44	193.70
10	4.146	0.793	49.22	213.57	403.26	189.69
15	4.883	0.804	41.89	220.46	406.00	185.54
20	5.716	0.816	35.83	227.44	408.66	181.23
25	6.651	0.828	30.77	234.48	411.24	176.76
30	7.698	0.842	26.52	241.61	413.71	172.11
35	8.865	0.856	22.94	248.81	416.08	167.27
40	10.160	0.871	19.89	256.11	418.33	162.23
45	11.592	0.888	17.29	263.50	420.45	156.94
50	13.171	0.907	15.05	271.02	422.41	151.39
55	14.907	0.927	13.12	278.69	424.19	145.51
60	16.811	0.949	11.44	286.53	425.76	139.24
65	18.894	0.974	9.97	294.59	427.09	132.49
70	21.170	1.003	8.68	302.95	428.10	125.15
75	23.651	1.036	7.53	311.68	428.71	117.03
80	26.353	1.076	6.50	320.93	428.81	107.87
85	29.292	1.127	5.56	330.91	428.17	97.26
90	32.487	1.194	4.68	342.02	426.40	84.38
95	35.958	1.298	3.83	355.20	422.55	67.36
100	39.728	1.544	2.80	374.97	411.79	36.83

Table 3-1 Excerpt from a vapor table for R134a

German engineer Richard Mollier (1863–1935) developed a state diagram. It enables refrigeration specialists to present important physical variables of refrigeration media and the corresponding processes in graphical form. Heat quantities, work and pressure differences appear as measurable distances, which considerably simplifies the calculation of process variables and, therefore, the sizing of refrigeration equipment components. Because of these features, enthalpy-pressure diagrams have become widely used in refrigeration. The application of the diagram to the refrigeration cycle of a compression refrigeration machine will be explained in detail separately.

Enthalpy-pressure diagram

In general terms, the enthalpy-pressure diagram shows the states of aggregation of a substance as a function of heat supply and pressure. Enthalpy means the same as heat content and is designated with the symbol h, and since the pressure p is usually shown on a logarithmic scale, the term *h-log p diagram* is used among specialists.
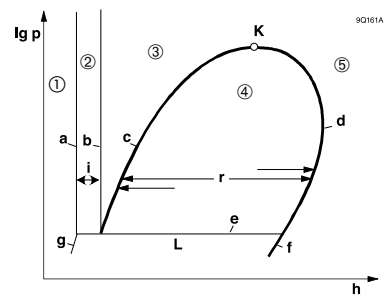

Fig. 3-1   Schematic view of the h-log p diagram

    1 Solid-state zone

    2 Fused-state zone

    3 Zone of subcooled liquid

    4 Wet vapor zone

    5 Zone of superheated vapor

    a Melting line

    b Solidification line

    c Boiling line

    d Saturated vapor line

    e Triple line

    f Desublimation line

    g Sublimation line

    i Latent heat of fusion/solidification

    r Latent heat of vaporization/condensation

    K Critical point

    L Solid/gas mixture ("smoke")

The diagram shows the following state zones:

• Solid-state zone	one phase:	solid
• Fused-state zone	two phases:	solid and liquid
• Zone of subcooled liquid	one phase:	liquid
• Wet vapor zone	two phases:	liquid and gas
• Zone of superheated vapor	one phase:	gas
• Sublimation zone	two phases:	solid and gas

3.1.1 State zones and terminology

Melting line a and latent heat of fusion i:
A given heat quantity i is required to melt a solid body (enthalpy of fusion). When the melting temperature is reached, heat can be continuously supplied to the body without the temperature rising until the whole body has assumed the liquid state. The term latent heat is used if no temperature change occurs although heat is being supplied. Melting line a connects the melting points. They are determined by pressure p and heat content h.

Solidification line b and latent heat of solidification i:
Inversely, if heat is removed from the liquid, the temperature remains constant until the entire body has solidified. The extracted latent heat of solidification corresponds to the latent heat of fusion i. Solidification line b connects the solidification points.

Zone of subcooled liquid and boiling line c:
When a liquid is being heated, it is referred to as subcooled until the boiling point is reached. This point depends on the boiling temperature and boiling pressure.
The boiling points are connected by the boiling line c.

Wet vapor zone and latent heat of vaporization r:
In order to transform a liquid into a gaseous state, additional heat must be supplied. If the boiling point is reached and the pressure remains constant, heat can be continuously supplied without the temperature rising until all of the liquid has evaporated. The supplied heat causes the liquid to evaporate. The mixture of boiling liquid and vapor that is present during the vaporization process is referred to as wet vapor. The heat required to generate 1 kg of vapor at a given pressure is referred to as the specific enthalpy of vaporization (latent heat of vaporization) r.

Saturated vapor line d and superheated vapor:
When the liquid has completely evaporated, the vapor is dry and saturated. For each saturation pressure there is a given saturation temperature; together they form the saturated vapor line d. If saturated vapor is further heated, its temperature rises rapidly. It is then referred to as superheated vapor or hot-gas.

Latent heat of condensation:
If heat is removed from the superheated vapor by cooling, its temperature falls.
It first becomes saturated vapor, then it condenses without a change in temperature. The same amount of heat must be removed from the vapor again in order for it to change into the liquid sate (condensation, liquefaction). The specific latent heat of vaporization is equal to the latent heat of condensation r at a given pressure.

Critical point K:
The specific latent heat of vaporization r decreases continuously with increasing pressure (and temperature) until it finally disappears completely at a given state of the vapor. The point at which there is no longer a difference between the boiling liquid and the saturated vapor is referred to as the critical point. It separates the boiling line from the saturated vapor line. Above the critical point, there is no longer a difference between liquid and vapor. The medium is in the critical state.

Sublimation line g and desublimation line f:
At temperatures below the melting point, solid bodies can also directly assume the gaseous state (sublimation) or vice versa (desublimation). Example, evaporation of a snow crystal: It can be observed in winter that, at temperatures below 0 °C and with dry air, thin layers of ice disappear in a few days even without solar radiation. Desublimation can be observed when humid air forms hoarfrost at temperatures below 0 °C without prior condensation. Snow crystals are also formed by desublimation.

3.1.2 Diagram range for refrigerants

A suitable range of the Mollier chart has been selected for actual refrigeration practice. It contains only the liquid and gaseous state types and their mixed forms.
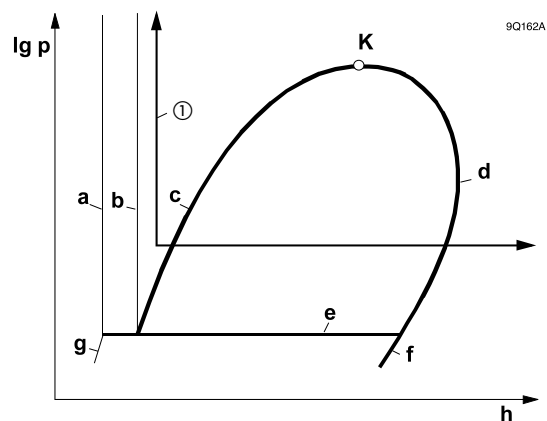

Fig. 3-2   Diagram range for refrigerants ?
   a Melting line
   b Solidification line
   c Boiling line
   d Saturated vapor line
   e Triple line
   f Desublimation line
   g Sublimation line
   K Critical point

3.2 Diagram structure

The state variables and state changes will be developed and described step by step taking as an example the refrigerant R134a, which is frequently used in practice.

3.2.1 Coordinates h, p

The *heat content (enthalpy h)* is plotted on a linear scale on the *horizontal axis,* the abscissa. The enthalpy expresses the amount of heat (in kJ) that is contained per kilogram of a substance. Since the diagram range is selected according to the problem in hand, it is not the absolute enthalpy but the relative enthalpy. Therefore, the enthalpy always refers to a given point, e.g. boiling liquid at 0 °C. The origin of the coordinate system is normally selected as a round whole number, such as 0, 100, 200 kJ/kg. The numerical range of the scale changes according to the refrigerant. The vertical lines are the isenthalps, i.e. lines of equal enthalpy.
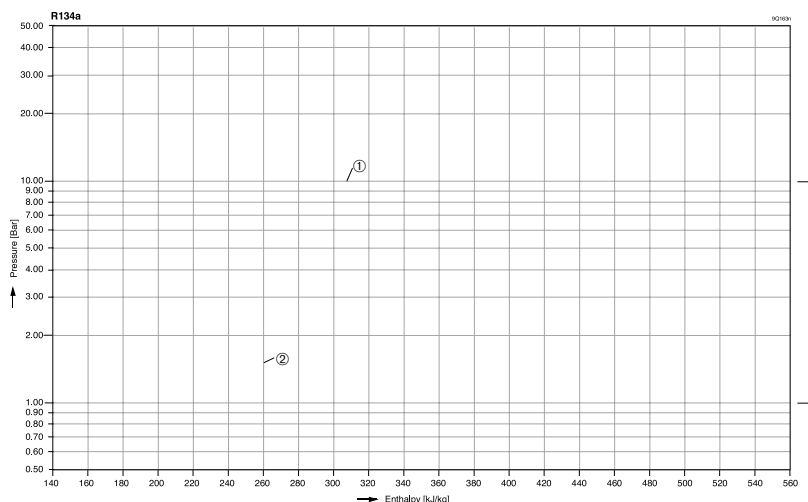
Fig. 3-3   Basic grid
        1 Isobars
        2 Isenthalps

The *pressure p* is plotted on the *vertical axis,* the ordinate. In order to make the most used areas of the charts more straightforward, the pressure is shown on a logarithmic scale. Logarithmic means that the same distance is drawn between the powers. The horizontal lines are the isobars, i.e. lines of equal pressure.

The h-log p diagram allows the large number of values from the vapor table to be presented in a simple manner. For each pressure p, the corresponding enthalpy of the boiling liquid h' and of the saturated vapor h" is transferred from the vapor table to the diagram.
Example: p = 2.929 bar, enthalpy of the boiling liquid h' = 200 kJ/kg, enthalpy of the saturated vapor h" = 397.56 kJ/kg. The latent heat of vaporization h" – h' can be measured directly on the diagram as distance r.
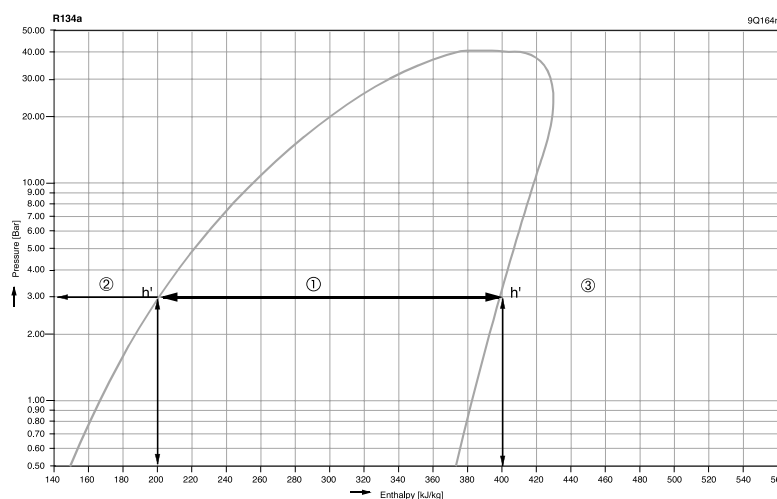

Fig. 3-4   Boiling line and saturated vapor line
        1 Liquid
        2 Wet vapor
        3 hot-gas

The values of h' and h" form the respective limit curves for the boiling line to the left of the critical point K and the saturated vapor line to the right. The limit curves clearly separate the liquid, wet vapor and hot-gas zones from each other. It can also be seen that the latent heat of vaporization r decreases with rising pressure until it disappears completely at the critical point K.

3.2.2 Ratio lines x

Wet vapor is a mixture of boiling liquid and saturated vapor. The ratio lines **x** allow the proportions of gas and liquid in the wet vapour zone to be determined. **x** expresses the proportion of gas, 1-**x** the proportion of liquid. Therefore, along the line **x** = 0.4, there is a mixture of 40 % gas and 60 % boiling refrigerant. **x** is not included in tables, because it is a series of ratio numbers only.
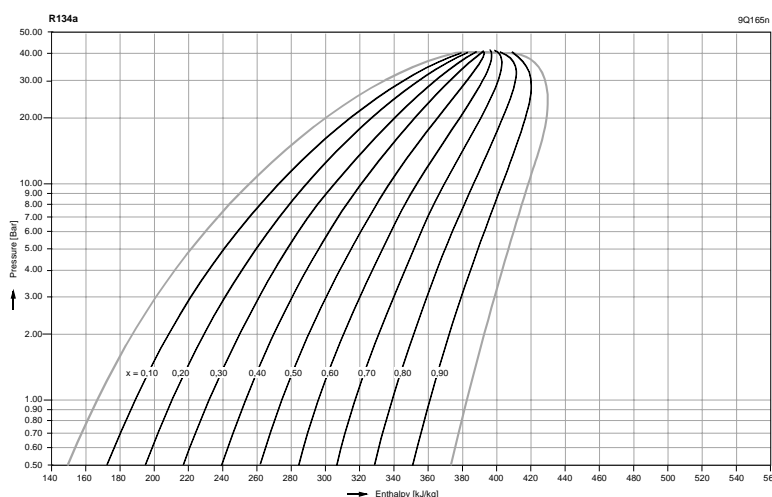

Fig. 3-5   Ratio lines **x** = constant

3.2.3 Isotherms t

Isotherms are lines of equal temperature. Their progression changes abruptly on a change of state. In real refrigerant diagrams, the isotherms are not drawn in the liquid and wet vapor zones for reasons of clarity. However, their progression is indicated by small angles on the boiling line.

In the wet vapor zone, the isotherms are horizontal, parallel to the isobars. Supplied heat is used exclusively to vaporize the medium (latent heat).

Example: At pressure p = 2.929 bar, boiling refrigerant at 0 °C
                    is converted to saturated vapor at 0 °C
                    by heat supply r = 197.56 kJ/kg.

In the hot-gas zone, the isotherms drop steeply. A small quantity of heat is sufficient to rapidly raise the temperature here (sensible heat). Therefore, the isotherms are noticeably closer together in this zone. Rising temperature intersects the isotherms from left to right. If saturated vapor at 0 °C is superheated by supplying 80 kJ/kg of heat, the temperature rises to approximately 80 °C.
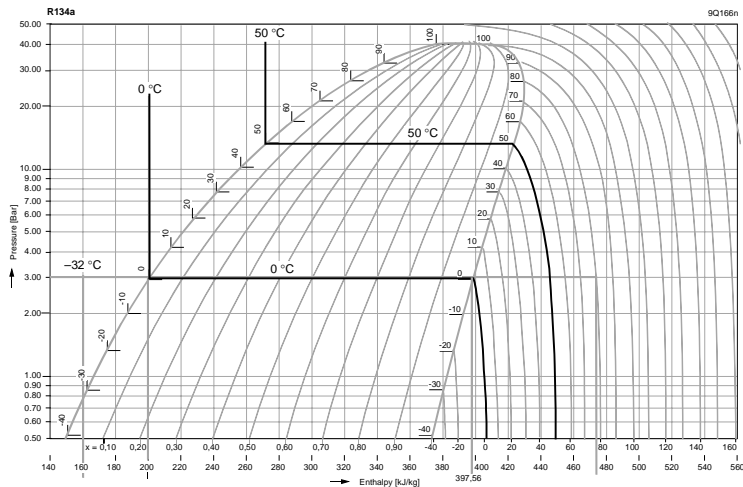
Fig. 3-6   Lines of equal temperature (isotherms, t = constant)

In the liquid zone, the isotherms fall increasingly steeply with increasing distance from the critical point. This again means that a change in sensible heat content has an immediate effect on temperature.

Example: If 40 kJ/kg of heat is extracted from a refrigerant at 0 °C, it cools to −32 °C.

The critical temperature no longer intersects the wet vapor zone. It simply contacts the critical point. Isotherms with supercritical temperatures run through the zone of superheated gas only.

3.2.4 Specific volume v   A further important variable in the diagram is the specific volume v. It is usually indicated in m^3/kg, and it shows the actual volume of one kilogram mass at a given pressure p. The vapor table shows two very different specific volumes, i.e. that of boiling liquid v' and that of saturated vapor v". The degree to which these values depend on the state of aggregation can be seen by comparison in the vapor table for R134a, Table 3-1.

At a pressure p of 2.929 bar and a temperature of 0 °C, v' = 0.772 dm^3/kg, (= 0.000772 m^3/kg, point A), whereas saturated vapor assumes a volume of 0.0691 m^3/kg (point B) at the same pressure. Therefore, the volume of the vapor is approximately 89 times greater. Because the volume per kg of gas is considerably greater than that of liquid at certain pressures, the unit of v" is frequently changed from dm^3/kg to m^3/kg (1,000 dm^3 = 1 m^3).
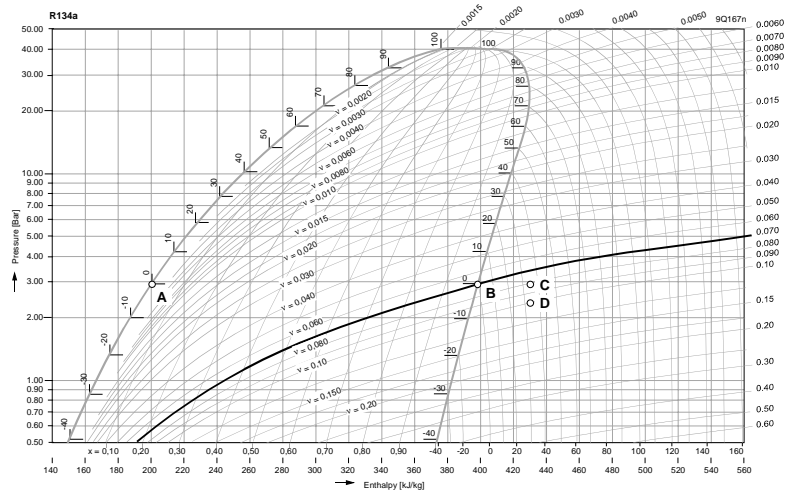
Fig. 3-7 Lines of equal specific volume (isochores, v = constant)

The specific volume is highly significant in refrigeration. In general terms, the specific volume of refrigerant should be as small as possible in order to transport as great a quantity of refrigerant $\dot{m}$ as possible with a small volume. The relationship between the refrigerant quantity $\dot{m}$ and the specific volume v is as follows:

$$\dot{m} = \frac{\dot{V}}{v}$$

$\dot{V}$:   volumetric flow in m³/h

$\dot{m}$:   refrigerant quantity in kg/h

v: specific volume in m³/kg

Let us assume that a compressor with constant delivery volume $\dot{V}$ transports a certain refrigerant quantity $\dot{m}$ at a given pressure p_1. A change of pressure to p_2 would have a lasting effect on the specific volume v and, therefore, on the transported refrigerant quantity $\dot{m}$.

Example: v_1 = 0.08 m³/kg, p_1 = 2.929 bar (point C)
                v_2 = 0.10 m³/kg, p_2 = 2.5 bar (point D)

The transported refrigerant quantity $\dot{m}$ falls by 20 %.

3.2.5 Isentropes s   In the wet vapor range and in the zone of superheated vapor there are *isentropes.* These are *lines of equal entropy s.* The increase in entropy is a measure of the heat losses occurring in technical processes. It is a caloric variable like enthalpy and has a specific value in each state. The absolute value of entropy is indefinite. It is calculated from a random point – usually the standard state (0 °C in the case of R134a).
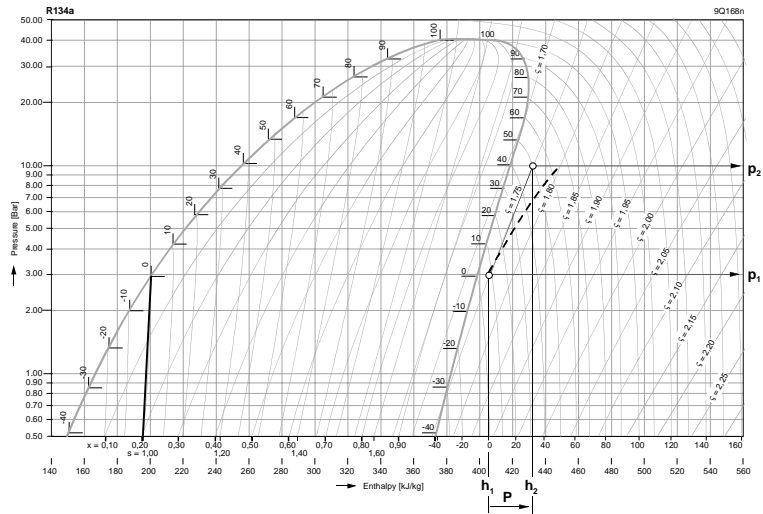
The unit of entropy s is $\dfrac{kJ}{kgK}$

Fig. 3-8   Lines of equal entropy (isentropes, s = constant)

The isentropes are of particular importance to the refrigeration specialist in the zone of superheated vapor. The compression in an ideal, i.e. loss-free, compressor follows a line of equal entropy. Therefore, the isentropes make it possible to determine the ideal (theoretical) compression work P per kg of refrigerant via a comparison of the initial and ultimate enthalpies (after compression). This in turn gives an indication of the actual power demand of a compressor.

Example: A compressor draws in refrigerant in the state h_1/p_1 and compresses it to h_2/p_2, where the technical work P corresponds to the enthalpy difference $h_2 - h_1$. A real compression process would involve an increase in entropy (losses). Therefore, the work curve would be to the right of point h_2/p_2 (dashed line ——).

3.3 Summary   The structure of the h-log p diagram is now complete. It allows the states of the refrigerant in its various phases to be accurately described. Six variables are available for this purpose:

Description	Symbol	Dimension
1. Pressure	p	bar
2. Enthalpy	h	kJ/kg
3. Density	ρ	kg/dm³
4. Specific volume	v	dm³/kg
5. Liquid-vapor ratio	x	* 100 in %
6. Temperature	t	°C
7. Entropy	s	kJ/kg K

The h-log p diagram allows refrigeration processes in particular to be presented simply with the operating points of the refrigeration machine, and it allows the effect of control actions on the machine to be shown clearly.

4. Design of the compression refrigerating machine

4.1 Introduction

In refrigeration and air conditioning today, the **compression refrigeration process** is applied in more than 90 % of all cases. The major characteristic of this process is the use of refrigerants that extract as great as possible a latent heat of vaporization from the medium to be cooled at the vaporization temperature t_o, that evaporate in the process and that can then be liquefied again under manageable pressures at the condensation temperature t_c.

The compressor provides the work output and refrigerant transport. The process is referred to as the **cold-vapor refrigeration process.**

A different type of refrigeration is possible with the **absorption refrigeration process.** In this process, the refrigerant is absorbed by a **solvent** in the absorber, and expelled again with the supply of heat in the generator (expeller). The operating principle of the absorption refrigeration machine is explained in chapter 9.

Fig. 4-1   Possibilities of heat extraction; air, water, evaporation

4.2 Compressor

Purpose of compression
The purpose of the compressor is to extract the gaseous refrigerant at low pressure and temperature from the evaporator and to compress it in order to bring it to a higher temperature and pressure at which it can be liquefied again.

Compressor types
A distinction is made between the following types of compressors:

Piston compressors
- Open
- Semihermetic
- Fully hermetic

Rotary piston compressors
- Screw compressor
- Scroll compressor
- Rotary blade piston compressor
- Wankel compressor
- Turbocompressor

4.2.1 Piston compressors
The most frequently used compressor in air conditioning is the piston compressor.
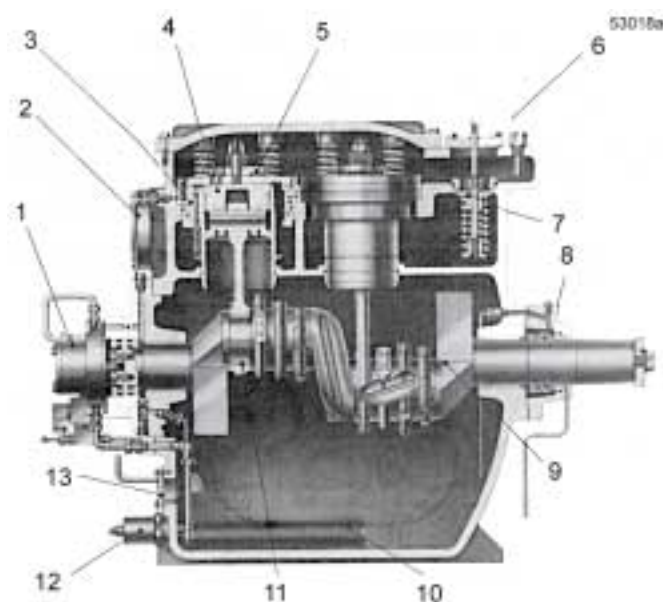

Fig. 4-2   Section – open plunger compressor, two-cylinder, series D, manufacturer: Linde AG
1 Gear oil pump
2 Suction side
3 Cylinder liner
4 Piston
5 Working valves
6 Discharge side
7 Overflow valve
8 Bearing ring seal
9 Crankshaft
10 Oil filter
11 Connecting rod
12 Crankcase heater
13 Oil sight-glass

Operating principle of the piston compressor cylinder

The compressor comprises a cylinder with valves and a piston that is moved back and forth by a motor. As the cylinder volume increases, the upper valve closes. When the valve lifts, the refrigerant is drawn in through the port. The piston compresses the gas by reducing the cylinder volume, and presses it into the upper chamber. During this process, the refrigerant is also transported through the refrigerant cycle (see Fig. 4-3 and Fig. 4-4).

Fig. 4-3   Section of the
cylinder head

Fig. 4-4   Schematic view of the operating
principle

1 Inlet valve
2 Gaskets
3 Cylinder head
4 Valve reed
5 Outlet valve
6 Noxious space
7 Piston
8 Cylinder wall
9 Cylinder

The compression process is easily explained by analogy to the pumping up of a bicycle tyre:

As the air is pumped into the tyre, it is compressed and forced into the tyre via the valve.

The lower end of the pump heats up, and the tyre pressure increases.

The work required for this process is provided by a human being.

**A pressure and temperature increase is achieved via mechanical work.
The power provided by the compressor has to be paid for.**

4.2.1.1 Open compressors

The distinction between open, hermetic and semihermetic compressors has nothing to do with the type of piston or anything of that sort. Rather, this expression applies to the connection between the compressor and drive.
The open compressor has a closed housing that is under refrigerant pressure. The drive is provided from outside the housing via the drive shaft.
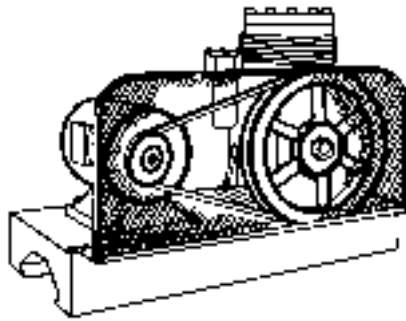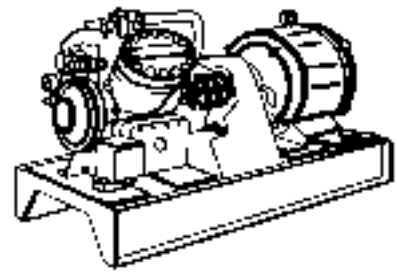
Fig. 4-5   Open compressor with belt drive   Fig. 4-6   Open compressor with direct drive

Drives:	Any possible prime mover (electric, gas motors, etc.), either directly on the drive shaft or via a belt drive.
Advantage:	Motor easy to exchange, better reparability.
Disadvantage:	Sealing of the shaft gland, friction and drive losses.
Use:	Suitable for 0 – 1 MW power range; less frequently used in air conditioning.

4.2.1.2 Semihermetic compressors

In the semihermetic compressor, the electric drive motor is enclosed in the same housing as the compressor.
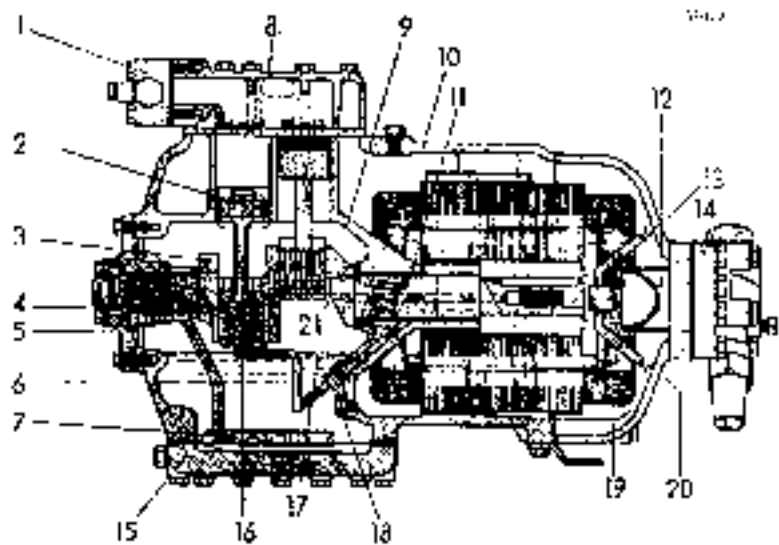
Fig. 4-7   Semihermetic compressor (R22 motor compressor, Carrier)

1	Pressure shutoff valve	12	Suction filter
2	Piston	13	Gas pressure equalization
3	Crankshaft	14	Suction shutoff valve
4	Oil pump	15	Oil sump connection
5	Bearing	16	Oil pressure regulating valve
6	Oil suction pipe	17	Oil level
7	Oil filter	18	Oil pressure return valve
8	Cylinder head	19	Motor
9	Crankcase	20	Pressure equalization pipe
10	Motor housing	21	Gas inlet
11	Main bearing		

The housing is flanged together with the drive motor and the compressor (encapsulated). The motor is specially insulated for operation in the refrigerant and is cooled by the (cold) refrigerant vapor. Since the cooling effect becomes greater with increasing refrigeration output, the drive motors can be sized on the small side.

Advantage: In spite of the encapsulated design, the motor can be easily repaired. The disadvantages of shaft sealing do not apply.

Disadvantages: Sensitive to contamination; there must be no humidity or dirt in the refrigeration system. The entire refrigeration system must be **evacuated prior to commissioning.**

Use: Suitable for 3 – 500 kW power range; wide range of applications in air conditioning.

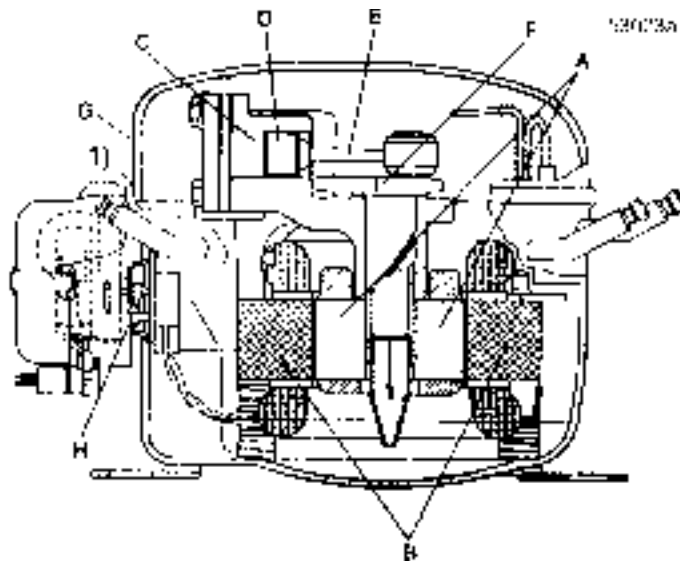**4.2.1.3 Hermetic compressors** The motor and compressor are enclosed in a tightly sealed, welded housing.

Fig. 4-8   Hermetic compressor
1) Housing, welded
A Rotor
B Stator
C Cylinder
D Piston
E Connecting rod
F Slider crank
G Encapsulating housing
H Electrical connections

The name already describes the design: it is a fully encapsulated device that is no longer accessible from the outside.

Advantage:	Low-cost due to compact design. Compressors of this type are often completely assembled with evaporator and condenser at the manufacturer's facility. Protection against contamination, high-volume production; the high precision of production provides for long service life.
Disadvantage:	Repair, oil check and valve exchange not possible.
Use:	Lower power range; domestic refrigerators, small air conditioning units.

Important!
Sufficient cooling and lubrication of the moving parts must be provided in all compressor types. Cooling is usually provided by the cold refrigerant vapor present in the cycle or by cooling fins and ambient air (except in case of open compressors). Lubrication is usually provided by an appropriate oil, although appropriate steps must be taken to fulfil the criterion of oil return in the refrigeration cycle.

4.2.1.4 Power Compressor power $\dot{Q}_o$ is determined by the refrigerant mass flow $\dot{m}_k$ delivered by the compressor and by the increase in the specific enthalpy difference $h_1 - h_3$ of the refrigerant in the evaporator. h_3 is the enthalpy of the liquid refrigerant upstream of the expansion valve; h_1 is the enthalpy of the vaporized refrigerant at the evaporator outlet.

$$\dot{Q}_o = \dot{m}_k \cdot (h_1 - h_3)$$

The main factors influencing the compressor's power are as follows:

a. Mechanical design
– Working volume; function of bore, stroke and number of cylinders
– Dead space; residual space between piston and cylinder
– Intake and discharge valves; shape and size are design factors

b. Application
– Rotor speed
– Intake pressure
– Compression pressure
– Refrigerant

4.2.2 Screw compressors **Two different types are distinguished:**

a. Single-screw compressor
A grooved roller is driven. Compression is achieved via one or more gears engaging in the grooves.

b. Twin-screw compressor
Compression is achieved by intermeshing rollers, where the driven roller has helical projections, and the other helical grooves. The different number of starts (normally drive = 4, grooves = 6) gives rise to compression pockets that move in axial direction.

Advantages with respect to piston compressors:
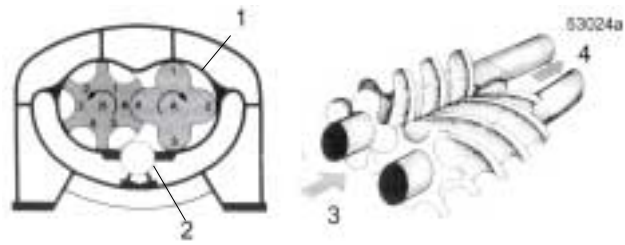Rotary movement only, no valves, infinitely variable speed control.

Functioning principle:

Fig. 4-9   Screw compressor
     1 Cylinder
     2 Control element
     3 Intake
     4 Discharge

4.2.3 Scroll compressors

The scroll compressor, like the screw compressor or rotary piston compressor, belongs to the group of rotary compressors.

Compression is achieved by two concentric spirals, one stationary and one rotating. This movement draws in gas and compresses it. The compressed gas is discharged in the center (Fig. 4-10, H). Since the spiral rotates continuously, the compressor always contains refrigerant at different pressures. Therefore, the process shown below is not step-by-step but virtually continuous (Fig. 4-10, A – H).
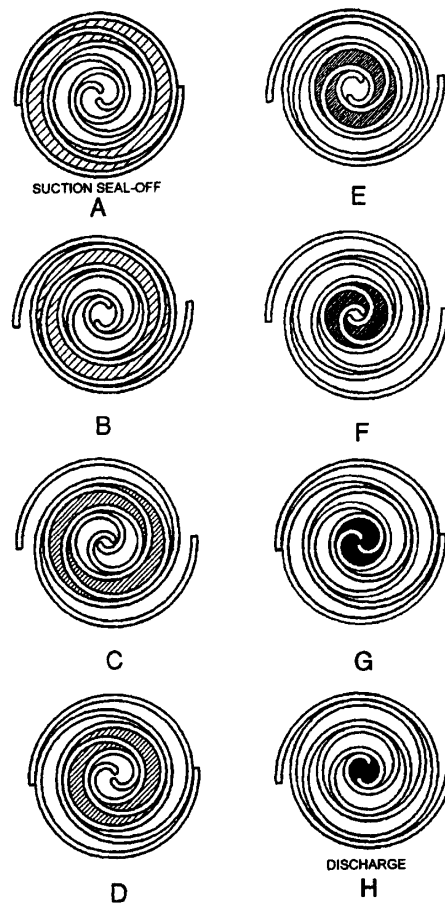

Fig. 4-10 Principle of compression with one
   rotating spiral and one stationary spiral

Fig. 4-11 Typical scroll com-
          pressor with vertical
          configuration

The scroll compressor is probably the most advanced compressor design currently available and is likely to successively replace the piston compressor in the low-power range (1 – 50 kW). Rotary compressors are suitable for continuous speed control via frequency converters, which provides for even more efficient energy use.

4.2.4 Rotating piston compressors

Compression is produced by a piston that is mounted eccentrically on the shaft and rolls around the inner wall of a circular cylinder.
Used in **hermetic compressors** only.
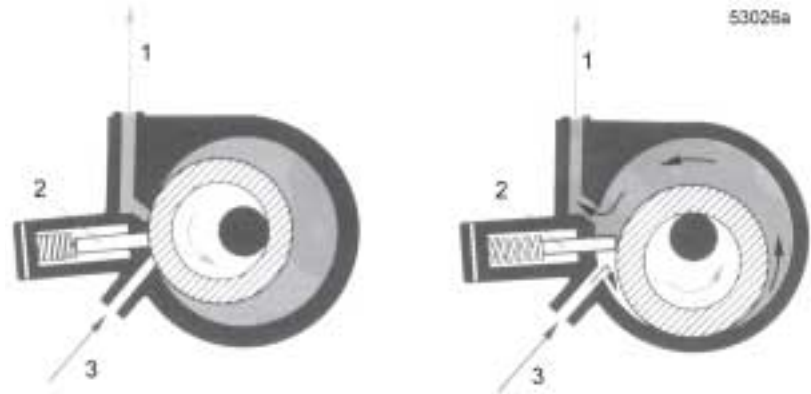
Beginning of compression stroke; intake and discharge closed.

Intake of refrigerant vapor and compression.

Fig. 4-12 Rotating piston compressor
    1 Discharge pipe
    2 Seal
    3 Intake pipe

4.2.5 Turbocompressors

Compression is achieved by accelerating the gas flow in the rotor and then converting the kinetic flow energy into a pressure increase in the diffuser. The design and operating behavior are similar to those of a radial fan.
Due to the high volume, turbocompressors are mainly found in large-scale refrigeration applications.

Advantages: Dynamic operating characteristics, low wear, good controllability.
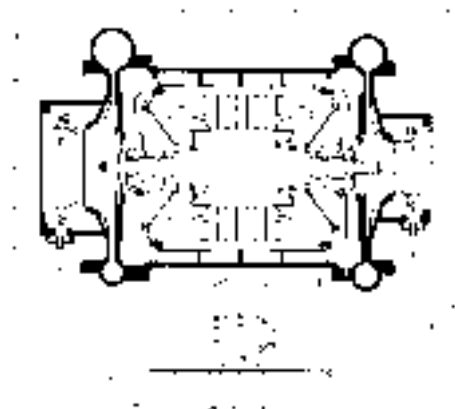

Fig. 4-13 Turbocompressor
    Motor turbocompressor with direct drive (Trane. Type: Cen Tra Vac). The rotor
    of the electric motor (6) and the two rotor disks (37 = first stage, 18 = second
    stage) are mounted on a common shaft (33) supported by two antifriction

bearings (12) with axial thrust collars. The axial thrust is equalized by the opposing orientation of the rotor disks.

The two bearings are protected against oil leakage by caps (5) and sealing rings (17). Forced oil supply from the oil pump (21) via pipe (3), drain via pipe (32) to the oil reservoir. Motor housing flanged together with spiral housings (10 and 31), sealed via labyrinth seal (13). Motor cooled by refrigerant vapor which is drawn by the fans (9) from the evaporator via the duct (8) and through the motor housing and then returned to the evaporator via connecting pipes (not shown). In case of motor cooling with water, circulation only in the enclosed casing cavity. Guide vanes (14) on both stages.

Refrigeration output in the air conditioning field 0.8 to 2.5 MW.

Power ranges of compression and absorption machines for use in the refrigeration, air conditioning and heat pump fields.

Compression and absorption machines for use in refrigeration, air conditioning and heat pump systems	Power ranges at t_0 ø 0 °C
Fully hermetic compressors (usually of reciprocating piston design, but also rolling or rotary piston compressors, including special designs)	> 0 – 50,000 W
Semihermetic piston compressors	> 0 – 300,000 W
Open piston compressors	> 0 – 1 MW
Screw compressors	0.2 – 5 MW
Absorption machines (with indirect heating: steam or hot water)	0.35 – 6 MW
Hermetic turbocompressors	0.35 – 6 MW
Open turbocompressors	0.35 – 30 MW

Fig. 4-14 Power ranges of compression and absorption machines

4.3 Condensers

The compression process in the refrigeration cycle increases the pressure and temperature of the refrigerant.

Advantage of the increased pressure level:

The refrigerant has a higher vaporization temperature. If the condensation process is considered, it is found that the refrigerant can condense at high pressure via the extraction of heat.

Purpose

Heat extraction of the energy supplied for evaporation and supplied by compression of the refrigerant (waste heat).

Heat transfer to a heating system (heat pump) or heat recovery of the energy supplied for evaporation and supplied by compression of the refrigerant.

The object in both cases is always to remove as much heat as possible from the vaporized refrigerant and to return it from the gaseous to the liquid state.

Condenser types:
- Water-cooled condensers
- Air-cooled condensers
- Evaporative condensers

4.3.1 Water-cooled condensers

Functioning principle: the most common water-cooled condensers in air conditioning are tube bundle or shell and tube condensers.
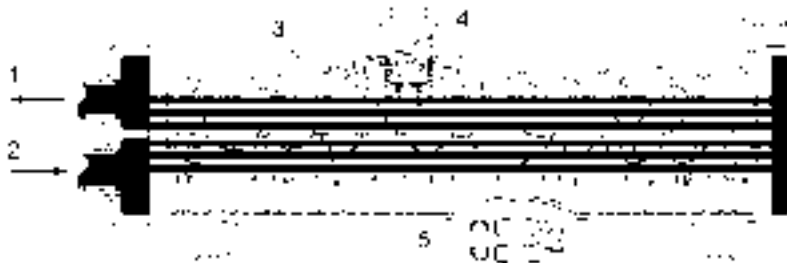

Fig. 4-15 Section through a tube bundle condenser
1 Coolant outlet
2 Coolant inlet
3 Internal tubes
4 Inlet of superheated refrigerant vapor
5 Liquid pipe connection

The functioning of such designs is very similar: The cold coolant flows through the internal tubes while the hot refrigerant cools and then condenses in the shell cavity or in the refrigerant pipes.

Coolants:
- Municipal water supply; expensive, uneconomical
- Well or river water; not permitted everywhere
- Sea water; salt content
- Closed-cycle systems; heat recovery

4.3.2 Air-cooled condensers

In air-cooled condensers, the refrigerant flows though a pipe where it is cooled by the ambient air and changes its state of aggregation.
If the ambient air is not circulated by mechanical means, the term **statically ventilated condenser** is used, e.g. rear side of a domestic refrigerator.
Used for low cooling powers.
In air conditioning, however, **finned-tube condensers** are used which can be installed directly in an air duct for heat recovery purposes or are frequently equipped with a fan and installed on building roofs.
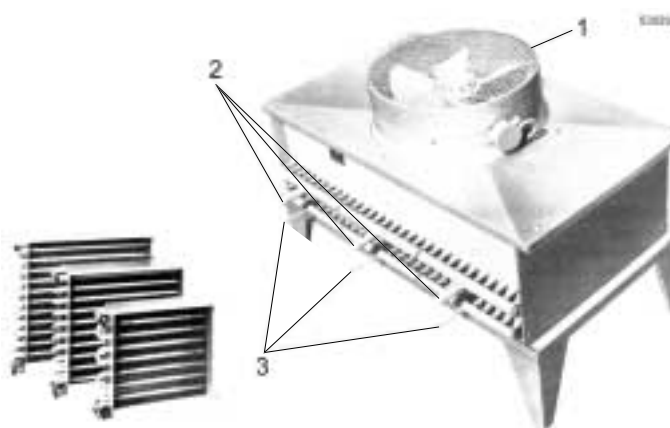

Fig. 4-16 Air-cooled condensers
1 Axial fan
2 Pressure pipe connection
3 Connection for refrigerant collector

Advantages over water-cooled condensers:
- Maintenance-free, easy to clean
- Water too expensive, aggressive, impure
- No risk of icing

4.3.3 Evaporative condensers

In evaporative condensers, in addition to air-cooling, a water flow is usually circulated and directed over the surface of the cooling coils containing the refrigerant.
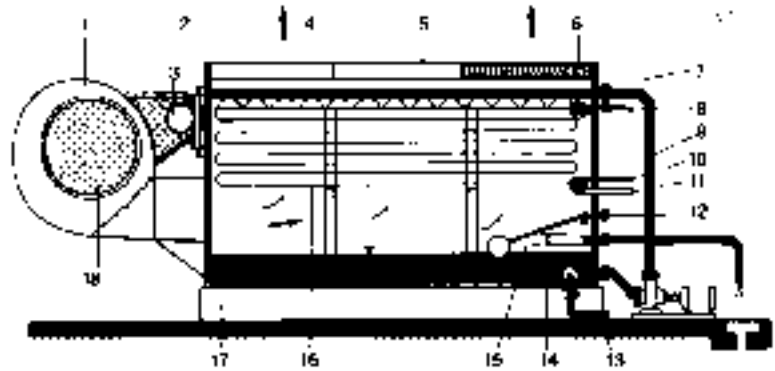

Fig. 4-17 Evaporative condenser

1 Radial fan	10 Pressure equalization connection
2 Belt guard	11 Refrigerant outlet
3 Electric motor	12 Float valve
4 Air outlet	13 Bleed
5 Spray nozzles	14 Intake strainer
6 Eliminator plate	15 Automatic clarification unit
7 Spray equipment	16 Condenser coils
8 Refrigerant inlet	17 Base
9 Butterfly valve	18 Intake grille

In this cycle, only the water flow required to wet the condenser pipes is circulated by pump. The evaporated water is replenished via a level regulator.

Because of the frost hazard, the system must be emptied and the condenser operated with air-cooling only in winter.

Condenser power

In order for a refrigeration plant to operate correctly, the condensation temperature and pressure must be kept within given limits.

High condensation pressures give rise to increased power demand at the compressor, which can overload the drive motor. For this reason, the high-pressure side is protected by appropriate limiters and safety devices.

Low condensation pressures give rise to an excessively low pressure of the liquid refrigerant at the expansion valve, which can cause irregular and unstable supply to the evaporator, thus affecting refrigeration capacity.

The capacity of a refrigeration plant increases if the condensation pressure falls within permissible limits = within the working range of the expansion valve. As with a valve in the water cycle, the output of the expansion valve depends on the pressure drop across the valve.

4.4 Expansion

Now that the refrigerant is available again in liquid form at the condenser outlet, it can be returned to the evaporator. This takes place via the **throttling device.**

Purpose of expansion

In the refrigeration process, the throttling device which is referred to as the regulator or usually as the **expansion valve,** has the purpose of bringing the liquid refrigerant from a higher temperature and pressure to a lower temperature and pressure. The refrigerant is expanded. An additional task of the expansion valve is to supply only as much refrigerant to the evaporator as the system can vaporize in the current operating state.

Types of expansion valves

- Capillary tube
- Manual throttling device
- Automatic throttling device
- Thermostatic throttling device
- Electronic throttling device

Operating principle of expansion valves

Expansion begins immediately after the narrowest point in the throttle valve, e.g. at the valve seat. During this process, part of the refrigerant begins to evaporate and extracts the enthalpy of vaporization from the liquid component. Therefore, the refrigerant temperature drops to the vaporization temperature (pressure) t_0 without heat being given off to the environment – h = constant.

The functioning of the thermostatic expansion valve is explained in the following, because this type of valve is used most frequently in ventilation and air conditioning.

4.4.1 Thermostatic throttling valves

The valves are not used to regulate the evaporator pressure but the superheating temperature downstream from the evaporator.
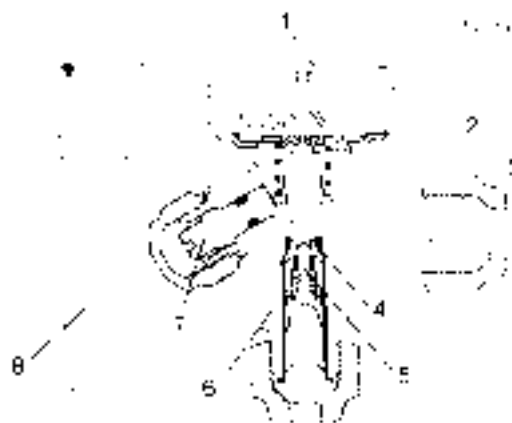The choice of valve is determined by the type of refrigerant used.

Fig. 4-18 Main components of a thermostatic valve
1 Capillary tube
2 Regulator
3 Regulator spring
4 Seat
5 Valve plug
6 Body
7 Adjusting screw
8 Temperature sensor

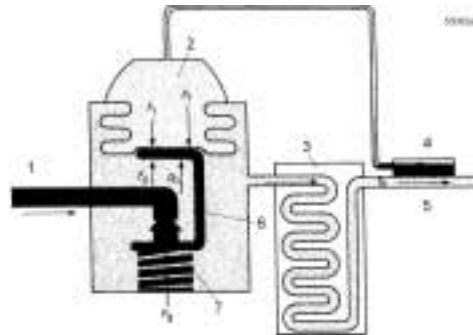The functioning of the valve is determined by the evaporator pressure p_0 and the sensor pressure p_1.

Fig. 4-19 Schematic view of a thermostatic valve
1 Liquid pipe
2 Space 2
3 Evaporator
4 Temperature sensor
5 Suction pipe
6 Pressure pin
7 Regulator spring

Sensor pressure p_1 prevails in space 2. Force F, from the sensor pressure and diaphragm surface area, always acts as the opening force. The pressure pin transfers the movement of the diaphragm to the valve plug.
The evaporator pressure p_0 and force F_0 prevail in the space below the valve and act in closing direction. The regulator spring can be used to determine the exact differential between sensor and vaporization pressure at which the valve begins to open. This ratio also determines the charging of the evaporator.
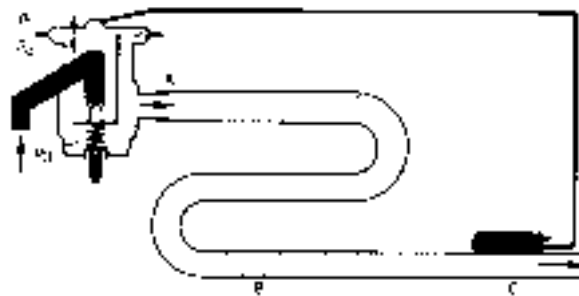
Fig. 4-20 Pressure ratios and evaporator charge $p_1 = p_0 + p_3$

The refrigerant inlet is at point A. The refrigerant should have evaporated by point B in order to be superheated between B and C.
A higher temperature at point C than point B is set using the regulator spring. Therefore, the operating superheating is preselected using the adjusting screw.

The lower the degree of superheating, the more complete the charging of the evaporator surface area will be. Point B moves to the right.

The degree of superheating cannot be selected arbitrarily. It depends on the evaporator type, on the temperature differential between the medium to be cooled and the evaporation temperature and on the design of the throttle valve. Additionally, superheating protects the compressor against incompletely evaporated refrigerant.

4.4.2 Electronic expansion valves

Unlike the thermostatic expansion valve, the electronic expansion valve is a control system that requires auxiliary energy and normally consists of the following components:

- The actual superheating controller
- The electronic control element – the throttle valve
- Sensors (temperature sensor, pressure sensor)

Control loop

In contrast to the thermostatic expansion valve, which can be considered as a P-controller, the electronic expansion valve control loop behaves like a PID control loop with the corresponding advantages.
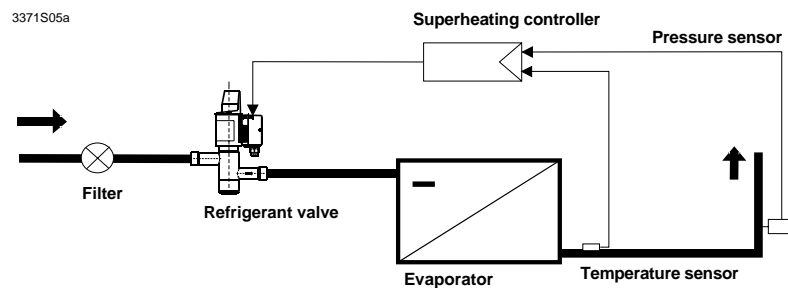
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Superheating controller

Pressure sensor

Filter

Refrigerant valve

Evaporator

Temperature sensor

Fig. 4-21 Control loop with electronic expansion valve

Function

The superheating controller checks (calculates) the differential between the suction gas temperature and the calculated vaporization temperature (TOH-TO), compares it with the setpoint ΔT and controls the electronic injection valve accordingly (modulating control mode). Setpoint ΔT is adjustable between 4 and 10 K.

The refrigeration valve is a self-contained valve that is robust, maintenance-free and hermetically sealed. Due to precise position control, the valve has a high resolution. The electronic interface is designed for an operating voltage of AC 24 V and an actuating signal of DC 0…10 V or DC 4…20 mA and delivers a DC 0…10 V position feedback signal. The valve is closed in the deenergized state.

4.5 Evaporators

Depending on the viewpoint or on the application of the refrigeration machine, the main purpose is either fulfilled by the evaporator (heat extraction) or by the condenser (heat transfer). In heat recovery systems, both units are equally incorporated in the process.

Purpose of evaporation

The evaporator has the task of extracting heat from a medium that is to be cooled. Therefore, the temperature in the evaporator must always be lower than that of the medium to be cooled.

The purpose of the evaporator is to absorb a quantity of heat and to transfer it to the refrigerant it contains. The refrigerant evaporates in the process.

Evaporators for cooling liquids
Evaporators for cooling air
Direct expansion cooling
Indirect expansion cooling

Operating principle of evaporators

Direct expansion cooling
In the case of direct expansion cooling, the evaporating refrigerant extracts heat directly from the medium to be cooled.
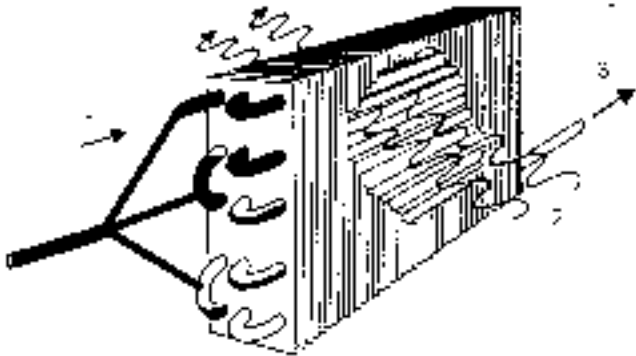

Fig. 4-22 Direct expansion evaporator
1 Liquid, evaporator inlet
2 Air to be cooled
3 Gas, evaporator outlet

Indirect expansion cooling
The only difference to direct expansion cooling is the fact that there is a second medium (usually water or a water-glycol mixture) between the evaporator and the medium to be cooled.

Direct expansion cooling	Indirect expansion cooling
• Lower procurement costs	• Simple planning, installation and operation
• Higher refrigerant temperatures	• Simple control at the consumers
• Smaller refrigeration compressors	• Better partial load operation
• Lower energy costs	• Leaks are less critical
• Use only appropriate in combination with individual evaporators or with a small number of evaporators	• No oil return problem
	• Refrigeration machines and accessories are located in the central machinery room, which is favorable in terms of maintenance
	• Both cooling and heating possible
	• Best solution in highly distributed systems

Table 4-1 Advantages and disadvantages of direct and indirect expansion cooling

4.5.1 Tube bundle evaporators

The most frequently used type is the tube bundle evaporator for liquid cooling. In this type of evaporator, the liquid refrigerant evaporates in the evaporator tubes, and the coolant flows around them.

Dry evaporator   This type of evaporation is also referred to as **dry evaporation.** Evaporator powers can be between 4 and 3,500 kW. Such evaporators are usually used for producing chilled water in large-scale, highly distributed ventilation systems. The controlled variable for the refrigeration cycle is the chilled water temperature, which is usually 6/12 °C, where Δt = 6 K. The small Δt indicates that air coolers have large surface areas.
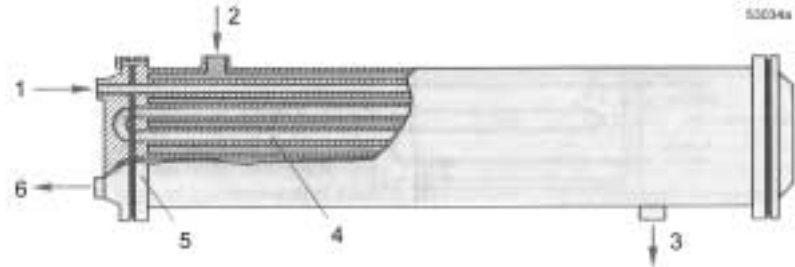
Fig. 4-23 Section of a tube bundle evaporator
     1 Liquid refrigerant in
     2 Coolant inlet
     3 Coolant outlet
     4 Refrigerant tubes
     5 Tubesheet
     6 Gaseous refrigerant out

The design is only slightly different from that of a **tube bundle condenser.**
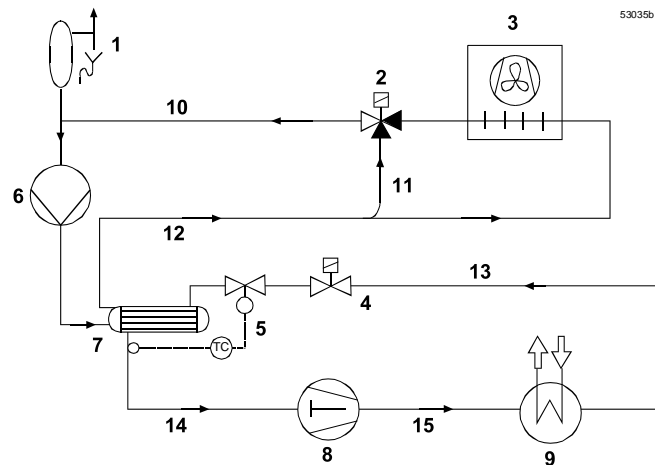
Fig. 4-24 Schematic of a chilled water cooling system in the air conditioning field
     1 Expansion vessel   9 Condenser
     2 Three-port valve   10 Chilled water return pipe
     3 Air cooler   11 Bypass pipe
     4 Magnetic valve   12 Chilled water supply pipe
     5 Thermostatic control valve   13 Liquid pipe
     6 Chilled water circulating pump 14 Suction pipe
     7 Tube bundle evaporator   15 Pressure pipe
     8 Piston compressor

4.5.2 Plate evaporator (plate heat exchanger)

Plate heat exchangers are compact devices made of stainless steel plates. They are used in the entire process engineering and heat-producing industry. Their main advantages with respect to other heat exchanger types are as follows:

- High heat transfer output with small volume, i.e. high degree of heat recovery and minimal heat losses
- Low liquid content, therefore short heating and cooling times; highly dynamic with regard to temperature changes in control processes
- Modular design, therefore easy and low-cost adaptation to changed output requirements as well as easy maintenance
- Low risk of fouling and simple chemical cleaning due to highly turbulent fluid flow and the absence of dead space; simple disassembly
- The embossing of the plates creates a highly turbulent fluid flow. This provides for very effective heat transfer even at low flow volumes

Fig. 4-25 Different types of plate heat exchangers and profiled plates

Design

Components of a plate heat exchanger:

- The plate package consists of a defined number of individually profiled plates (Fig. 4-25)
- Plate heat exchangers for use in refrigeration systems must be refrigerant-resistant. In order to prevent the refrigerant from escaping, the individual plates are soldered together on the refrigerant side. Normal seals are usually used on the water side
- The media involved in the heat exchange enter and leave the exchanger via the connections

Functioning and flow routing

Adjacent profiled plates with admission ports form a package of flow gaps. The two media involved in the heat exchange flow through alternate gaps.

One-way heat exchangers are normally used. They are characterized by 100 % counterflow of the two media. The supply and return pipes are connected to the fixed plate.

Multiple-way plate heat exchangers also achieve 100 % counterflow of the two media if the number of ways is equal for each. This offers the advantage that the temperature differential of the two media can be fully utilized.

4.5.3 Air-cooling evaporators   Air-cooled evaporators (and condensers) are of similar design. The mechanical design and configuration of the tube rows are important for good heat exchange.

The liquid refrigerant is supplied to the evaporator via a distributor (spider) so that it flows evenly over the evaporator's surface area.
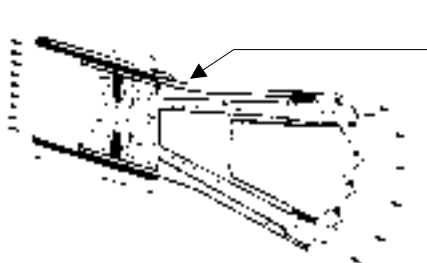

Fig. 4-26 Distributor nozzle   Fig. 4-27 Distribution at the evaporator
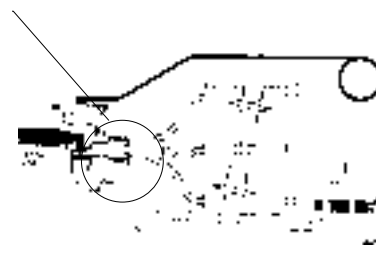
$p_1 - p_3$ = overall pressure drop in the distributor
E   = impact and stagnation area with momentary flow deflection
F   = testrictor
G   = greatest reduction of flow area
H   = turbulence zone due to uncontrolled expansion

Most evaporators used in air conditioning are for the application range $T_0 > 0$ °C and air to be cooled > 0 °C. These values are frequently below 0 °C in food refrigeration. The problem with operating points around 0 °C is the possibility of **moisture condensation,** i.e. appropriate steps must be taken to prevent moisture condensation on the evaporator surface or to defrost it.

The most frequently used defrosting methods are as follows:
• Defrosting via fan overrun
• Defrosting with electric heating (mainly in cold storage rooms)
• Defrosting with hot refrigerant vapor

4.5.4 Ice banks   The **ice bank** is a different type of refrigerant cooling. These are frequently used where short-term loads occur or to bridge evaporator standstill times, i.e. they are used wherever **refrigeration reserves** are required.
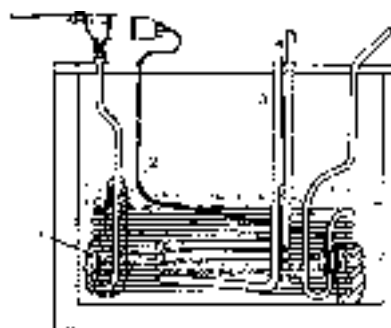
Fig. 4-28 Section of an ice-water tank
        1 Refrigerant coil
        2 Water level
        3 Ice reserve

The evaporator is installed in a large water-filled container. The evaporation process causes a layer of ice to form on the surface, which keeps the water cold longer.

Additional explanations and examples will be given in the "Cold stores" chapter.

4.6 Safety in the refrigeration cycle

As you will by now have realized, the refrigeration cycle is a critical thing. Why?

- The medium can be liquid and gaseous
- Different pressures prevail in the system
- The medium changes its state of aggregation at different pressures
- It must be ensured that the medium has defined states at specific points in the cycle
- Sufficient cooling and lubrication of the equipment must be ensured

In order to ensure that these conditions are met, the refrigeration cycle contains additional components:

- Safety valves
- Shutoff valves
- Overpressure safety switches
- Underpressure safety switches
- Differential pressure pressostats
- Low/high-pressure pressostats
- Dryers
- Collectors
- Inspection windows
- Superheaters
- Device to limit the switching frequency (e.g. max. 6 times per hour)
- Maximal operating pressure (MOP) circuit

These components and functions are only of special interest to the refrigeration system contractor, so they are only mentioned in this chapter.

Fig. 4-29 Possibilities of refrigeration control

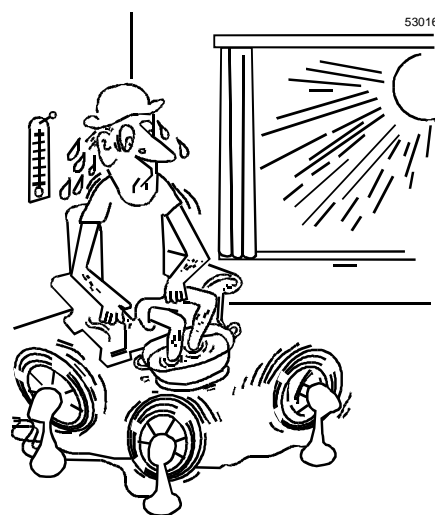# 5. The compression refrigeration cycle on the h-log p diagram

5.1 General

With over 90 % of all installed systems, the compression refrigeration machine has become the most important. Special attention is paid to the presentation of the respective process states on the h-log p diagram. The possibility of suction gas control has also been incorporated into the explanations. Further possibilities for refrigeration machine output control are covered in chapter 6 ("Continuous output control in the refrigeration cycle").

5.1.1 Components and their functions

A refrigeration machine consists of four major elements:

Evaporator (Fig. 5-2, 1): In the evaporator, liquid refrigerant evaporates at low pressure and temperature. The heat required for evaporation is extracted from the medium to be cooled (air, water).

Compressor (Fig. 5-2, 2): The compressor draws the refrigerant vapor out of the evaporator via the suction pipe. The gas is compressed and forced into the condenser via the pressure pipe. The refrigerant leaves the compressor at high pressure and in a highly superheated state (approx. 60 – 120 °C) but still as a gas.

Condenser (Fig. 5-2, 3): In the condenser, the hot-gas transfers its heat to the coolant (air, water, etc.) and condenses. The latent heat of vaporization and the compressor work that has been converted to heat must be dissipated here. The liquid refrigerant is then usually routed to a liquid collector.

Expansion valve (Fig. 5-2, 4): The purpose of the expansion valve is to reduce the high pressure of the refrigerant liquid to the low pressure prevailing in the evaporator. Additionally, the expansion valve also controls the charging of the evaporator under the various load conditions. This is done by controlling the superheating of the gas at the evaporator outlet.
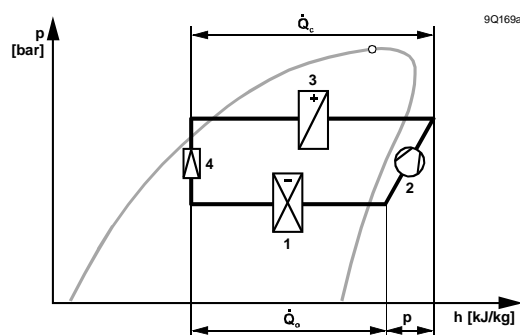

Fig. 5-1   Refrigeration cycle with the major components on the h-log p diagram
   1 Evaporator
   2 Compressor
   3 Condenser
   4 Expansion valve

5.1.2 The refrigerant and its operating states

The refrigerant is the actual heat transfer medium. It transports heat within the refrigeration cycle. In principle, any substance that can be liquefied and vaporized at technically achievable pressures and at the desired temperatures can be used as a refrigerant. The fact, however, that only few substances have been successfully introduced and become established as refrigerants is due to the multitude of different requirements regarding physical and chemical properties that are determined by the design and operating principle of the refrigeration machine. The choice of refrigerant depends on the field of application of the refrigeration machine. The most commonly used refrigerants in air conditioning today are R134a, R407C, R404A and R507.

The states of the refrigerant in a refrigeration machine can be presented in general terms as follows:
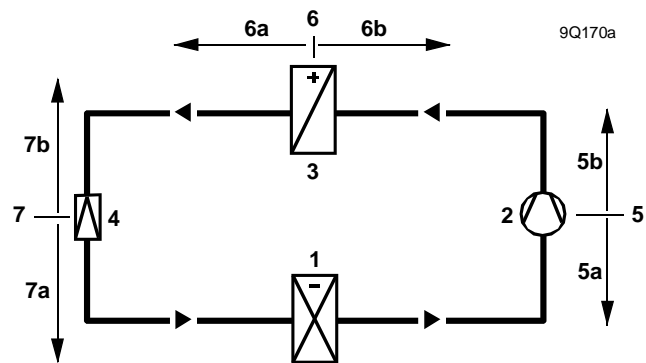
Fig. 5-2   Refrigeration cycle (principle)
    1 Evaporator
    2 Compressor
    3 Condenser
    4 Expansion valve

The various zones are basically distinguished according to
5 Temperature: 5a low, 5b high temperature
6 State: 6a liquid, 6b gaseous
7 Pressure: 7a low, 7b high pressure

5.1.3 System

In the following, the physical processes will be explained in the sequence of the refrigeration process using the example of an actual refrigeration system, and then they will be presented on the h-log p diagram.

The letters used in the text refer both to the pictorial schematic of the system (Fig. 5-3) and to all other drawings in the following, i.e. they are used to identify the operating points.

The system shown is an air-cooling system for a computer center. It consists of a single-stage, suction gas cooled compressor, a finned tube evaporator, a thermostatically controlled expansion valve and an air-cooled condenser with two-stage control. The condenser is installed on the roof of the building and the compressor and liquid collector are on the next floor down. The evaporator is installed in an air conditioning system two floors lower down.

System data:

Cooling output	$\dot{Q}_0$:	25 kW
Compressor power	P :	8.75 kW

Operating data:

Refrigerant		R134a
Vaporization temperature	t_o :	+5 °C
Vaporization pressure (p_{abs})	p_o :	3.5 bar (3.497)
Condensation temperature	t_c :	+45 °C
Condensation pressure (p_{abs})	p_c :	11.6 bar (11.592)

Fig. 5-3   Pictorial schematic of the system

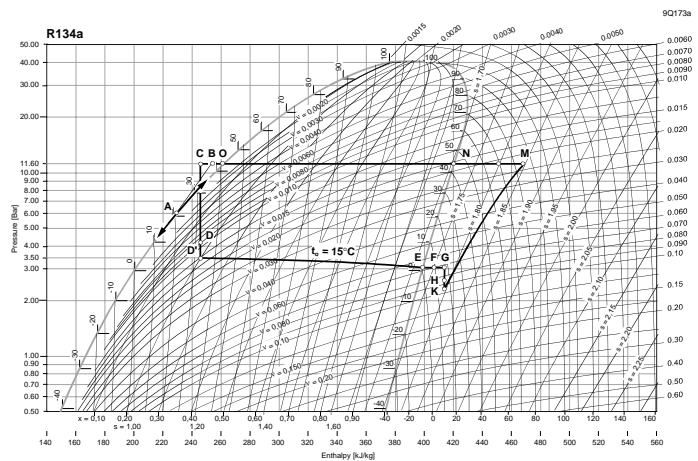|			
1	Evaporator	7	Riser
2	Compressor	8	Suction pipe
3	Condenser	9	Heat exchanger
4	Expansion valve	10	Solenoid valve
5	Differential pressure valve	11	Insulation packing
	Pressure equalization pipe	12	Refrigerant distributor
6	Liquid collector		

Fig. 5-4   The process on the h-log p diagram

5.2 The refrigeration cycle on the h-log p diagram
5.2.1 Liquid collector

The liquid collector is located immediately downstream from the condenser. The condensed refrigerant gathers in it. Its volume is selected such that the charge fluctuations occurring in operation due to temperature/load changes and control actions can be compensated (Fig. 5-3).

If the compressor is at a standstill, the liquid collector contains liquid and gaseous refrigerant. A given pressure occurs for every temperature in the collector. This is the boiling pressure, which is shown as point A on the h-log p diagram (Fig. 5-4). As the temperature changes, the pressure also changes: point A travels along the boiling line when the system is at a standstill.

When the compressor starts up, the pressure in the condenser and in the liquid collector rises until the working pressure is reached (approx. 11.6 bar in this example). The hot-gas condenses at 45 °C (boiling point of R134a at 11.6 bar). In the condenser, more heat is extracted from the refrigerant than is necessary for condensation, so the medium is subcooled. The liquid temperature is generally about 5 K below the condensation temperature. Therefore, point B is to the left of the boiling line, at 11.6 bar and 40 °C in this example.

The refrigerant leaves the collector via the liquid pipe and flows to the heat exchanger. The warm condensate cools by several degrees there. The heat is transferred to the cool suction gas. In Fig. 5-5, the state point travels from B to C at constant pressure. The temperature at point C is 35 °C, and the enthalpy is 249 kJ/kg.
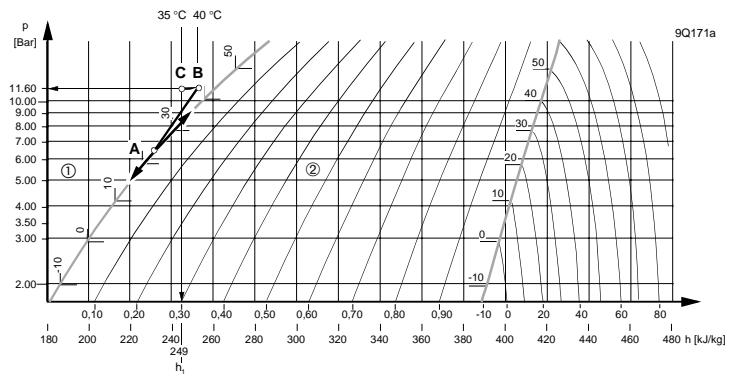
Fig. 5-5   Progression of the boiling pressure (A) of the refrigerant in the liquid collector
1 Liquid
2 Wet vapor

5.2.2 Expansion valve

The expansion valve controls the quantity of refrigerant flowing into the evaporator, reducing its pressure at the same time. Therefore, this valve forms the boundary point between the high-pressure and low-pressure side of the system. In the case shown here, the vaporization temperature $t_o = 5$ °C, which corresponds to a vaporization pressure of 3.5 bar. Therefore, the condensation pressure must be reduced from 11.6 bar to 3.5 bar.

The expansion valve initially reduces the pressure to approximately 4.4 bar (line C-D in Fig. 5-6), and the refrigerant distributor downstream from it gives rise to a further pressure drop of approximately 0.9 bar (line D-D').

On the h-log p diagram, the operating state downstream from the expansion valve travels vertically downwards along the isenthalp that intersects point C. Point D describes the state downstream from the expansion valve, and point D' the state downstream from the distributor at the evaporator inlet. D' is characterized by vaporization pressure p_o = 3.5 bar, enthalpy h_1 = 249 kJ/kg and vaporization temperature t_o = + 5 °C.

Additionally, point D' enables the proportion of flash gas x to be determined. In this example, it is approximately 20 % (x = 0.2). This portion of the refrigerant already evaporated during expansion (without heat exchange), i.e. only 80 % of the overall latent heat of vaporization is extracted from the medium to be cooled. Therefore, it is important that point C should be as far to the left in the liquid zone as possible, i.e. the liquid refrigerant should be subcooled as much as possible.
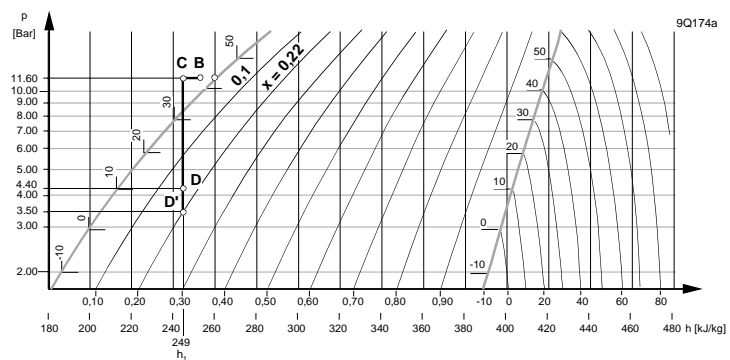

Fig. 5-6   Progression of the refrigerant's operating state across the expansion valve

5.2.3 Distributor and evaporator

The refrigerant distributor feeds the evaporator tubes, which are connected in parallel, so that the refrigerant travels exactly the same distance in each case. This ensures an even distribution to the evaporator tubes. The further continuous change of state takes place in these tubes. The proportion of gas increases continuously until all of the liquid has evaporated.

The saturated vapor generated in the evaporator is identified as point E on the diagram.
E is to the right of D' but has a slightly lower pressure, where the difference corresponds to the pressure loss in the evaporator. The pressure at point E p_{oE} is 3.2 bar; the enthalpy h_2 has risen to 399 kJ/kg K.

The heat absorbed by the refrigerant in the evaporator is slightly more than the energy required to reach the saturated vapor point. This means that the refrigerant is superheated by the time it leaves the evaporator – usually by about 5…8 K.

The end point of the superheating is identified as F on the h-log p diagram (Fig. 5-7). Superheating is necessary in order to ensure that no droplets of liquid refrigerant can enter the compressor, which could be damaged by slugging as a result.

The superheating temperature is the actual controlled variable of the expansion valve.
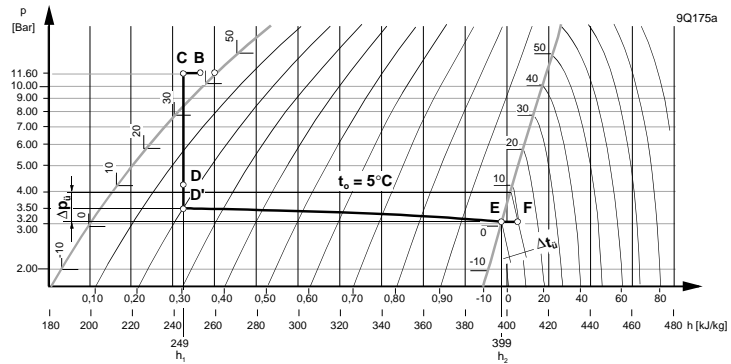
Fig. 5-7   State progression of the refrigerant across the evaporator

Therefore, the saturated vapor point E is not at the evaporator outlet (see Fig. 5-10, ←E→) and is not fixed; instead it moves depending on the respective load change.

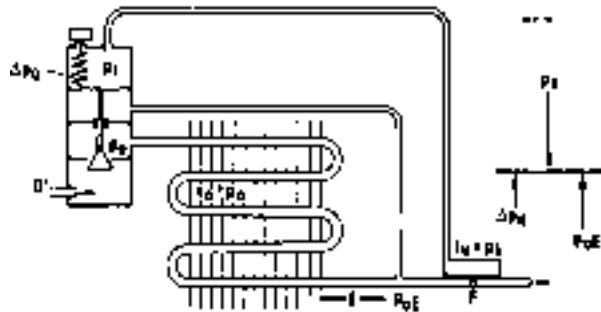Fig. 5-8   Superheating at the evaporator

In the example, superheating $\Delta t_ü$ corresponds to a pressure differential $\Delta p_ü$ of 0.8 bar. $\Delta p_ü$ is adjustable at the valve and, together with the evaporator pressure p_o acting on the valve diaphragm, it provides the counterforce to the sensor pressure p_1. The sensor pressure corresponds to the temperature at the sensor. It is produced by the evaporation of the sensor fluid.

In the balanced state, the refrigerant injected into the evaporator can evaporate (E) and also overheat (F). If the load decreases (e.g. due to a lower air inlet pressure at the evaporator) the saturated vapor point E approaches the evaporator outlet, and the superheating decreases. As a result, the sensor pressure p_1 decreases, and the valve closes, reducing the charge of the evaporator until a new balanced state where $p_1' = p_o' + \Delta p_ü$ occurs. In order to prevent the pressure losses in the distributor and evaporator from having too great an impact on the superheating, the pressure p_{oE} at the evaporator outlet is measured as a comparison value instead of po at the evaporator inlet and applied to the valve diaphragm (see Fig. 5-3, No. 13 and 8). $P_1 – p_{oE} = \Delta p_ü$ = const.

5.2.4 Suction pipe and heat exchanger

The gas passes from the evaporator to the compressor via the intake pipe. Due to the risk of water condensation or icing on the intake pipe, it is generally insulated. The cold suction gas then enters the heat exchanger, where it is additionally heated (F – G), while on the other hand, the warm condensate is further subcooled (B – C).

The gas is then drawn into the compressor via the suction pipes and the suction throttling valve. Attention must be paid to the pressure drop, first in the pipe: point G falls to H where Δp = 0.5 bar, then in the valve: point H continues to fall to K. The pressure drop across the fully opened suction throttling valve should not exceed 0.3 bar.

Note: The control valve controls the refrigeration capacity between 25 and 100 %. The operating principle of the control and the necessary equipment on the refrigeration machine side are not covered in this article. Therefore, the valve is simply displayed in the fully open state where Δp ~ 0.3 bar (H – K) on the h-log p diagram.
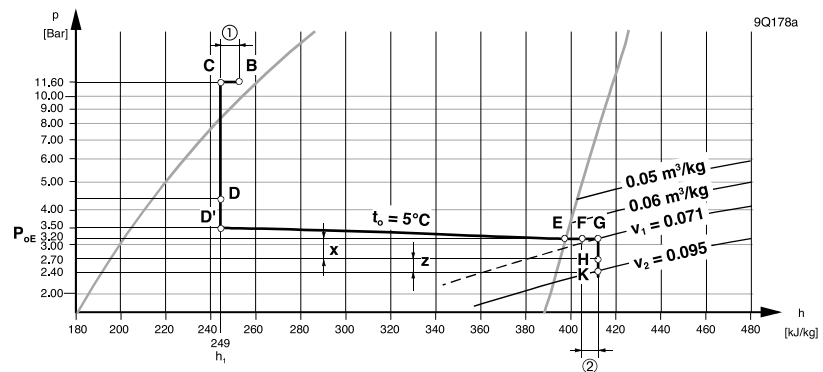

Fig. 5-9   Pressure loss x in the suction pipe (G → H)

z = residual pressure drop at the control valve
1 = superheating
2 = subcooling

5.2.5 Compressor   The compressor produces a pressure differential and, therefore, transports the refrigerant. Three different compressor types are distinguished: hermetic, semihermetic and open. In the first two types, the electric motor is enclosed in a common housing with the compressor and is cooled via the suction gas. The motor is specially insulated for operation in the refrigerant. This design offers the advantage that no refrigerant can escape at the shaft gland between motor and compressor. Semi-hermetic compressors are used most frequently in air conditioning refrigeration systems.

The compressor draws in the gas in state K. The pressure differential G – K is the pressure drop in the suction pipe, including the fully opened suction throttling valve. K corresponds to the intake pressure p_{os} of approximately 2.4 bar, an enthalpy h_3 = 413 kJ/kg and a suction gas temperature t_{os} = 17 °C. Point K is also located on the isentrope with

s_1 = 1.8 kJ/kg K.

In an ideal, i.e. loss-free, compressor, the gas compression would follow this line of equal entropy until the desired pressure was reached (adiabatic compression).

The ultimate pressure corresponds to the condensation pressure pc. This pressure depends on the sizing of the condenser and is 11.6 bar in this example. The theoretical compressor power P_{th} is the product of the theoretical work

w_{th} = (h_4 - h_3) and the refrigerant mass flow $\dot{m}$:

(Determining the circulating refrigerant volume, see chapter 5.5)

$$p_{th} = \dot{m} \cdot (h_4 - h_3) \Rightarrow \frac{600 \text{ kg/s}}{3600 \text{ s/h}} \cdot (450 - 413) \text{ kJ/kg} = 6.16 \text{ k}$$

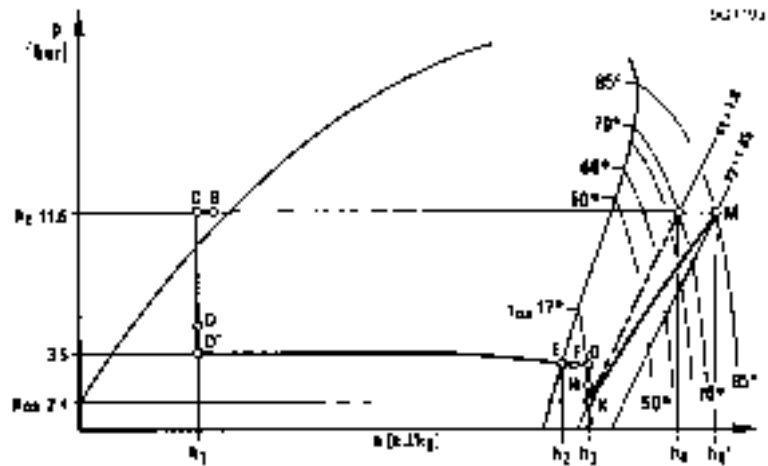The theoretical temperature on the gas discharge side would be 70 °C.

Fig. 5-10 Theoretical and real (K → M) progression of the operating state change in the compressor

However, a real compressor operates with considerable losses.
A distinction is made between, on the one hand, volumetric losses caused by:

- thermal expansion of the suction vapor entering the compressor as a result of heating on the warmer surfaces (motor winding, piston, cylinder, etc.)
- leakage between piston and cylinder
- leakage of the working valves
- "unusable space" in the cylinder, which remains filled with a remainder of gas that expands again during the intake stroke thus reducing the intake of fresh refrigerant
- and, on the other hand, mechanical losses due to
  – friction between the moving parts
  – additional work output (e.g. of the oil pump)

Depending on the actual efficiency, the additional work to be provided by the compressor due to the losses moves point p_c/h_4 in Fig. 5-11 a long way to the right to point M, while the pressure head remains the same. Point M can be located by calculating the enthalpy difference $h_4 - h_3$ from the effective (given) compressor power P_{eff} and the mass flow $\dot{m}$. In order to do so, the equation used above for the theoretical compressor power must be modified, i.e. resolved for the enthalpy difference:

$$h_4' - h_3 = \frac{p}{\dot{m}} = \frac{8.75}{600} \cdot 3600 = 52.5 \text{ kJ/kg}$$

If $h_3 = 413$ kJ/kg the result is: $h_4' = 465.5$ kJ/kg

Point M is located on the isotherm t = 85 °C. This is the temperature of the compressed gas. It can be in the range 80…120 °C. However, it must in every case be below the decomposition temperature of the lubricating oil dissolved in the refrigerant, because decomposed lubricating oil reacts with the refrigerant to form acids that destroy metal parts and motor winding insulation.

5.2.6 Hot-gas pipes and condenser

The refrigerant leaves the compressor in a greatly superheated state (point M) via the hot-gas (discharge) pipe and eventually enters the condenser. At the front end of the condenser, the hot-gas must first be cooled to the condensation temperature t_c given by the pressure.

The fans, controlled by pressostats, force cooling air through the fine, large-area fins of the condenser, which gives rise to an intensive exchange of heat between the now condensing gas and the air. Towards the back end of the tube rows, the refrigerant has completely condensed and is now subcooled due to the temperature differential to the cooling air. The refrigerant now returns to the collector via liquid pipe B.

The three zones of the condensation process are visible on the h-lop p diagram in the form of the heat extraction line segment M → N, the actual condensation N → O and subcooling O → B.

The cooling of the condenser is usually controlled via the condensation pressure or temperature.
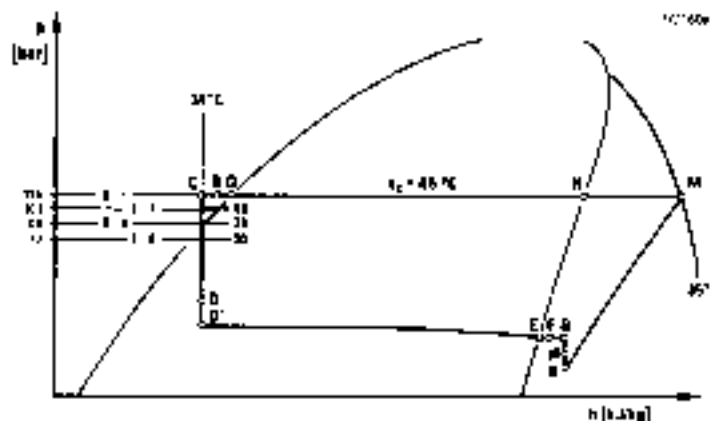

Fig. 5-11 Three phases of the condensation process and operating points of the condenser fan control

The condenser fan control operating points are as follows in this example:

if t_c	=	+ 30 °C	1st fan off	I	=	0
		+ 35 °C	2nd fan off	II	=	0
		+ 40 °C	1st fan on	I	=	1
		+ 45 °C	2nd fan on	II	=	1

5.2.7 Pressurization of the liquid collector

Fig. 5-4 clearly shows that a relatively low pressure prevails in the collector at low liquid temperatures (point A). When the refrigeration machine starts, the pressure differential between A and D will usually be insufficient to force enough refrigerant through the expansion valve. The result would be insufficient liquid in the evaporator. The machine would be shut down again by the low-pressure pressostat. Therefore, gas is forced into the liquid collector via a bypass pipe and differential pressure valve (5) on the discharge side of the compressor as soon as the condenser temperature t_c (approx. 30 °C) falls below a given value (see Fig. 5-3 and Fig. 5-5).

5.3 Further plant parts and their problems

Liquids are practically incompressible. Therefore, the compressor would be damaged if liquid refrigerant were to be supplied to it. The piston would run up against a medium that cannot be compressed. Such impacts in the stroke area are referred to as slugging.

5.3.1 The solenoid valve prevents damage

When the system is shut down, liquid refrigerant can gather upstream of the compressor and be drawn in when it starts up again, thus causing slugging. In order to prevent this happening, a solenoid valve is often installed on the high-pressure side, upstream of the expansion valve; the solenoid valve closes when the system is shut down, which stops the flow of refrigerant into the evaporator. The compressor then continues to run until the pressure on the intake side falls below the value set at the low-pressure pressostat. This is referred to as evaporator pump down. At the same time, this measure also protects the low-pressure side against excessive pressure arising due to the evaporation of any residual liquid on shutdown.

5.3.2 Oil transport

In addition to the refrigerant, the refrigeration circuit usually contains a certain quantity of lubricating oil. The proportion of oil is low compared with the volume of refrigerant (approximately 2 %). The oil is used to lubricate the moving parts of the compressor and cannot be completely separated from the transported fluid flow. It travels through the entire refrigeration system with the refrigerant. It reaches its greatest viscosity in the evaporator where the temperature is lowest.

Downstream from the evaporator, the oil is drawn along partially in the form of suspended droplets. However, the larger portion is transported along in the form of a film on the pipe walls. In the process, the oil must overcome considerable vertical distances (two floors in this example). In order to ensure reliable oil transport, the sizing of the refrigerant pipes must provide for a sufficiently great flow velocity. However, higher velocities give rise to greater pressure losses. In very general terms, there is an inverse relationship between pressure losses, pipe cross-sections and flow velocities. Therefore, pipes are sized according to an economically justifiable compromise. In our example, we assume an intake pipe diameter of 42 mm (A = 1,385 mm^2). With a specific volume v = 0.071 m^3/kg (point G) and a refrigerant flow = 600 kg/h, the flow velocity $\vec{v}$ at full power is:

$$\vec{v} = \frac{\dot{m} \cdot v}{A} = \frac{600 \cdot 0.071 \cdot 10^6}{3600 \cdot 1385} = 8.5 \text{ m/s}$$

Experience shows that a minimum velocity of approximately 5 m/s must be maintained in the present case. This requirement is met.

The equation gives rise to $\vec{v} = 11.4$ m/s at point K.

If the circulating refrigerant volume is governed by capacity control, special attention must be paid to oil return. In such cases, the pipes are divided into two legs with different diameters at vertical rises (see Fig. 5-3, item 7 Riser). If the oil transport fails due to the reduced gas velocity as a result of the capacity control, the oil gathers in the lower bend (I) and interrupts the gas flow in the main pipe.

The refrigerant vapor is then transported via the bypass pipe with the narrower cross-section (II). This pipe is sized such that oil transport is guaranteed even at minimum load.

Again: lubricating oil and refrigerant

The oil absorbs refrigerant. The colder the oil, the more refrigerant it can absorb. Increasing pressure also increases the refrigerant's solubility. On standstill, the oil mainly gathers in the crankcase. On startup, the pressure drops abruptly and the dissolved refrigerant evaporates again, which causes the oil to foam up in the crankcase.

This foaming has further consequences:

1) The oil pump cannot transport the foam, so the oil pressure collapses.
2) The foam gives rise to slugging in the compression space.
3) Oil migration can leave insufficient oil in the compressor.

In order to minimize the amount of refrigerant absorbed by the oil, the crankcase is electrically heated when the system is switched off. In large compressors, oil transport is monitored by differential pressure pressostats.

5.3.3 Safety devices

If the coolant for condensing the hot-gas fails for whatever reason, the pressure downstream from the compressor will rise above the permissible value p_{cmax} and overload the compressor. In order to protect the system, there is a *pressostat* on the high-pressure side, which cuts off the drive motor if the maximum permissible pressure is exceeded.

The switch-off point p_{cmax} in the system under discussion is 14 bar, which corresponds to a condensation temperature t_c of 52.5 °C. In this case, the compression temperature would rise far higher than in normal operation (85 °C).

On the other hand, it can occur that the volume of circulating refrigerant or the amount of heat supplied for evaporation is insufficient. In either case, the pressure on the low-pressure side of the compressor, and with it the vaporization temperature t_o, decreases. Therefore, the compressor is shut down by a *low-pressure pressostat* (LP pressostat) if the pressure falls below a given value p_{omin}.

The switch-off point p_{omin} of the LP pressostat is given here as 1.8 bar, which corresponds to a temperature of –12 °C.

Suction gas cooled compressor motors always require a minimum amount of refrigerant for self-cooling (approximately 40 % of the overall volume). Without the refrigerant, there is a risk that the motor will overheat. In order to prevent damage due to overheating, a *thermal cutout* (Klixon) is installed on or in the winding of all semihermetic and hermetic compressors. This device interrupts the power supply in case of overheating.

If the lubrication oil pressure falls below a given value, an oil pressure differential safety switch deactivates the drive motor after a certain time delay. This switch is short-circuited on compressor startup.

A *thermal overload* on the motor protection switch also interrupts the power supply if the current consumption exceeds the set value, e.g. in case of overloading.

On startup, an electric motor draws many times its operating current. The following measures reduce the starting current:

- *Star-delta starting*
- *Part-winding starting*
- *Starting load reduction* (a solenoid valve opens a bypass between the high-pressure and low-pressure sides of the compressor to reduce the load by decompression).
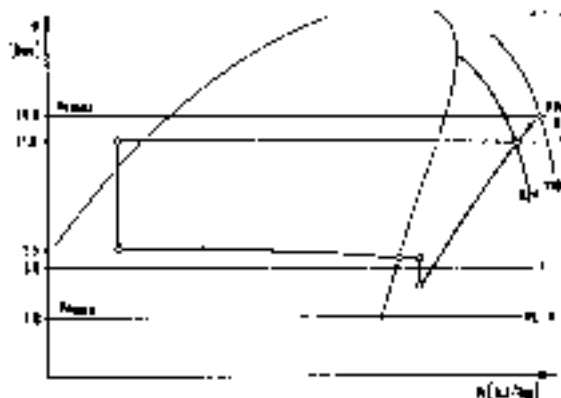

Fig. 5-12 Compressor switch-on and switch-off points (1 = on, 0 = off)

The machine stops …   If no more cooling capacity is required, the solenoid valve closes the liquid pipe. The compressor pumps down the evaporator before being switched off by the LP pressostat. The valve does not usually close absolutely tightly, so even at standstill a certain quantity of refrigerant enters the evaporator and evaporates, which causes the pressure to rise. If the pressure rises above the value set at the LP pressostat, the compressor starts briefly and pumps down the evaporator again.

… and starts again   The starting command from the external control opens the solenoid valve in the liquid pipe again. If all safety circuit elements are closed (LP and HP safety pressostats, Klixon, oil pressure) the compressor starts with reduced load on the cylinders. After a given delay, the oil pressure switch is brought online, and the starting load reduction is removed. The high-pressure side gradually warms up, and the operating state is achieved again after a few minutes.

The motor winding heats up considerably during each startup. The more frequent the starts, the greater the risk. A *restart inhibit* prevents the compressor from starting again within a certain timeout period, thus preventing the motor from overheating.

5.4 Energy balance

The heat quantities supplied and extracted per kg of refrigerant appear as measurable distances on the h-log p diagram. The corresponding heat output can be obtained by multiplying the respective enthalpy difference Δ_h by the volume of circulating refrigerant:

$$\dot{Q} = \dot{m} \bullet \Delta h$$

Heat output of evaporation (= refrigeration capacity):

$$\dot{Q}_o = \dot{m} \bullet (h_2 - h_1)$$

Compressor power:

$$P_{eff} = \dot{m} \bullet (h_4' - h_3)$$

Condenser output:

$$\dot{Q}_c = \dot{m} \bullet (h_4' - h_1)$$

Since the latent heat of vaporization and the compressor work that has been converted to heat have to be dissipated again in the condenser, the balance is as follows:

$$\dot{Q}_c = \dot{Q}_o + P_{eff}$$

The heat flows $B - C = h_1' - h_1$ and $F - G = h_3 - h_2'$ are identical and represent heat exchange within the system only. The superheat $E - F$ is drawn partly from the cooled air (evaporator) and partly from the environment (suction pipe).

Instead of dissipating the entire heat quantity $\dot{Q}_c$ to the environment, appropriate measures (see chapter 6 "Heat recovery in the refrigeration cycle") could be implemented in order to put this heat flow to meaningful use for heating purposes because of its temperature level.
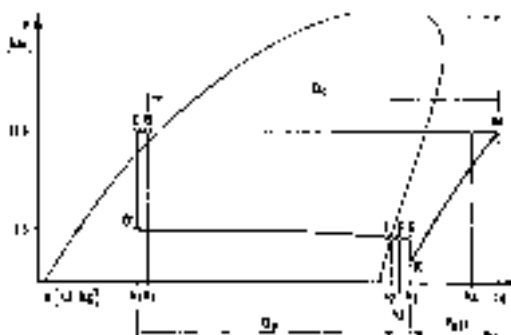

Fig. 5-13 T.b.d.

In order to determine the efficiency of a refrigeration machine, the coefficient of performance ε (epsilon) is used to compare the benefit (refrigeration output $\dot{Q}_o$) with the cost (drive power P):

$$\varepsilon = \frac{\dot{Q}_o}{P_{eff}} = \frac{h_2 - h_1 \bullet \dot{m}}{(h_4' - h_3) \bullet \dot{m}} = \frac{h_2 - h_1}{h_4' - h_3}$$

In order to obtain the real coefficient of performance for the overall system, the energy expenditure for the transport of the heat transfer media on the cold and warm sides of the refrigerant cycle must also be taken into account both with regard to energy consumption and with regard to the resultant heat output (e.g. fan heat).

5.5 Determining the volume of circulating refrigerant

The measurable distance D' – E on the h-log p diagram (see Fig. 5-7) is the usable latent heat of vaporization Δh. It can be determined exactly via the enthalpy difference $h_2 - h_1$ and is important for calculating the volume of refrigerant m circulating in the machine. The volume of circulating refrigerant $\dot{m}$ can be calculated for a given refrigeration capacity $\dot{Q}_o$:

$$\dot{Q}_o = \dot{m} \bullet \Delta h = \dot{m} \bullet (h_2 - h_1)$$

$\dot{m}$ = refrigerant volume in kg/h

$\dot{Q}_o$ = refrigeration capacity in kW

Δh = enthalpy difference $h_2 - h_1$ in kJ/kg

$$\dot{m} = \frac{\dot{Q}_o}{h_2 - h_1} \bullet 3,600 \rightarrow \frac{25 \text{ kJ/s}}{399 - 249 \text{ kJ/kg}} \bullet 3,600 \text{ s} = 600 \text{ kg/h}$$

$\dot{m}$ = constant

The refrigerant mass flow is the same at all points in the cycle for a given operating state, whether the medium is liquid (in the liquid pipe) or gaseous (suction pipe, compressor, pressure pipe). On the other hand, the specific volume varies greatly according to pressure, and the flow volume changes as a function of volume and pipe cross-section. These considerations are important for the discussion about oil return.

The circulating refrigerant volume can also be determined graphically using the refrigeration valve selection diagrams. In order to do so, the refrigeration capacity $\dot{Q}_o$, the vaporization temperature t_o and the refrigerant liquid temperature t_{fl} at point C must be known.

In this example:

$\dot{Q}_o$: 25 kW

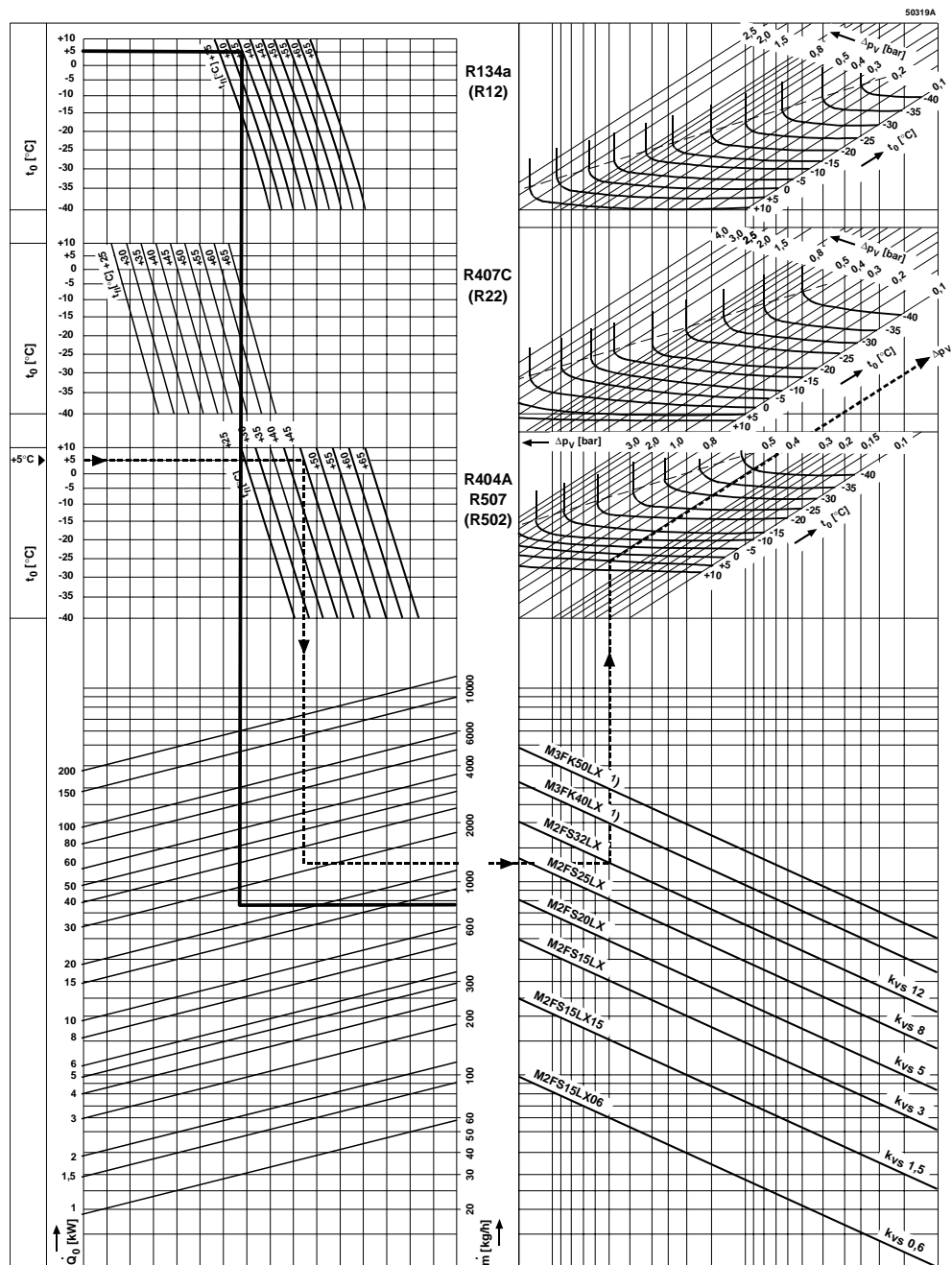t_{fl} : + 34 °C

t_o : + 5 °C

Refrigerant : R134a

Fig. 5-14 Sizing chart for determining the mass flow and the refrigeration valves

5.6 Summary   This chapter described the technical and physical states of a refrigeration system, primary in full-load operation or at standstill. In practice, the operating states vary over a wide range according to the load on the evaporator and according to the outdoor air conditions at the condenser.

It is in the interest of the refrigeration specialist to:

- keep the pressure losses in the suction and pressure pipes low, because they have a major impact on compressor sizing,
- reduce the proportion of flash gas in order to increase evaporator power.

Additionally, the system contractor must take all necessary precautions in order to guarantee reliability and long service life of the system.

6. Heat pump technology

6.1 Introduction

By utilizing environmental heat, the heat pump can normally produce two to three times more heat energy than the auxiliary energy used to operate it. Therefore, the heat pump enables a highly effective use of various types of energy (e.g. electricity) for heating buildings.

6.2 Operating principle of the heat pump

The heat pump exploits the thermodynamic properties of a refrigerant (e.g. Freon R134a) in a closed-cycle process (see Fig. 6-1).

Heat from the environment

The special property of refrigerants is that they evaporate at very low temperatures. This means that the plentiful quantities of energy in the environment (outdoor air down to −20 °C, lake or ground water from 4…12 °C and ground from 0…20 °C) are perfectly sufficient as heat sources in terms of the temperature level to vaporize the refrigerant. The heat source is cooled by several Kelvin in the process. Energy is always required to vaporize a liquid. In this case, the vaporization energy is drawn from the environment. The vaporized refrigerant has absorbed this vaporization energy in the **evaporator** without a rise in temperature. The low temperature level does not permit this medium to be used directly in heating systems.
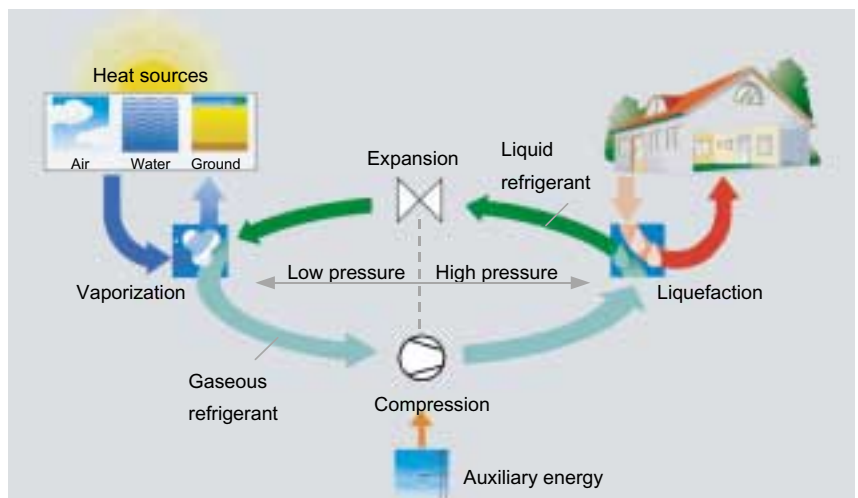

Fig. 6-1   Refrigerant cycle in a heat pump

Vaporization and condensation temperature

The temperature at which a medium evaporates when heat is added to it is the same temperature at which it condenses (liquefies) when it cools, i.e. heat is removed from it. Therefore, that temperature is referred to as the vaporization temperature in the one case and as the condensation temperature in the other.

The vaporization or condensation temperature depends on pressure. With increasing pressure, the vaporization or condensation point also rises in temperature. Based on these physical facts, the next step is highly logical: increase the pressure in order to raise the vaporization/condensation point to a level at which the condensation process can be used for a heating system.

This is done using a **compressor,** which draws in the now gaseous refrigerant and compresses it. Auxiliary energy (e.g. electricity) is required for this process. In the case of a suction gas cooled compressor, this energy (motor heat) is not lost but is transferred to the refrigerant to be compressed and heats it up.

In the subsequent **condenser,** the heating water cools the hot-gas down, causing it to condense and heating up the heating water.

Downstream from the condenser, the refrigerant is completely lique-fied but still at a high pressure. The pressure is dissipated using an **expansion valve,** and the process cycle begins again.
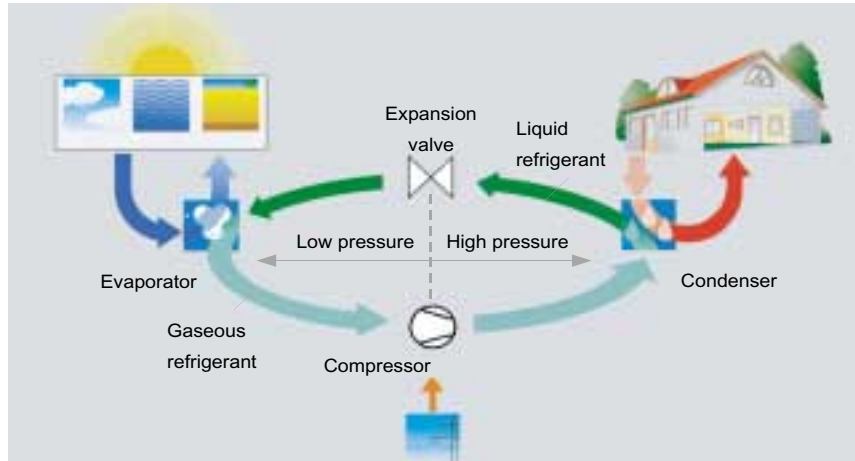

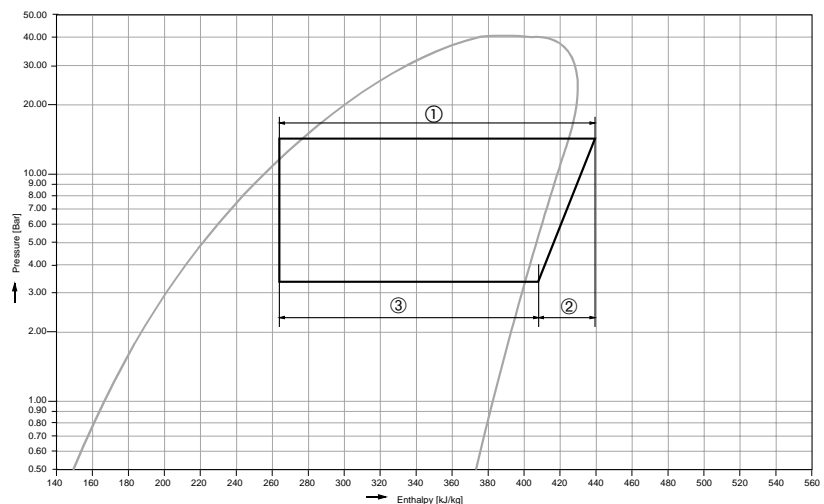Fig. 6-2   Major mechanical components of a heat pump

Fig. 6-3   The heat pump cycle on the h, log p diagram (simplified)

      1 Heat output (Δh • $\dot{m}$)

      2 Applied compressor power

      3 Power (Δh • $\dot{m}$) extracted from the heat source

Origin of the name "heat pump"   The heat pump gets its name from this physical process: Heat energy absorbed at a low temperature level is "pumped up" to a level at which it can be used for heating purposes.

6.3 Heat sources

The heat source provides the heat pump with the necessary heat of vaporization.

6.3.1 Heat source: outdoor air

Fig. 6-4   Heat source: outdoor air

Constantly available heat source

Outdoor air is highly available, so it is frequently used. However, the following properties must be borne in mind:

– The heat pump and its drive power must be designed relatively large for the coldest day (when the coefficient of performance is smallest; see 6.6.1)

– During mild weather, with correspondingly low thermal heat demand, a major excess of heat pump output is available, which may have to be stored

– At outdoor temperatures in the range +5 °C to –10 °C, major icing occurs on the evaporator (condensing airborne humidity freezes on the evaporator surface at a temperature of < 0 °C). This give rise to a sharp drop in evaporator performance. The ice must be regularly thawed off by an appropriate (energy consuming!) method

– The air circulation can cause annoying fan noise, which must be reduced by appropriate soundproofing

6.3.2 Heat source: ground

Fig. 6-5   Heat source: ground

In order to use the ground as a heat source, either **ground collectors** (large area pipe network, usually filled with frost-proof liquid, e.g. water/glycol, installed at least 1.5 m below ground surface) or **ground probes** (deep drilling required) are used.

In order to use ground collectors, an appropriately large plot of land must be available, and investment costs are normally high. The use of ground probes requires drilling, which also gives rise to correspondingly high investment costs.

When using the ground as a heat source, it is very important to ensure that the heat source can regenerate (possibly installing a backup system such as solar collectors). Otherwise, the ground temperature will decrease too greatly so that the necessary output will no longer be available.

For the same reason, it is important with ground probes to ensure that the amount of heat extracted per meter of probe is not too large. Otherwise, the yearly energy coefficient will inevitably deteriorate.

With proper sizing and design, the ground is one of the least problematical heat sources for heat pump operation.
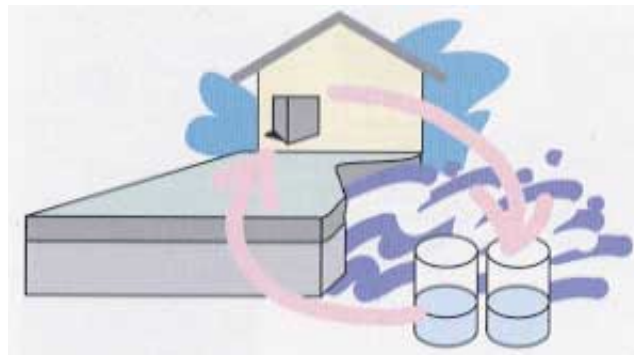
6.3.3 Heat source: ground water

Fig. 6-6   Heat source: ground water

The best but least available heat source

The greatest problems with using **ground water** as a heat source are its availability and quality. However, if it is available in sufficient quantity and quality and at a suitable temperature level, it is an almost ideal heat source for heat pump operation (subject to authorization!).

6.4 Heat pump name

(In Germanic usage) heat pumps are named according to the following principle:

X – Y – Z heat pump, where:

X: Heat source heat transfer medium (e.g. air, water, brine, etc.)
Y: Heating system heat transfer medium (e.g. water, air, etc.)
Z: Compressor drive energy type (electricity, diesel, gas, etc.)

Examples:

Heat pump naming

Heat source	Heat pump name
Outdoor air	Air-water electric heat pump
Ground	Brine-water electric heat pump
Ground water	Water-water electric heat pump

6.5 Operating modes

6.5.1 Monovalent operation

Monovalent operation (mono = one, singular)

In a monovalent heat pump heating system, the heat pump alone (Fig. 6-8) provides the required heating energy in all possible operating states. Therefore, the heat pump must be designed for the maximum heat demand of the building heating system. The maximum possible heating water supply and return temperatures must be designed for the maximum permissible condenser outlet temperature.
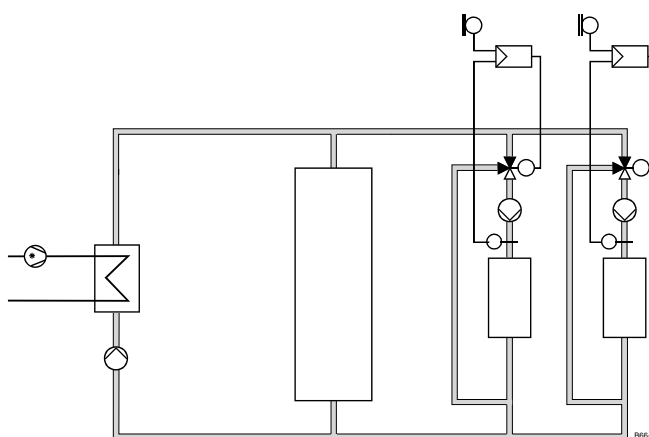

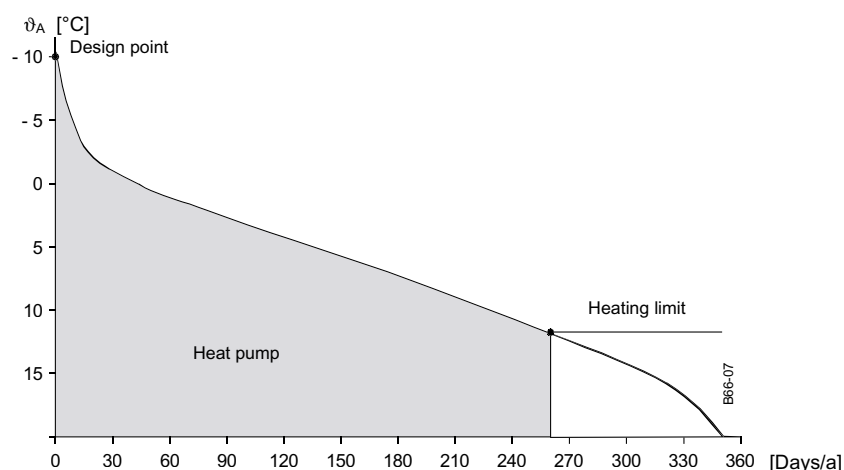Fig. 6-7   System with monovalent operation with storage tank and heating groups

Fig. 6-8   Temperature frequency curve for monovalent operation

In case of failure of the heat pump, no alternative heating is available in a monovalent system.

6.5.1.1 Special case, monoenergetic operation

Since the maximum capacitiy of a plant only needs to be available for a relatively short time, a frequently used solution in single-family homes is an air-water heat pump with electrical auxiliary heating system to cover peak loads. This is actually bivalent/alternative operation (see 6.5.2.1) but since **only one type of energy is supplied –** electricity in this case – the term monoenergetic operation is used.
Experience shows that a plant requires less energy if the switchover is performed manually. Additionally, night setback should be dispensed with at low outdoor temperatures so that boost heating does not become necessary.

6.5.2 Bivalent operation   Bivalent operation (bi = two, twin)

In a bivalent heat pump heating system, the heat pump alone provides the necessary thermal heat during mild weather and average winter weather. In case of very low outdoor temperatures, the heat energy demand is covered partially (parallel) or completely (alternatively) by an auxiliary heating plant (see Fig. 6-9).
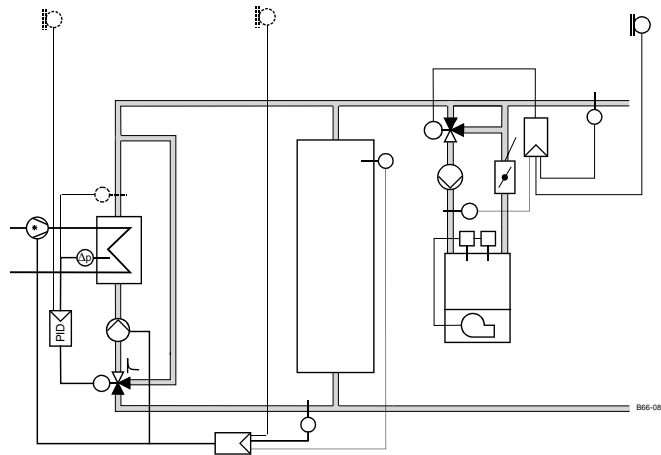

Fig. 6-9   System with bivalent operation with a heat pump, storage tank and boiler to cover peak heat demand

Therefore, the heat pump need only be designed for part of the maximum heat demand of the building heating system.

The auxiliary heating system can be operated with the heat pump in different ways and must be designed and used accordingly. The following operating modes are distinguished:

– Bivalent alternative operation
– Bivalent parallel operation
– Alternative/parallel bivalent operation

6.5.2.1 Bivalent alternative operation   In this case, the heat pump only operates in mild weather and average winter weather. In case of very low outdoor temperatures and to cover the maximum heat demand, the heat pump is switched off and the auxiliary heating plant is switched on.
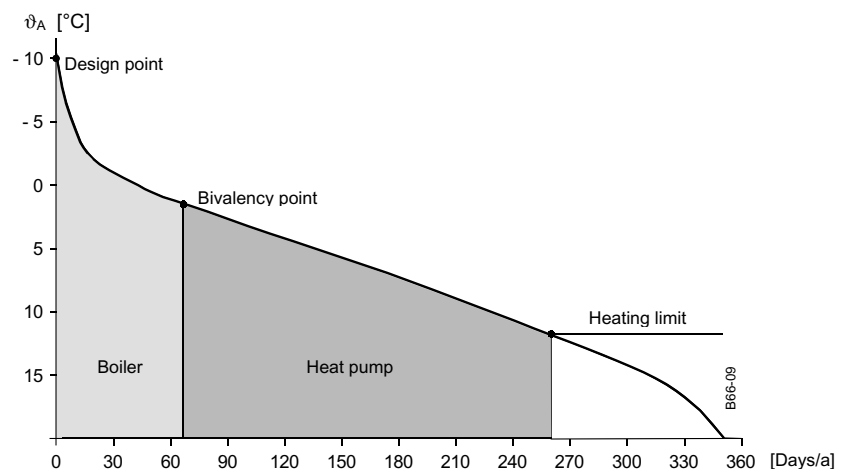

Fig. 6-10 Temperature frequency curve for bivalent alternative operation

The heating water supply and return temperatures must be designed for the maximum permissible condenser output temperature for the load states with heat pump operation. For the load conditions with alternative auxiliary heating plant operation, the heating water supply and return temperatures may exceed these maximum limits. However, the hydraulic connection of the auxiliary heating plant to the heating water circuit must be such that no heating water can circulate through the heat pump's condenser during auxiliary heating plant operation (high-pressure operating limit).

The auxiliary heating plant must be designed for the total maximum thermal heat demand.

During operation, switchover from the heat pump to the auxiliary heating plant must occur as soon as the heat output of the heat pump is no longer sufficient. This is performed by the control equipment based on the outdoor temperature and/or heat source temperature.

6.5.2.2 Bivalent parallel operation   In this case, the heat pump and auxiliary heating plant are operated simultaneously to cover the maximum heat demand of the building heating system.
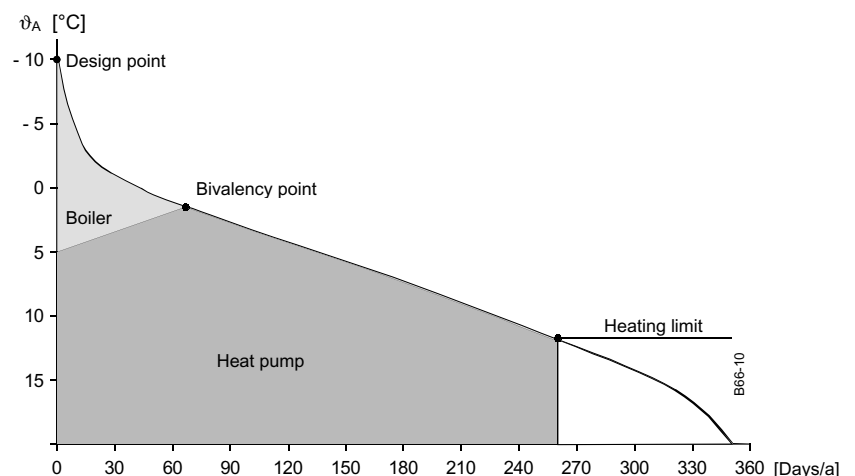

Fig. 6-11 Temperature frequency curve for bivalent parallel operation

The heating system must be designed for the maximum permissible return temperature (condenser inlet temperature). The auxiliary heating plant must be hydraulically connected in series with the heat pump in the heating water supply. The auxiliary heating plant is used to raise the condenser outlet temperature to the necessary supply temperature.

The auxiliary heating plant must be designed for the portion of the maximum heat demand that is not covered by the heat pump.

The auxiliary heating plant is brought online as soon as the heat output of the heat pump alone is no longer sufficient. This is performed by the control equipment based on the heating water supply temperature.

6.5.2.3 Alternative/parallel bivalent operation

This is a combination of parallel and alternative operation.

Low to medium heat demand is covered by the heat pump alone. If the heat demand rises above the heat pump's heat output, the auxiliary heating plant is operated in parallel according to the supply temperature. If the heat demand continues to rise above the heat pump's operating limit, it is switched off (based on outdoor temperature or heat source temperature), and the entire maximum heat demand is covered by the auxiliary heating plant.
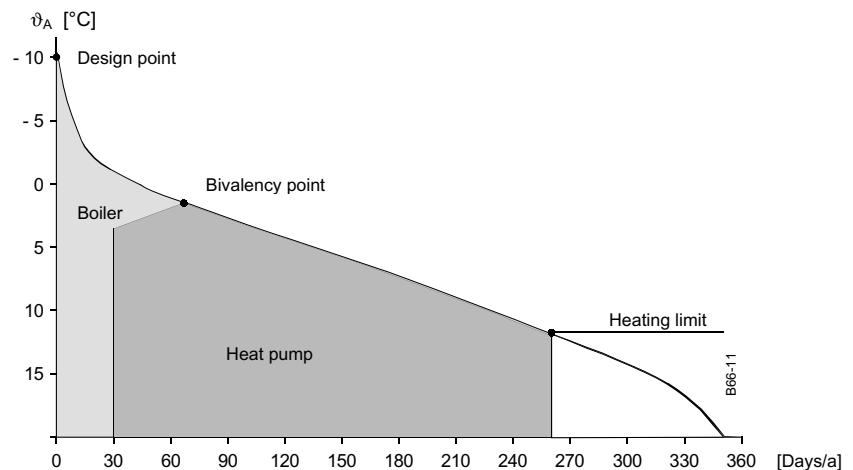

Fig. 6-12 Temperature frequency curve for parallel/alternative bivalent operation

The hydraulic integration of the auxiliary heating plant in the system must be such that:

– it is connected in series in the heating water supply during parallel operation
– no heating water can circulate through the heat pump's condenser during alternative operation
– The auxiliary heating plant must be designed for the total maximum heat demand

6.5.3 Operating mode selection

Correct operating mode

The choice of the most appropriate operating mode (optimal in terms of energy and cost/benefit ratio) depends on the following criteria:

– Yearly progression of the building's heat demand
– Yearly progression of the heat pump's heat output
– Demand-dependent progression of the heating water supply and return temperatures
– Yearly frequency of the various heating load states

6.6 Heat pump characteristics
6.6.1 Coefficient of performance ε

Possibility for comparison of heat pumps

The coefficient of performance ε (epsilon) provides for a comparison of different heat pumps and is the ratio of the current thermal heat output to the (electrical) energy supplied to a heat pump plant (electrically operated) in order to produce it.

Coefficient of performance $\varepsilon = \dfrac{\text{current heat output}}{\text{applied (electrical) power}}$

⇒ The greater the value of ε, the greater the energy efficiency of the heat pump's operation.

The operating concept of a heat pump heating system must take the fact into account that ε (and therefore the heat output of the heat pump) increases with a decreasing differential between the condensation temperature and the vaporization temperature.
This means in practical terms that a heat pump for building heating:

– has the lowest coefficient of performance, i.e. the least heat output, when thermal heat demand is greatest

– has an increasing coefficient of performance, i.e. increasing heat output, with decreasing thermal heat demand

– has the highest coefficient of performance, i.e. the greatest heat output, when heat demand is least

A certain **coefficient of performance ε applies only to a given, momentary operating state.**
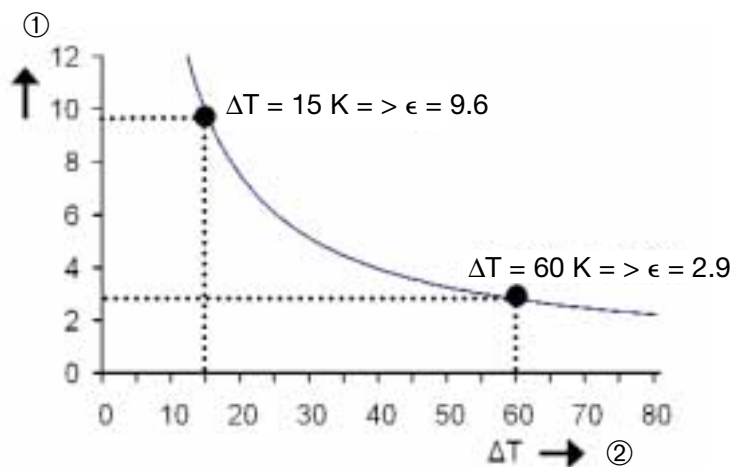

Fig. 6-13 Example for the progression of the coefficient of performance as a function of the temperature differential between the condensation and vaporization temperatures
1 Coefficient of performance
2 Temperature difference

6.6.2 Yearly energy coefficient β

The yearly energy coefficient β provides a measure of the actual efficiency of a heat pump plant.

Yearly average is important

The yearly energy coefficient β is the average of the coefficients of performance ε occurring within an operating year of a heat pump plant.

Typical yearly energy coefficients β occurring in actual practice are as follows, according to heat source:

Heat source	β
Outdoor air	2.5
Ground	3
Ground water	3.2

The yearly energy coefficient β is determined by simultaneous measurement of the yearly electrical energy consumption of the compressor and auxiliary drives (in kWh) and of the yearly heat production (also in kWh) and heat losses of the storage system.

$$\text{Yearly energy coefficient } \beta = \frac{Q_{HP} - Q_{ST}}{W_{HP} + W_{Pumps} + W_{Control} + W_{...}}$$

Q_{HP}	=	quantity of heat produced by the heat pump
Q_{ST}	=	heat losses in the storage system
W_{HP}	=	energy consumption of the heat pump
W_{Pumps}	=	energy consumption of the evaporator and condenser pumps
$W_{Control}$	=	energy consumption of the control equipment
$W_{...}$	=	energy consumption of other components, such as defrosting equipment, Carter heater, …

This requires an appropriate measuring concept (planning phase) for the heat pump plant, and the plant must be equipped with the necessary sensors and meters (electricity and heat meters).

6.7 Heat pump controllability

A heat pump without controllable heat output produces excess heat during partial load heating operation.

The correct choice of heat pump output control must be specified by the heat pump manufacturer and taken into account in plant sizing and design.

6.7.1 Heat output control directly at the heat pump
6.7.1.1 Hot-gas bypass or suction throttling

Heat pump output control using a proportionally controlled hot-gas bypass or suction throttling valve is pointless, because in either case a reduction of the heat output does not give rise to an even approximately equivalent reduction of drive power consumption. Therefore, both hot-gas bypass and suction throttle control give rise to **very poor yearly energy coefficients** for the heat pump.

6.7.1.2 Compressor valve unseating

In appropriately equipped multiple cylinder piston compressors, individual cylinders can be activated or deactivated in stages. This is achieved by unseating the suction valves of the cylinders to be deactivated (e.g. by electrohydraulic means). However, this type of heat pump output control is not energy efficient because considerable friction losses occur in reduced power operation and the inertia of the inactive pistons must still be overcome. Therefore, compressor valve unseating gives rise to a **relatively poor yearly energy coefficient** for the heat pump.

6.7.1.3 Compressor speed control
Good heat pump control

Heat pump output control via multistep (step switch acting on three-phase multispeed motor) or stepless (frequency converter acting on three-phase motor) speed control provides **almost optimal energy use.**

6.7.2 Heat pump ON/OFF control
Most frequently used heat pump control

Electric heat pumps with installed drive loads of up to approximately 40 kW are normally controlled by ON/OFF control today, because the previously mentioned output control methods do not provide optimal energy use, or they give rise to high investment costs. It should be noted with regard to this type of control that frequent switching of heat pumps reduces the life of mechanical parts, increases losses at standstill and causes frequent electrical network fluctuations due to the high starting currents.

Therefore, in order to prevent excessively frequent switching, the heat pump heating system must have sufficient **heat storage capacity** which can temporarily store the excess heat produced by the heat pump and also temporarily cover the heating system's heat demand when the heat pump is off.

To be on the safe side, the heat pump should also be switched **with a delay** in order not to exceed a maximum permissible number of starts per hour (e.g. max. 3 starts per hour). The permissible frequency of starts is often specified by the electrical utility.

7. Ice banks

7.1 Introduction

The reasons for the use of thermal stores in refrigeration are normally plant or application related.

Thermal stores are necessary if a possible failure of the refrigeration machine's drive energy must be covered, e.g. in IT systems, operating theatres or telecommunications installations. Thermal stores reduce the load on the emergency generating equipment.

Thermal stores can also provide economic advantages if storage can be used to reduce the peak load of the refrigeration system or of the entire building, thus reducing the demand rate for electric power. Additional advantages can result from shifting thermal store charging into the low-tariff period. The results depend on the tariff structure of the respective energy supplier.

The influence on investment costs depends greatly on the facility in question; in favorable cases, a tangible reduction can be achieved. The higher and briefer the cooling load peak with respect to the average daily demand, the greater the economic advantages that can be expected from using thermal stores.

Two types of thermal stores are normally used to store refrigeration energy:

- Chilled water storage tanks
- Ice banks

Chilled water storage tanks

Chilled water storage tanks are usually used as buffers in the chilled water system (e.g. 6/12 °C) in order to store refrigeration energy for short periods (minutes or a few hours). This allows refrigeration equipment operation to be optimized, or they are used to improve control stability, especially if the fill volume (= storage mass) of the chilled water system is small in proportion to the installed compressor power.

Similar or identical considerations apply to chilled water storage tanks as to hot water storage tanks with regard to hydraulic circuits, control, etc. – but the other way round, of course. Therefore, this type of refrigeration energy storage will not be covered in any greater detail.

Ice banks

Due to the phase transition from ice to water (and vice versa), ice has an 80 times greater thermal storage capacity than water. Therefore, ice banks are also referred to as PCM (phase change material) devices.

Ice banks offer a much greater storage density (kWh/m^3 or kWh/m^2 of required floor space) than chilled water storage tanks.

The temperature level of the ice bank is around 0 °C due to the phase transition, which has additional advantageous effects.

The use of ice banks offers many advantages but requires appropriate basic control knowledge and correct hydraulic integration into the overall refrigeration system. These topics will be covered in the following.

7.2 Application fields of ice banks

A distinction is made between the use of ice banks in air conditioning, which is the primary focus of this document, and commercial cooling applications.

7.2.1 Use in air conditioning

Ice banks are used in air conditioning for a number of reasons. Ice banks are considered a favorable investment if:

- the peak electrical load in the building is on a summer day, and if the overall peak electrical load can be lowered by reducing the peak refrigeration load (electricity cost optimization)
- electricity can be used for refrigeration during the low-tariff period (electricity cost optimization)
- electrical demand can be made more uniform through the use of ice banks (electricity cost optimisation)

or

- the capacity of an existing refrigeration system is to be increased without upgrading (or replacing) the refrigeration machine
- the refrigeration supply has to be safeguarded for a number of hours during a power outage, e.g. in operating theatres, IT systems, …

7.2.2 Use in commercial cooling

Different considerations come to bear in commercial cooling applications than in air conditioning.

Ice banks are usually used in commercial cooling systems that have a high refrigeration load for a few hours each day, e.g. in breweries, dairies, etc., where the refrigeration machine would be burdened for brief peak periods only.

In such applications, ice banks allow the refrigeration energy to be generated continuously at relatively low power, e.g. over a 12 to 14 hour period (see 7.4.2).

7.3 Design and function of the ice bank

Three main ice bank designs are used:

- Direct-expansion systems
- Ice banks with water-filled plastic balls
- Ice banks with a spiral arrangement of heat exchanger tubes (Calmac)

Direct-expansion systems

In direct-expansion systems (Fig. 7-1), the refrigeration machine's evaporator is located directly in a tank, and ice is produced on the evaporator surface. In order to provide for uniform ice formation on the cooling tubes or plates, the water in the tank is agitated using pumps, or air is injected at the bottom of the tank. Because ice is a poor thermal conductor, efficiency decreases with increasing ice thickness. Ice thickness testers are provided to prevent total freezing, but they are problematical, especially at partial loads and partial load losses.
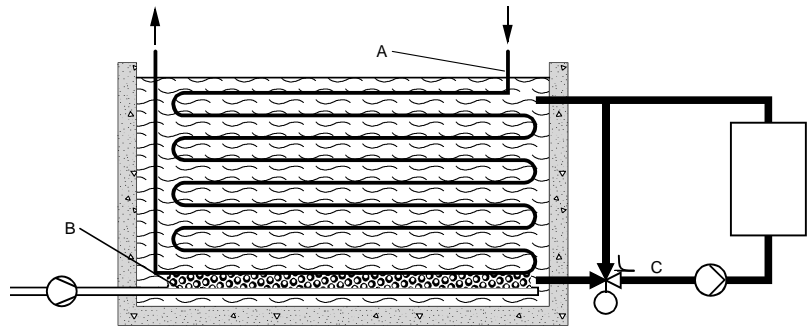
Fig. 7-1 Direct-expansion system
A Direct-expansion evaporator with refrigerant
B Aeration
C Cooling water circuit

Ice banks with water-filled plastic balls

These ice banks are integrated into a brine cycle that is cooled by an evaporator. The tank used for storage is filled to approximately 50…70 % with specially shaped, water-filled plastic balls, which are elastic and can withstand the expansion on freezing. The brine cycle must accommodate the expansion (⇒ expansion vessel).
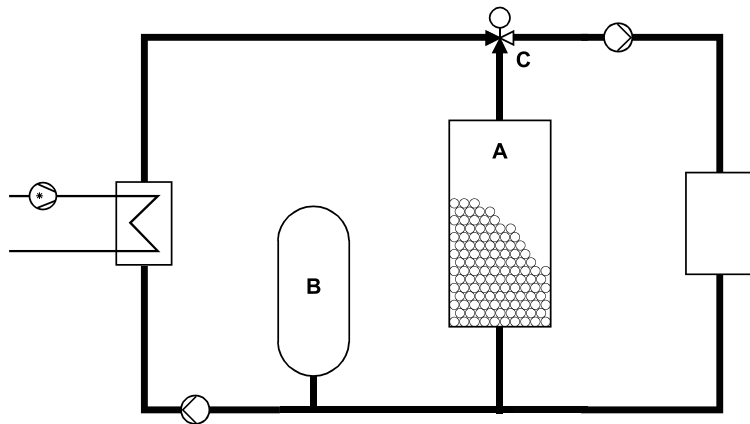

Fig. 7-2 Ice bank with plastic balls integrated into a brine cycle
A Ice bank
B Expansion vessel
C Diverting valve

Ice banks with heat exchanger tubes

In these ice banks, which are frequently used today (e.g. Calmac), a brine solution (glycol-water mixture) flows through the heat exchanger tubes. The heat exchanger tubes are wound in a spiral arrangement with numerous layers (see Fig. 7-3). They completely fill the ice bank container, which means that the heat transfer surface area and, therefore, the output is greater than in direct-expansion systems. There is sufficient space above the heat exchanger tubes to accommodate the water that is displaced during ice formation.
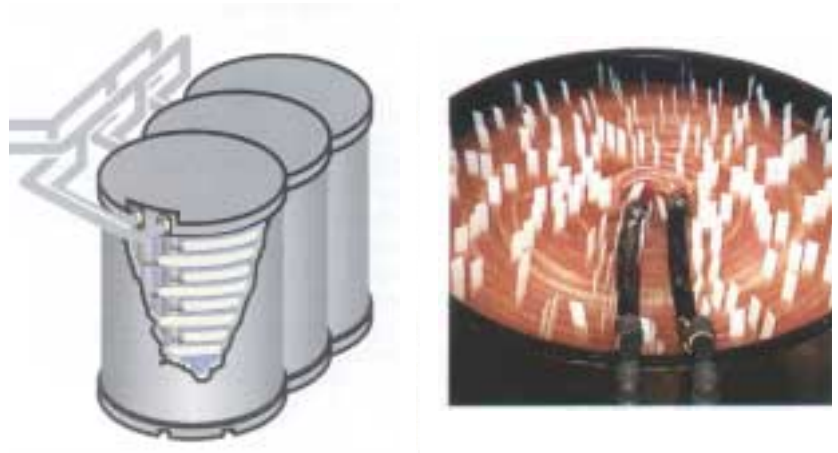
Fig. 7-3   Ice bank with a spiral arrangement of heat exchanger tubes (Calmac)

Ice banks of this kind (container and heat exchanger tubes) are made of plastic (PE), which makes them very easy to transport and relocate. The storage medium (water) and the heat transfer medium are not filled in until after installation.

The containers are suitable for a wide variety of installation locations. They cannot only be installed inside the building but also outdoors, and some even buried in the ground.

Plant parts adapted to glycol-water circuit

The glycol-water circuit is a closed, pressurized system. The glycol-water mixture never comes into direct contact with the stored ice, and it requires neither treatment nor filtering. No heat exchanger is required between consumers and ice banks, although the plant elements (e.g. air cooler) must be adapted to the circulating glycol-water mixture in terms of output and design.

Due to the special arrangement of the heat exchanger tubes, the ice melts from the inside out, not from the outside in as with conventional direct-expansion systems (cf. Fig. 7-1).
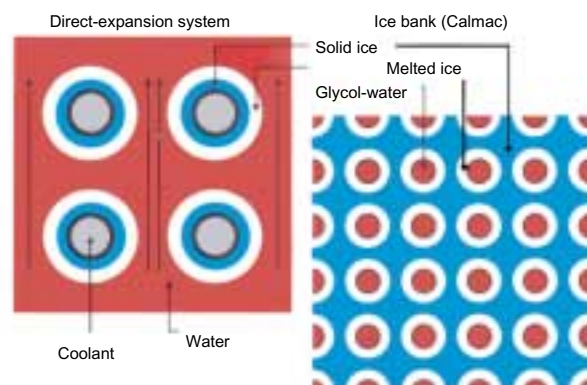

Fig. 7-4   Melting process in a direct-expansion system and a standard ice bank

Ice thickness monitoring is not necessary in this type of ice bank. Due to the dense arrangement of the tubes, less space is required than with direct-expansion systems, and the heat transfer surface area and, therefore, the output is greater.

7.4 Thermal store design

Refrigeration output and cooling energy

In air conditioning systems, refrigeration capacity is usually calculated in kW; in ice banks, however, it is calculated in kWh because they store cooling energy. The relationship between stored cooling energy, cooling load and refrigeration machine output is discussed in the following.

In actual practice, air conditioning systems rarely need 100 % of the cooling output for which they are designed and built during operating periods. Demand peaks generally occur in the afternoon when external influences are greatest. This effect, however, is slightly different for each system, and it depends greatly on building structure and use. Fig. 7-5 shows the typical daily progression of an air conditioning system's refrigeration demand. The maximum refrigeration capacity (200 kW) is only needed for 2 hours of the overall operating time. The demand is less during the other hours. Therefore, the daily cooling energy requirement is 1,640 kWh.
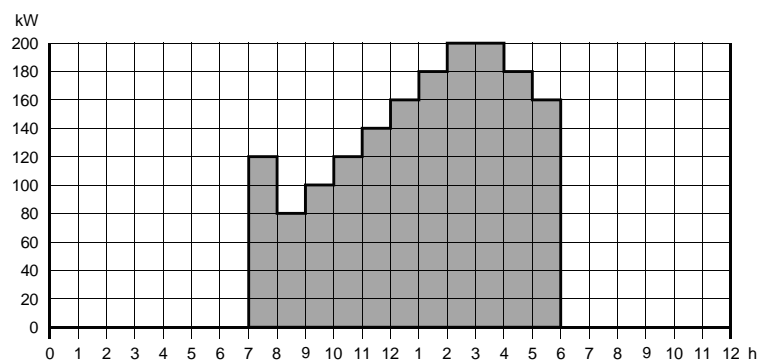

Fig. 7-5   Progression of a system's refrigeration demand (example)

In order to cover the peak demand of 200 kW, an appropriate refrigeration machine with an output of 200 kW is necessary. This refrigeration machine can supply a maximum cooling energy of 2,200 kWh over the operating period (see Fig. 7-6).
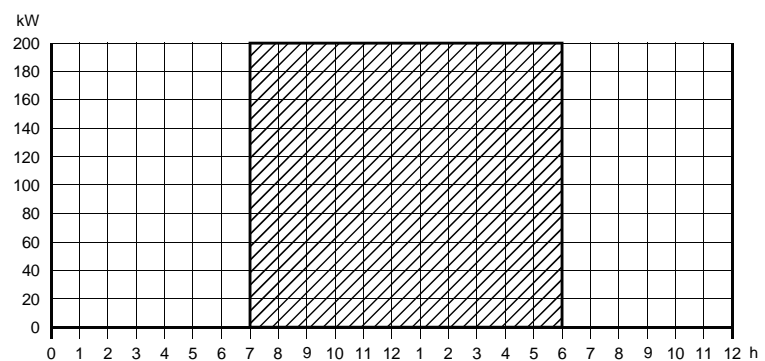

Fig. 7-6   Maximum possible cooling output over the system's operating period (example)

Average factor | The ratio between actual refrigeration demand and total available refrigeration energy is referred to as the average factor. Therefore, the following applies to the refrigeration machine in the example under consideration:

$$\text{Average factor \%} = \frac{\text{actually required kWh}}{\text{total available kWh}} = \frac{1{,}640 \text{ kWh}}{2{,}200 \text{ kWh}} = 75 \text{ \%}$$

The lower the average factor, the less favorable the cost utilization of the air conditioning system will be.

7.4.1 Refrigeration machine and thermal store cover peak demand | With the use of a thermal energy store, refrigeration energy production is optimized, because the necessary peak output is provided in part by the refrigeration machine and in part by the thermal store. This means that the refrigeration machine can be sized considerably smaller.

Average cooling demand | Dividing the building's kWh demand by the number of operating hours of the refrigeration machine gives the building's average cooling demand during the cooling period. This is decisive for the sizing of the refrigeration machine. However, additional considerations must be taken into account for correct sizing of the refrigeration machine and thermal stores.

7.4.2 Partial and full storage
Partial storage | The refrigeration machine is in operation 24 hours a day. The thermal store is charged during the night. During the operating period of the air conditioning system, the refrigeration demand of the consumers is covered by the refrigeration machine with the support of the thermal store. By extending the refrigeration machine's operating time to 24 hours, the lowest possible refrigeration output is achieved (example: 1,640 kWh / 24 h = 68.3 kW, see Fig. 7-7).
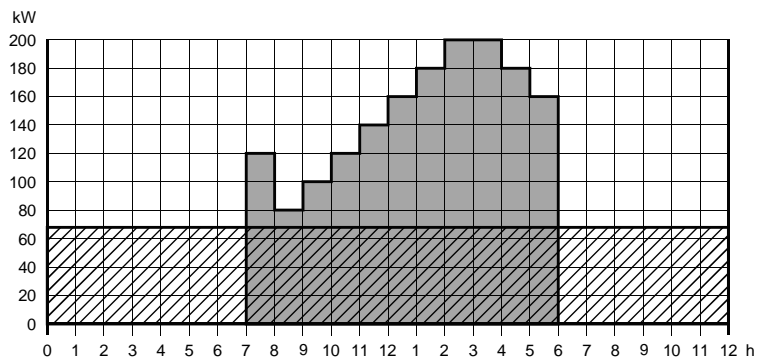

Fig. 7-7 Partial storage: refrigeration demand of the system and average refrigeration capacity

Full storage | In new systems, partial storage is often the most practical and least costly method.

Full storage is applied in systems where, due to the electricity supplier's tariff structure (high and low tariff), the refrigeration machine can only be operational for a given number of hours (e.g. usually at night). During the high-tariff period, the necessary refrigeration energy is taken from the thermal store only. This also reduces the necessary power of the refrigeration machine (example: 1,640 kWh / 10 h = 164 kW, see Fig. 7-8) but not to the same degree as with partial storage.
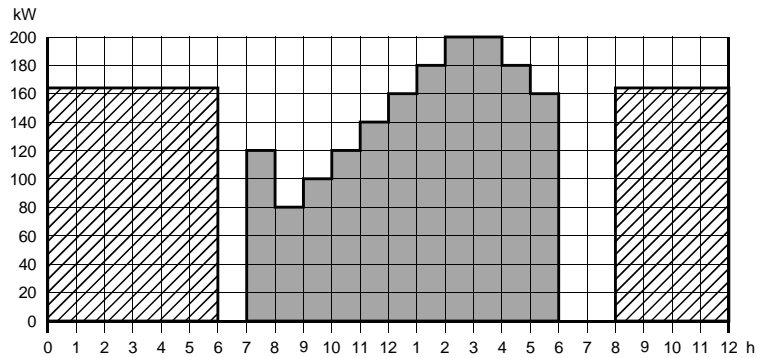
Fig. 7-8   Full storage: refrigeration demand of the system and average refrigeration
            capacity in the low-tariff range

Full storage is meaningful in many systems, especially for reasons of operating costs (tariff structure, base load charges, etc.).

7.5 Hydraulic circuits with ice banks

With regard to the hydraulic integration of ice banks into the overall refrigeration system, a distinction is made between three basic operating modes:

- Charging mode
- Discharging mode
- Bypass mode

7.5.1 Charging mode

In charging mode (usually at night), a glycol-water mixture whose temperature is below the freezing point (min. −3 °C) circulates between a standard water chiller and the heat exchanger in the ice bank. This freezes all of the water in the store to ice. In some ice banks, the heat exchanger tubes are configured for counterflow, which provides for uniform ice formation.
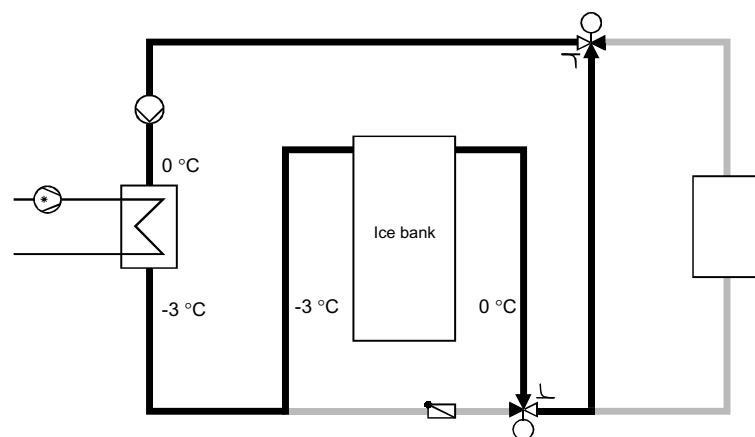

Fig. 7-9   Charging mode of an ice bank

In charging mode, the diverting valve is set such that none of the glycol-water mixture circulates to the consumers (see Fig. 7-9).

During the freezing process, the water crystallizes uniformly in the entire ice bank, starting at the tube surfaces. This prevents the occurrence of stresses and damage to the ice bank.

7.5.2 Discharging mode (series connection)

In discharging mode (during the day), the ice bank cools the glycol-water mixture, e.g. from 10 °C to 1 °C. The supply temperature to the consumers is controlled to the desired cooling temperature (e.g. 6 °C) by a three-port mixing valve via a bypass. The return from the consumers at approximately 12 °C is cooled by the water chiller to 10 °C, for example.
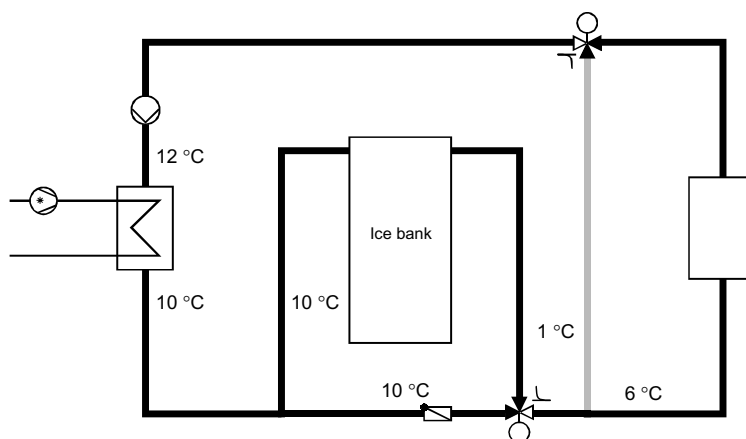

Fig. 7-10 Discharging mode of an ice bank (refrigeration machine and ice bank in series)

7.5.3 Bypass mode

During the intermediate season, the refrigeration machine is able to supply the necessary cooling energy without ice bank operation. All of the glycol-water mixture now flows through the bypass (see Fig. 7-11).
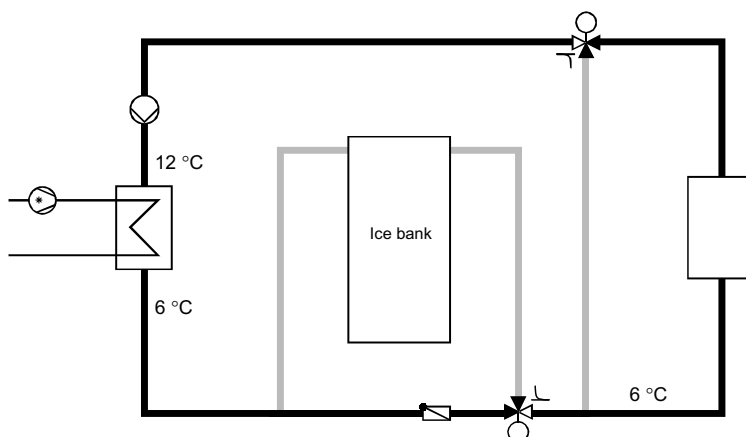

Fig. 7-11 Bypass operation of an ice bank (refrigeration machine only)

Depending on the electricity supplier's tariff situation, it may also be meaningful during the intermediate season to charge the ice bank at night and to draw the cooling energy from it during the day (see 7.4.2 Partial and full storage and 7.5.4).

7.5.4 Hydraulic circuit for full storage

The circuits shown in 7.5.1 to 7.5.3 are suitable for an ice bank system with partial storage (see 7.4.2). In full storage systems, it is not possible with the circuit shown so far to realize cooling with the ice bank only. A different, more complex hydraulic circuit must be used for this purpose.
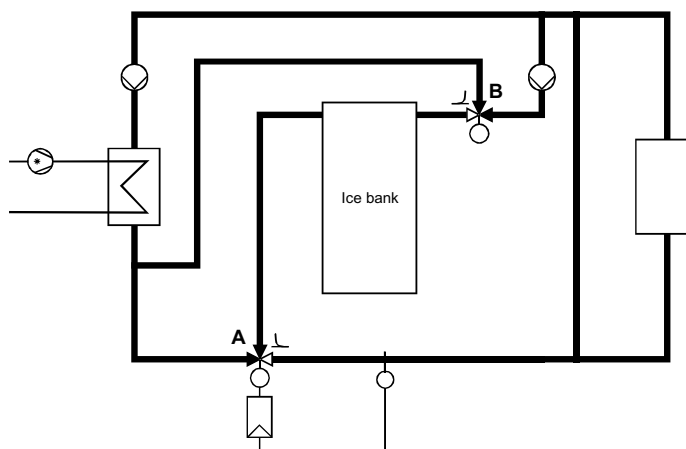
Fig. 7-12 Possible hydraulic circuit for full storage
(with control valve A and diverting valve B)

7.5.5 Different refrigeration machine operating points

It is apparent from the descriptions in the above that the refrigeration machine must operate with different chilled water temperatures depending on the plant situation (charging, discharging in series or bypass) which must be taken into account very precisely in the refrigeration machine's sizing.

7.6 Ice bank control

The refrigeration machine control is usually supplied with it by its manufacturer.

The main functions of the ice bank control are:

- Control of the glycol-water mixture's temperature during discharging in series
- Control of the diverting valve according to the operating mode
- Control of ice bank charging (possibly with consumption forecast)

7.6.1 Control of the glycol-water mixture's temperature

The three-port valve downstream from the ice bank (Fig. 7-13, A) mixes glycol-water from the ice bank (approximately 1 °C) and glycol-water from the refrigeration machine to the desired cooling temperature for the installed consumers.

This valve has no control function in bypass mode. The refrigeration machine supplies the glycol-water at the desired temperature.
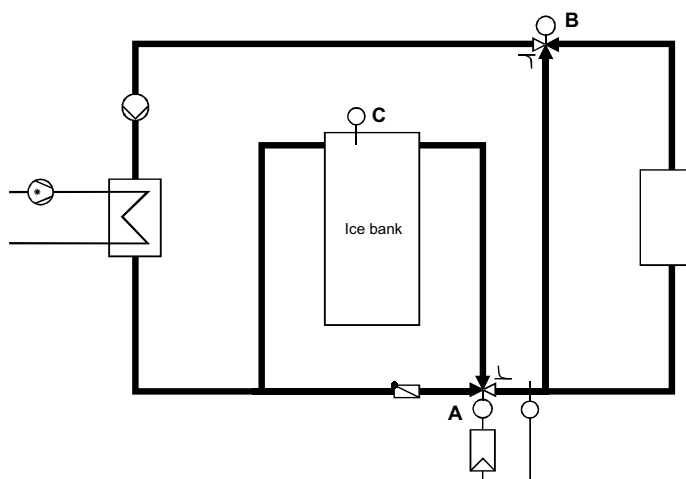

Fig. 7-13 Control and diverting valve for ice bank operation

7.6.2 Control of the diverting valve according to operating mode

The diverting valve (Fig. 7-13, B) is set to the desired position according to the operating mode. It is set to diversion for charging mode and to straight-through flow in discharging and bypass mode. It is normally installed in the return as a mixing valve and has no control function.

7.6.3 Ice bank charging control

Ice banks are normally equipped with a charge state sensor (Fig. 7-13, C), which delivers a standardized signal (DC 0…10 V or 4…20 mA). This information can be used for control purposes, e.g. to start and stop the charging process.

Expected cooling energy forecast

In the intermediate season in particular, it can be useful to not fully charge the ice bank. Therefore, an attempt is made in some systems to forecast the cooling energy required for the following day. Frequently used parameters for such a forecast are:

- Current and attenuated outdoor temperature
- Outdoor air humidity
- Room temperature progression
- Charge state progression over the previous days
- …

The parameters that are used depend very greatly on the actual plant. Additionally, it has been found in practice that overly complex forecast scenarios are not worthwhile.

7.7 Economic considerations

In the case of ice banks, not only the investment costs must be taken into account but above all the operating costs and the electricity supplier's tariff structures (high/low tariff, basic charges, peak demand charges, etc.).

Since these factors vary greatly from system to system, they must be analyzed in detail in each case.

It can be assumed that, in many systems, the combination of refrigeration machine and ice bank will give rise to lower investment costs than the solution with a refrigeration machine sized for maximum refrigeration output. Additionally, operating costs can also be reduced on the basis of the electricity supplier's tariff structures.

8. The absorption cycle

8.1 Introduction

An absorption process refers to the uptake of a gas by a liquid or solid substance in the form of a physical bond. However, absorption only takes place if the absorbing substance and the gas to be absorbed (working fluid pair) are chemically compatible and only at a given pressure/temperature ratio, which differs for each working fluid pair.

An absorption process is also reversible, i.e. the absorbed gas can be desorbed again at a different pressure/temperature ratio. Therefore, it can be operated as a closed-cycle process.
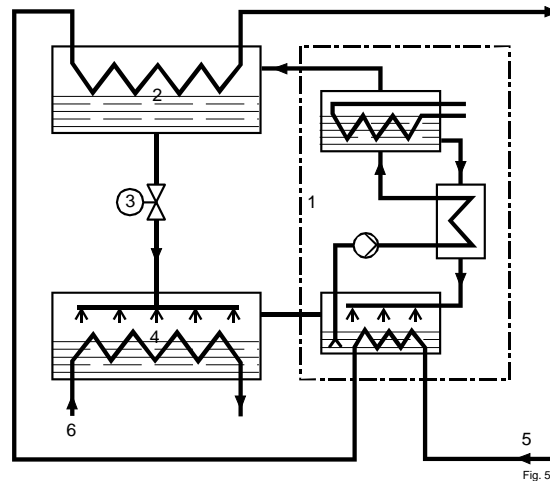

Fig. 50

Fig. 8-1   Absorption cycle with solvent cycle as a "thermal compressor"
   1 Thermochemical compressor
   2 Condenser
   3 Throttling/dosing device
   4 Evaporator
   5 Effective cycle as heat pump
   6 Effective cycle as refrigeration machine

A comparison of the refrigerant cycle of the absorption refrigeration machine with that of the compression refrigeration machine (chapter 4) allows the four functional elements to be identified at a glance:

- Evaporator (4)
- Compressor (1)
- Condenser (2)
- Throttling/dosing device (3)

In this case too, a pure refrigerant (e.g. water) is vaporized in the evaporator at low pressure with the supply of external heat. The vapor is compressed to a higher pressure and temperature. It is then liquefied in the condenser, where the latent heat of vaporization is transferred to an external coolant; and it is finally expanded to low pressure in the expansion valve.

The principle difference is in the compressor, which is replaced by the solvent cycle and has the following subfunctions:

- Instead of the suction process in the compressor, the low-pressure refrigerant vapor is absorbed by an appropriate liquid solvent or working fluid in the absorber
- Instead of the compression and discharge of the compressed, hot refrigerant gas from the compressor, the liquid solvent, which is enriched with refrigerant, is transported to the so-called expeller. In the expeller, external heat is applied to the solution. This raises the temperature and pressure of the solution. The refrigerant evaporates and flows through the hot-gas pipe of the refrigerant cycle to the condenser

Solvent cycle replaces compressor

In the absorption machine, therefore, the mechanical compressor is replaced by the solvent cycle. Therefore, the solvent cycle is referred to as a "thermochemical compressor".

Applied heat instead of mechanical drive energy

All other functional elements of the refrigerant cycle, such as the condenser, throttling/dosing device and evaporator, are the same as in the compression machine. Instead of the mechanical drive energy required by the compressor, however, the energy required to sustain the absorption cycle is supplied in the form of heat (steam, hot water, oil/gas burner, etc.). Mechanical energy is only required to drive the solvent pump.

8.2 Field of application of the absorption refrigeration machine

The field of application of absorption refrigeration machines covers practically the entire range of the piston and turbocompressor machines, i.e. from approx. 30 kW to over 5,000 kW of refrigeration capacity.

The choice between a compression machine and an absorption machine depends largely on the available operating energy type. If, for example, a steam or hot water boiler is available that would otherwise only be optimally utilized in winter, it is an obvious choice to use its free capacity in summer for refrigeration with an absorption refrigeration system.

The use of an absorption machine is optimal if waste steam is available from a production process or backpressure turbine. Another interesting application arises from the combination with a turbochiller. In this case, the turbocompressor is driven by a backpressure turbine. The low-pressure steam from the backpressure turbine is used to heat the expeller of the absorption machine before returning to the steam boiler as a condensate.

Absorption machines that are heated directly via oil or gas are usually configured as heat pumps, which can be switched over to cooling operation in summer.

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Finally, the decisive advantages of the absorption machine are its practically silent, vibration-free operation and simple output control between 0 and 100 %.

Disadvantages are the relatively high energy consumption, the high condenser output and, therefore, the high cooling water consumption. However, these disadvantages can frequently be compensated by the considerably reduced energy costs if waste heat is used.

8.3 Working fluid pairs

The currently best known working fluid pairs for absorption refrigeration machines / heat pumps are as follows:

– Water-lithium bromide (LiBr)   (with water as the refrigerant)
– Ammonia (NH_3)-water   (with ammonia as the refrigerant)

Other working fluid pairs that are used in special systems and are, therefore, less well known are as follows:

– Ammonia-lithium nitrate
– Methylamine-water
– Methanol-lithium bromide

where the substance named first is the refrigerant in each case.

Whereas the proven refrigerant ammonia (NH_3) is primarily used for vaporization temperatures between 0 °C and –60 °C, the working fluid pair water-lithium bromide (LiBr) is primarily used in the air conditioning field today. However, water only permits vaporization temperatures above 0 °C; otherwise it freezes.

Different working pressures for NH_3-water and water-LiBr

A further major difference between the ammonia-water cycle and the water-LiBr cycle is the different working pressures of the systems. Whereas the ammonia machines operate at pressures between approximately 1.5 and 16 bar, the working pressures in the evaporator and absorber of water-LiBr machines are considerably below atmospheric pressure. The evaporator pressure is around 0.008 bar, corresponding to a vaporization temperature of approximately 3 °C, and the condenser pressure is around 0.1 bar, corresponding to a condensation temperature of approximately 50 °C. Such low pressures (vacuum) require a very well sealed and sturdy machine design.

The absorption refrigeration machine / heat pump operates with two cycles which, although they interact at different phases, can be described separately in functional terms (Fig. 8-2). The two cycles are as follows:

• Refrigerant cycle
with the compressor, condenser, throttling/dosing device and evaporator
• Solvent cycle
which assumes the role of the compressor in the refrigerant cycle

8.3.1 Refrigerant cycle

The warm refrigerant vapor flows at condensation pressure out of the **expeller** of the solvent cycle and into the condenser; there it comes into contact with the coolant coils and condenses. The latent heat of vaporization that is released is transferred to the coolant, which is heated up by it. This heating of the coolant constitutes the actual heat output of the heat pump. In the case of the refrigeration machine, it is the waste heat that must be dissipated.

The liquid refrigerant, which is still at condensation pressure, is expanded to low pressure in the **throttling device** and then spayed over the pipe coils in the **evaporator.** The pressure in the evaporator is kept so low that the refrigerant (e.g. water) already begins to evaporate at +3 to +5 °C. If, therefore, chilled water at an average temperature of approximately +10 °C is circulating through the pipe coils, the refrigerant evaporates on contact with them. The necessary latent heat of vaporization is extracted from the chilled water cycle, which is cooled accordingly. This cooling constitutes the refrigeration output of the refrigeration machine. The low-pressure refrigerant vapor produced in this way flows back into the **absorber** of the solvent cycle.
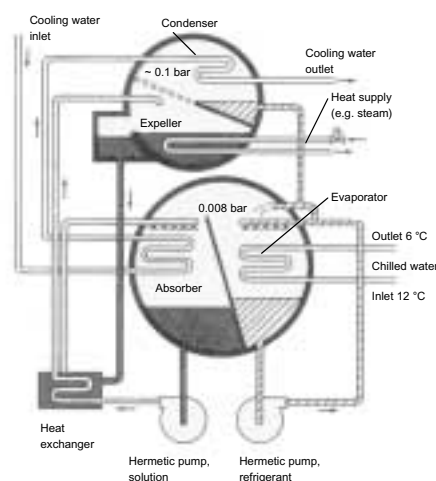

Fig. 8-2 Solvent and refrigerant cycles of the absorption process

8.3.2 Solvent cycle

The solvent cycle is explained here using the example of the working fluid pair water-LiBr (Fig. 8-2). LiBr reacts with water in the same way as common salt. In the **absorber,** the solvent is sprayed with a high concentration of LiBr. In this way, the refrigerant vapor (steam) from the **evaporator** comes into intensive contact with the solvent spray and is absorbed. During this chemical process, reaction heat is released, which must be dissipated. This is achieved via a pipe coil through which the same coolant is flowing that subsequently also provides the condenser cooling in the refrigerant cycle.

The solvent, which is now "diluted" with refrigerant and has a low LiBr concentration, is transported by the **solvent pump** through the **heat exchanger** to the **expeller.** External **heat is supplied** to the expeller (also known as the generator). The heat can be applied via steam, hot water or an electric heater or via direct heating by combustion of oil, gas or solid fuel. This heat supply causes the refrigerant to evaporate out of the solvent, giving rise to the necessary pressure and temperature increase. While the refrigerant vapor arising in this way flows into the **condenser** of the refrigerant cycle, the solvent, which now has a high LiBr concentration again, is directed back through the heat exchanger to the absorber, and the solvent cycle begins again.

In the **heat exchanger,** the cold solvent from the absorber is preheated by the warm solvent flowing back from the expeller, which represents **process-internal heat recovery.**

8.4 The closed-cycle process of the absorption machine

In the closed-cycle process of the absorption machine, not only the refrigerant but also the solvent circulates (working fluid pair). The concentration ξ (xi) of the refrigerant in the mixture is a major characteristic quantity of the respective operating state. Therefore, a log p-1/T diagram is used to represent the process (in contrast to the h-log p diagram in the compression cycle). In the log p-1/T diagram, the lines of equal concentration ξ are shown as almost straight lines.
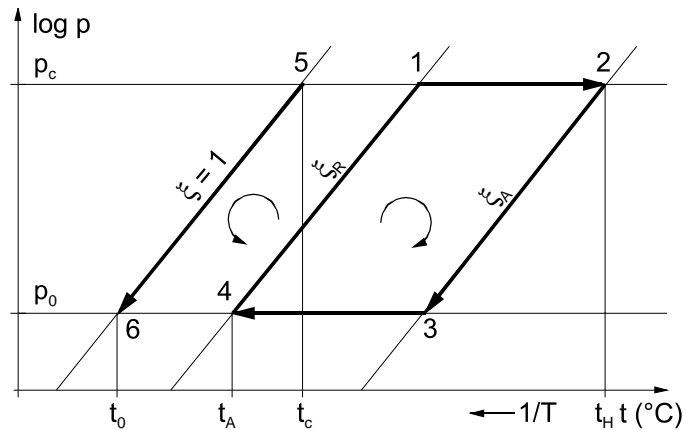

Fig. 8-3 The absorption refrigeration process on the log p-1/T diagram (simplified, without heat exchanger)

t_A = absorption temperature

t_H = expeller temperature

ξ = concentration

t_c = condensation temperature

t_o = vaporization temperature

4-1 = pump

1-2 = expeller

3-4 = absorber

6-4 = evaporator

1-5 = condenser

5-6 = throttling/dosing device (refrigerant is expanded)

Fig. 8-3 shows a schematic view of the simple process (without heat exchanger) on a diagram of this kind. The refrigerant-rich solution with concentration ξ_R enters the expeller (point 1). It is heated to the process temperature t_H (point 2), where the refrigerant is desorbed, and the concentration of solution decreases to ξ_A (low amount of refrigerant). The desorbed refrigerant reaches a concentration of practically $\xi = 1$ and is liquefied at the process temperature t_c (point 5).

The point of intersection of t_c with the line $\xi = 1$ determines the pressure p_c on the hot side of the process. From this pressure, the refrigerant is expanded via the expansion valve to pressure p_o, which is determined by the desired process temperature t_o on the cold side (point 6). The lean solution is also expanded to pressure p_o and enters the absorber (point 3). The absorption of the refrigerant vapor by the lean solution releases condensation and solution heat. Cooling to process temperature t_A (point 4) increases the absorptivity to concentration ξ_R, so the refrigerant vapor from the evaporator (point 6) is fully absorbed.

The pump raises the pressure of the refrigerant-enriched solution to pressure p_c (point 1) again.

The difference $\xi_R - \xi_A$, known as the degassing range, is determined by the available or permissible expeller temperature t_H and the achievable ultimate absorption temperature t_A, depending on the coolant. The smaller the degassing range, the greater the solution circulation required for 1 kg of pure refrigerant.
The degassing range decreases the greater the differential $t_c - t_o$ becomes at a given differential $t_H - t_A$. Therefore, as in the case of compression systems, two-stage or multistage processes are required for greater temperature differentials $t_c - t_o$.

The log p-1/T diagrams for the most commonly used working fluid pairs, water-LiBr and NH_3-water, are shown below.
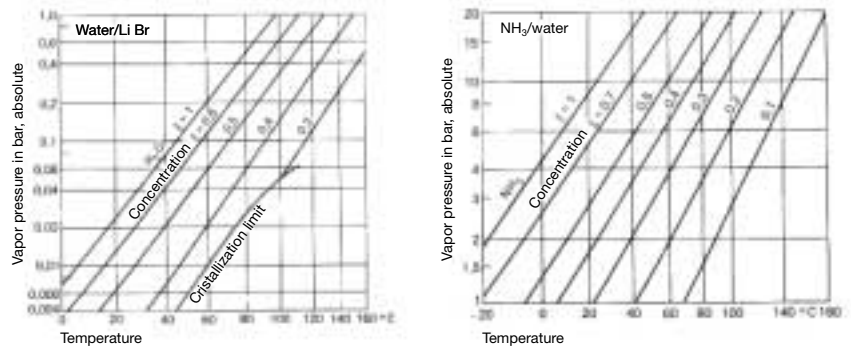

Fig. 8-4   Log p-1/T diagrams for the working fluid pairs water-LiBr and NH_3-water

8.5 Heat ratio ζ   The characteristic quantity for evaluating the absorption process is the co-called heat ratio ζ (zeta). It shows the relationship between the achieved refrigeration capacity and the heat supply (motive power).

The following applies to the refrigeration machine:

$$\text{Heat ratio } \zeta_K \;=\; \frac{\dot{Q}_o}{\dot{Q}_H} \;=\; \frac{\text{refrigeration capacity}}{\text{heat supply}}$$

The following applies to the heat pump:

$$\text{Heat ratio } \zeta_W \;=\; \frac{\dot{Q}_c + \dot{Q}_A}{\dot{Q}_H} \;=\; \frac{\text{heat output}}{\text{heat supply}} \;=\; \zeta_K + 1$$

$\dot{Q}_c$ = heat emission at the condenser
$\dot{Q}_A$ = heat emission at the absorber
$\dot{Q}_H$ = heat supply at the expeller
$\dot{Q}_o$ = refrigeration capacity

8.6 Design and equipment of the absorption refrigeration machine

LiBr machines are used almost exclusively for refrigeration in air conditioning systems today. Such machines normally have a two-vessel or single-vessel design (see 8.6.2).

In the two-vessel design, one vessel contains the absorber and evaporator, and the second vessel contains the expeller and condenser. In the single-vessel design, all four functional units are accommodated in one large vessel.

These two machine types are described in greater detail in the following sections.

8.6.1 Two-vessel design

The mechanical design of the two-vessel absorption machine is shown in Fig. 8-2 (schematic) and Fig. 8-5. The upper vessel contains the expeller and condenser, which operate at higher pressure, and the lower vessel contains the evaporator and absorber, which operate at lower pressure.

The two-vessel machine is mainly distinguished from the single-vessel type by the following design details:

– A separate fluid sump is required beneath each tube bundle
– Longitudinal slots in the two vessels are used to reduce the transfer of heat between the two process stages, which operate at different temperatures
– The difference in level between the two vessels provides for a natural flow of the liquid refrigerant from the condenser to the evaporator, on the one hand, and of the LiBr solution from the expeller to the absorber on the other
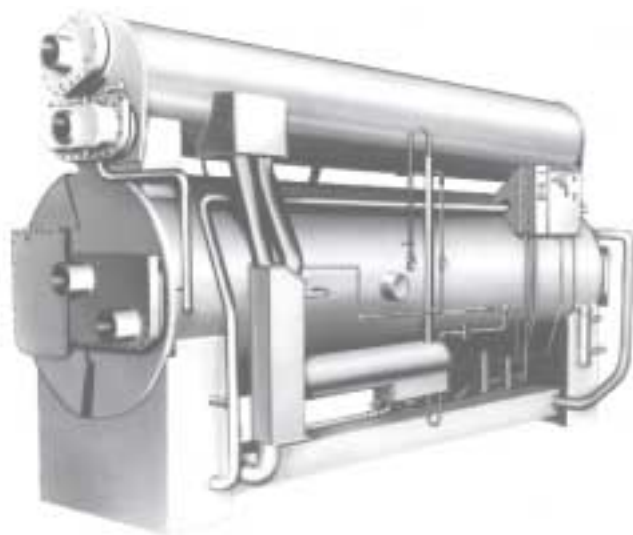

Fig. 8-5   Absorption refrigeration machine with two-vessel design (expeller and condenser above, absorber and evaporator below)

The greater overall height of the two-vessel design can be a disadvantage under certain circumstances. On the other hand, the two vessels can be transported and brought into the building separately.

The evaporator section of the lower vessel is insulated to prevent heat loss and condensation.

Since the temperature differential between standstill and operation can be more than 200 K in the tube bundles, only U-tube bundles that are rolled in on one side can be used, because this allows the differences in thermal expansion to be compensated without difficulty.

8.6.1.1 Deaeration system

The aqueous LiBr solution is highly aggressive in conjunction with oxygen. Furthermore, so-called "noncondensable" gases reduce the efficiency of the absorption process. Operation under vacuum allows the ingress of air through minute leaks. A sophisticated deaeration system provides for periodic removal of the noncondensable gases and also gives an indication of how well sealed the system is. Fig. 8-6 shows the design of such a deaeration system.
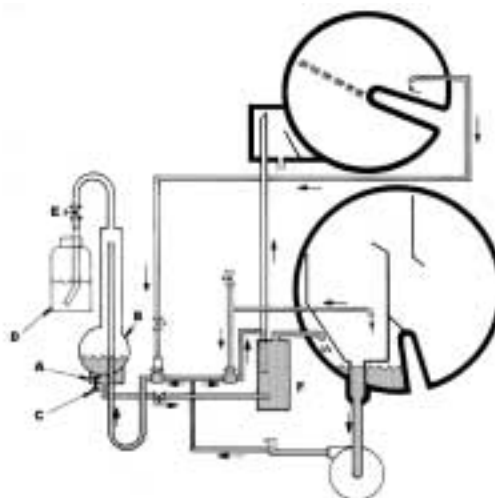

Fig. 8-6   Deaeration system of the two-vessel machine
   A Degasser
   B Storage container
   C Fill level indicator
   D Deaeration bottle
   E Deaeration valve
   F Secondary heat exchanger

8.6.1.2 Circulating pumps

Since shaft seals always offer the possibility for the ingress of air into the machine, solution and chilled water pumps must have a fully hermetic design. The pumps and their motors are cooled by the transported medium. Because of the corrosion hazard, the stator and rotor of the pump motor are enclosed in stainless sheet steel shrouds.

8.6.1.3 Cooling water   In LiBr absorption refrigeration machines, the cooling water temperatures should in principle be kept constant within given limits, because a rapid temperature drop – as is possible with cooling tower operation – can cause the solution to crystallize in the machine.

The water from the cooling tower is delivered to the absorber at a temperature of approximately 28 °C, leaves it at approximately 34 °C, then flows through the condenser and leaves it at approximately 38 °C. Appropriate cooling water control keeps the supply temperature constant at approximately 28 °C.

External cooling water control can generally be dispensed with in modern machines, because the cycle temperatures are automatically stabilized by an internal control system.

8.6.2 Single-vessel design   This machine has the same operating principle as the two-vessel design, and it is used in the same power ranges. In the single-vessel design, the four functional units – expeller, condenser, absorber and evaporator – are accommodated in one large vessel (Fig. 8-7). This very compact design gives rise to smaller external dimensions and lower manufacturing costs than with the two-vessel design.
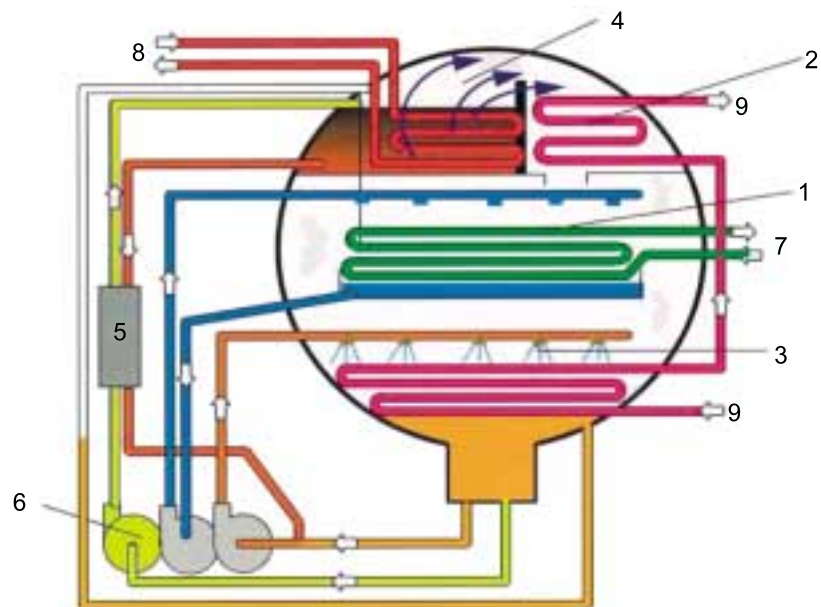

Fig. 8-7   Absorption refrigeration machine with single-vessel design (function schematic)
1 Evaporator
2 Condenser
3 Absorber
4 Expeller
5 Heat exchanger
6 Pumps
7 Chilled water circuit
8 Energy supply
9 Cooling water circuit

8.6.2.1 Deaeration   In principle, the same deaeration system could be used as in the two-vessel design. However, since advanced single-vessel machines are completely assembled at the production facility, have welded instead of flanged pipe connections, and are subjected to a helium leakage test, the ingress of air is less than with two-vessel machines. Therefore, a more simple deaeration system is sufficient. It consists of a so-called deaeration chamber that is inserted into the absorber section in such a way that it encloses part of the absorber tube bundle. In this way, the tubes enclosed by the deaeration chamber have less contact with the refrigerant vapor, which keeps the pressure and temperature in the chamber lower than in the rest of the absorber. Therefore, the noncondensable gases enter the deaeration chamber from where they are periodically extracted by a two-stage vacuum pump.

8.6.3 Two-stage absorption refrigeration machine   This machine is a further development of the designs presented above. As shown in Fig. 8-8, this machine has two expeller stages, where the desorbed steam from the first stage is used to heat the second stage before it is liquefied again in the condenser.
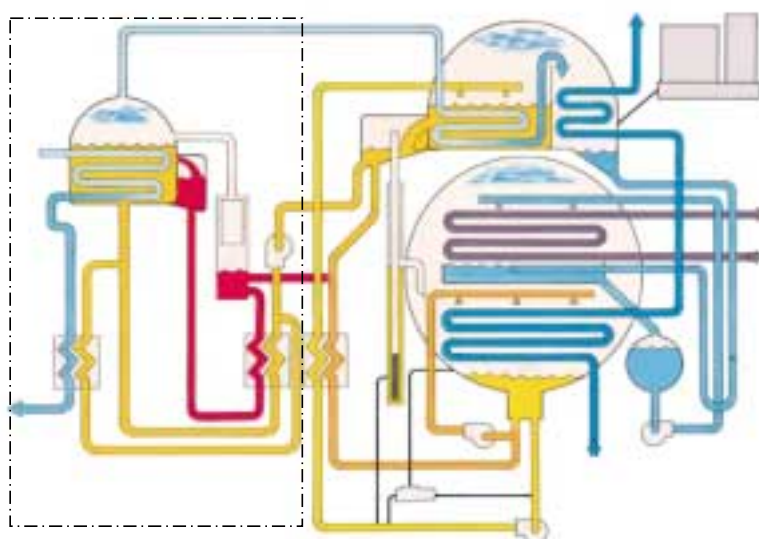

Fig. 8-8   Two-stage absorption refrigeration machine (2nd stage boxed on the left)

This two-stage circuit gives rise to considerable energy savings on the heating side and to reduced cooling water demand in the condenser. A saving of 25…30 % is possible. However, the first stage must be heated to a considerably higher temperature than 100 °C so that there is still a sufficient temperature gradient between the superheated refrigerant vapor and the solution in the expeller stage. Therefore, this machine should be operated with steam at approximately 9…11 bar, corresponding to a temperature of 170…180 °C.

8.7 Capacity control of the absorption refrigeration machine   The following two methods are used today to control the output of LiBr absorption machines:

- Throttling of the operating energy
- Bypass control of the solvent concentration

8.7.1 Capacity control by throttling the operating energy

The capacity of the absorption machine can be continuously controlled (e.g. according to the chilled water supply temperature) by throttling the energy supply to the expeller (see Fig. 8-2). If steam or heating water provides the operating energy, this is done using a normal flow control valve installed in the supply pipe. This control method allows the machine's capacity to be reduced to approx. 10 % of full load. At the same time, the energy supply also decreases almost proportionally to the cooling load.

8.7.2 Bypass control of the solvent concentration

This control method is based on changing the concentration of the LiBr solution, for which purpose a throttle valve is installed in the connecting pipe between the heat exchanger and the expeller. The throttling valve is controlled according to the temperature of the diluted solution. At partial loads, the diluted LiBr solution flowing to the expeller is throttled, and the retained portion of the solution is diverted to the absorber. The energy supply to the expeller is not throttled. Therefore, the reduced solution quantity due to the throttling is heated to a greater degree in the expeller, so it becomes more concentrated. The supply of solution to the expeller can be completely stopped, which reduces the machine's refrigeration capacity to zero.

Therefore, bypass control can be used over the full capacity range from 0 to 100 %. The energy consumption is slightly below the percentage load, because the heat transfer surfaces achieve a more favorable specific efficiency at partial loads.

Therefore, this control method gives rise to a slightly lower energy consumption in partial load operation. However, the relatively high purchase costs of the three-port valve restrict this type of control to absorption machines with relatively high refrigeration capacities.

In many cases, the most economical type of capacity control is achieved with a combination of the two control methods.

The information in this document contains general descriptions of technical options available, which do not always have to be present in individual cases. The required features should therefore be specified in each individual case at the time of closing the contract.

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