

**Refrigeration/
Air-Conditioning**

Training Manual

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Introduction

Contents and Purpose

This training manual provides technical guidance to service personnel performing work on refrigeration systems installed in trucks, passenger cars and busses.

Based on the most important thermodynamic fundamentals, first the functions and operating principle of the four principal component of the refrigeration system (compressor, condenser, throttling devices, evaporator) are described. Building on this, the construction and functions of the main components and all other parts of a refrigeration system are explained in the chapter "Components". Under the topic Anlagentechnik the issues of the safety chain and the various possibilities of capacity control are dealt with. The chapter "Commissioning" contains the required tools and aids for commissioning as well as important notes on the correct procedure of placing a refrigeration plant into service. The chapter "Troubleshooting" provides details on the basic practice of the performance of fault diagnosis activities as well as a list of possible causes of malfunctions.

Suggestions for improvements and modifications

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1 Fundamentals of Thermodynamics

1.1 Introduction

Refrigeration technology is the branch of thermodynamics concerned with the properties of solid, liquid and gaseous matter.

“thermo“ heat
 “dynamics“ the study of motion

Thermodynamics deal with “movements of heat” (supply and rejection of heat). Strictly speaking, the term “cold” does not exist in thermodynamics since “cold” only results from the fact that heat is transported from one place (having higher temperature) to another place (having lower temperature). When so doing, the cooled space, substance or body still contains heat provided the temperature lies above $-273.15\text{ }^{\circ}\text{C}$.

1.2 Temperature

The **temperature** of a substance can be understood to be a measure for the intensity of the movements of the constituents of matter (atoms, molecules and groups of molecules).

In the case of solids, lattice atoms or molecules vibrate around a vibration center. When this motion is increased through the application of heat to such an extent that the binding energy of the lattice structure is overcome, the rigid lattice collapses. The solid starts to melt and becomes liquid. During this process there are still binding forces acting underneath the lattice. These forces are overcome as more heat energy is applied, and the molecules can move freely in the vapor or gaseous state. These changes in state can be reversed by heat absorption.

For the **temperature measurement** all temperature-dependent physical properties of the bodies (volume expansion, electric resistance, etc.) are suitable.

Most **thermometers** are based on the thermal expansion of liquids such as mercury and alcohol. A temperature scale needs to be established.

Since in nature given physical processes always take place at the same temperature under the same conditions, fixed points are obtained among which the melting point of ice and the boiling point of water are best known.

A **temperature scale** is obtained by subdividing the expansion of a liquid between two fixed points in a certain way. The Swedish astronomer A. Celsius (1701 - 1744) devised a temperature scale in 1742, the so-called **Celsius scale**. He used mercury as the liquid and put 100 evenly spaced marks between those two points. The magnitude of one degree Celsius was thus defined. By uni-

formly extending the Celsius scale beyond the boiling point and below the freezing point one obtains a temperature scale that is valid for all temperature ranges. As you can see, the establishment of such a temperature scale was quite arbitrary.

The lowest temperature attainable is $-273.15\text{ }^{\circ}\text{C}$. This value is also referred to as **absolute zero**. The thermodynamical temperature scale starts with absolute zero at 0 K (**Kelvin**, named after the English physicist W. Thomson, the later Lord Kelvin, 1824 - 1904). Based on this fact, a new basic quantity for the temperature, the **thermodynamic temperature** was introduced.

The absolute zero can never be attained by conventional heat transfer owing to the fact that the heat contained in any body can only be given off to a body with a lower temperature.

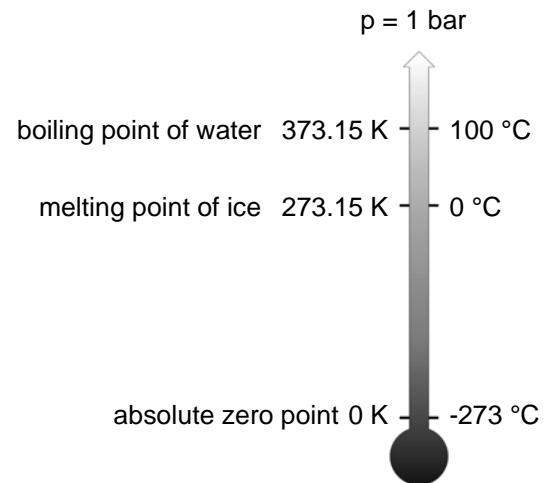


Fig. 1-1. Kelvin and Celsius temperature scale

1.3 Pressure

The term “**pressure**” refers to the force applied to a unit area of surface. The Comité International des Poids et Mesures defined that the **unit of pressure** is **Pascal (Pa)**. This unit is also referred to as Newton per m^2 (N/m^2).

$$\text{Pressure } p = \frac{\text{Force } F}{\text{Area } A} \quad \text{in Pa, N/m}^2$$

(Pascal, named after B. Pascal, French philosopher and mathematician, 1623 - 1662)

(Newton, named after I. Newton, English physicist, 1643 - 1727)

The legal unit “**bar**” derived from “Pascal” is also an approved unit of pressure.

The following applies:

$$1 \text{ bar} = 100\,000 \text{ Pa} = 10^5 \text{ Pa}$$

$$1 \text{ bar} = 1\,000 \text{ mbar}$$

1 Fundamentals of Thermodynamics

1.3.1 Atmospheric pressure

Atmospheric pressure p_{amb} is measured by means of a **barometer**.

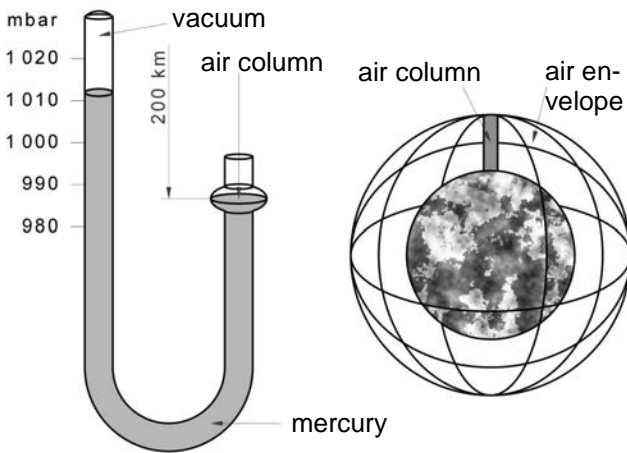


Fig. 1-2. Air pressure measurement by means of a barometer

The earth is surrounded by an envelope of air extending to a height of 200 km, with the air density (and hence also the atmospheric pressure) decreasing with altitude. The average sea level atmospheric pressure is $p_{amb} = 1.013$ bar corresponding to a mercury column of 760 mm.

The atmospheric pressure also changes as a result of climatic influences in the atmosphere:

- When air is heated, it expands and rises; at a given altitude the air will push sideways, the air pressure decreases.
- Upper atmosphere cooling causes the hot air masses to become more dense and sink back to the earth. The air pressure increases.

1.3.2 Gauge pressure and absolute pressure

Most **pressure gauges** used in practice are (for cost reasons) overpressure gauges. When they measure the pressure, the quantity obtained usually excludes the ambient atmospheric pressure and is called **overpressure p_e** . To obtain the **absolute** (“real”) **pressure p_{abs}** the ambient pressure p_{amb} must be added to the pressure gauge readout (p_e). The **absolute pressure** is required for calculations and the determination of physical properties.

$$p_{abs} = p_{amb} + p_e \quad \text{in bar}$$

p_{abs}	absolute pressure
p_{amb}	atmospheric pressure
p_e	overpressure (gauge pressure)

When a refrigeration system is evacuated the gauge pressure measured will be a negative value. When the meas-

urement is taken in relation to the surroundings the readout must be 0 bar.

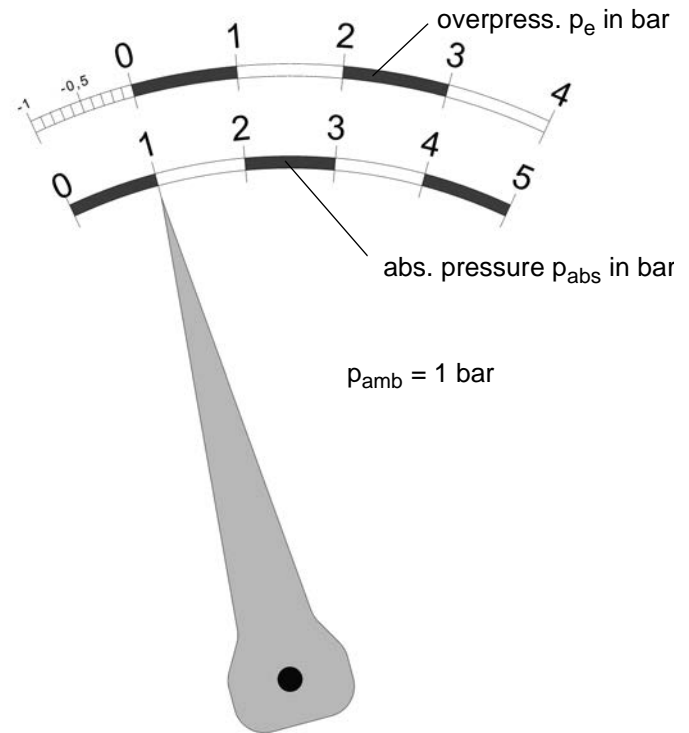


Fig. 1-3. Pressure gauge readout, absolute pressure and overpressure

Example

A newly installed refrigeration system which has neither been evacuated nor charged with refrigerant has an atmospheric pressure of approximately $p_{amb} = 1$ bar.

Charging the plant with the refrigerant will result in an overpressure p_e . What is the absolute pressure in the system if it was filled up to an overpressure of $p_e = 3$ bar?

$$p_{abs} = p_{amb} + p_e = 1 \text{ bar} + 3 \text{ bar} = 4 \text{ bar}$$

Evacuating the plant results in an “underpressure” (negative overpressure). What is the absolute pressure in the system if it was evacuated down to a pressure below the atmospheric pressure of $p_e = -0.6$ bar?

$$p_{abs} = p_{amb} + p_e = 1 \text{ bar} - 0.6 \text{ bar} = 0.4 \text{ bar}$$

1.4 Enthalpy

Enthalpy H used to be termed “heat content“. Enthalpy indicates how much heat (relative to a given temperature) a substance contains. The enthalpy value “zero“ is in most of the cases adapted to the Celsius temperature scale for practical reasons. In the case of water vapor it is commonly based on the temperature of 0 °C (273.15 K). The **specific enthalpy h** is given with respect to 1 kg of a given substance.

This means that water of 0 °C has the specific enthalpy $h = 0$ J/kg. Above 0 °C the value for the enthalpy is positive, below 0 °C it is negative.

For refrigerants, specific enthalpy values at different states or temperatures can be taken from tables and diagrams.

1.5 Density and specific volume

Density ρ is the mass of a substance given with respect of a volume of 1 m³.

$$\text{Density } \rho = \frac{\text{Mass } m}{\text{Volume } V} \quad \text{in kg/m}^3$$

The **specific volume v** is the relationship between the volume of a substance and a mass of 1 kg.

$$\text{spec. volume } v = \frac{\text{Volume } V}{\text{Mass } m} \quad \text{in m}^3/\text{kg}$$

1.6 Phase changes of substances

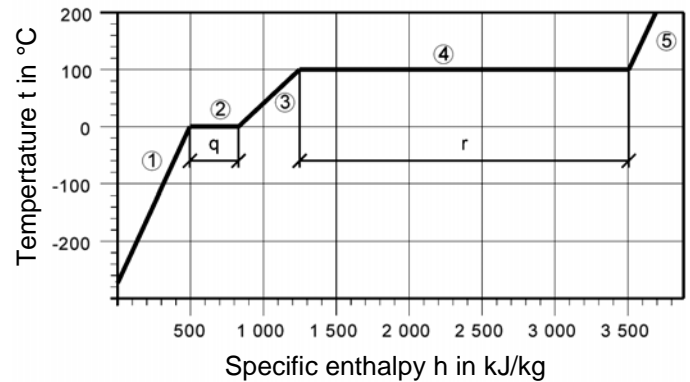
A substance will undergo several **phase changes** when it is subjected to constant heating or cooling. The **state of a substance** depends on its temperature and the pressure exerted on it. When a phase change occurs the temperature remains constant until all of the substance has changed its state.

Example

Let us assume that heat is constantly applied to a solid. First, the temperature of the substance rises. Once the **melting point** has been reached, the temperature remains constant and the substance starts to melt. The thermal energy required to melt a substance is referred to **heat of fusion q** . After all molecules of the substance have overcome the binding energy of the lattice structure and are able to move freely within the melting mass the temperature rises when more heat is applied. The temperature continues to rise until another change in phase (the

transition from the liquid to the gaseous state) occurs. This process is called **vaporization** and the required energy **vaporization heat r** . Also in this phase change the enthalpy of a substance increases whereas the temperature remains constant.

The vaporization heat is significantly greater than the heat of fusion due to the greater binding forces existing in a liquid.



- q Heat of fusion
- r Vaporization heat
- 1 Ice
- 2 Ice and water
- 3 Water
- 4 Water and steam (wet steam)
- 5 Superheated steam

Fig. 1-4. Temperature enthalpy diagram for water at $p = 1.013$ bar

Every substance can exist in three states (solid, liquid and gaseous). Highly superheated vapor is called “gas“ (e.g. at ambient conditions air exists in a highly superheated state). To this date no technical set of rules has defined the superheating state from which onward one speaks of “gas“ instead of “vapor“.

1.6.1 Phase diagram

At sea level water vaporizes at a temperature of 100 °C, at an ambient pressure $p_{\text{amb}} = 1.013$ bar.

If thermal energy is applied to the water on a mountain it vaporizes at a lower temperature. On a 2 000 m high mountain the ambient pressure p_{amb} is approx. 0.8 bar. The vaporization temperature of water at this altitude is approx. 93.5°C.

The lower the pressure exerted on the substance is, the easier it is for the molecules to overcome the binding energy and the lower the vaporization temperature is.

1 Fundamentals of Thermodynamics

Phase diagrams are used to represent the temperatures and pressures affecting the different phase changes of a substance.

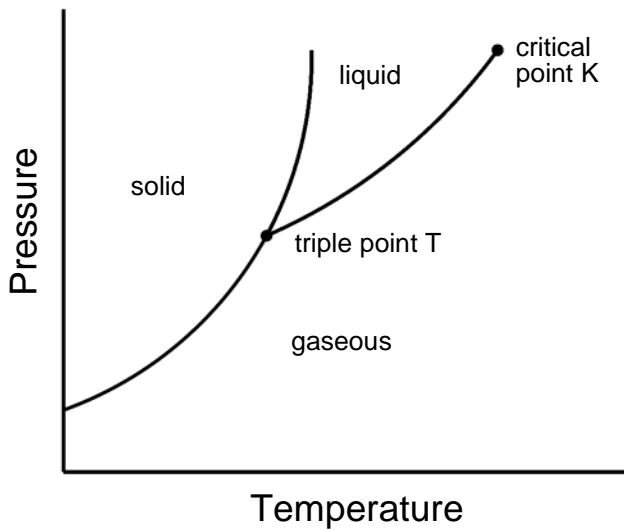


Fig. 1-5. Phase diagram of a refrigerant

The three boundary lines of the diagram delineate areas in which only the solid or only the liquid or only the gaseous phase can exist. In the point where all three areas border on each other, the triple point, all three phases co-exist. In the points on the curves, two phases exist together. The curve section between the triple point T and the critical point K is called **vapor pressure curve** of the liquid or also the **boiling curve**.

As the phase changes of a substance are very important in the area of refrigeration technology we would like to describe this in more detail taking water as an example.

Along the **vapor pressure curve** water and water vapor exist together. Upon the transition from water to water vapor a large amount of heat needs first to be supplied so that all liquid molecules can overcome their binding forces. During this phase it happens that one part is already in a vapor state and the other part still liquid.

The vapor coming into existence at the beginning of vaporization is called **wet vapor** given the fact that it still contains moisture. Vapor that contains no liquid particles and whose temperature equals that of the boiling temperature, is called **saturated vapour**. If heat continues to be applied and the vapor temperature rises further, the vapor is called **superheated vapor**.

The phase changes described above occur in the reverse order of sequence when superheated vapor is cooled. The amount of energy required for vaporization, i.e. the **evaporation enthalpy**, is identical to the amount of energy released during condensation.

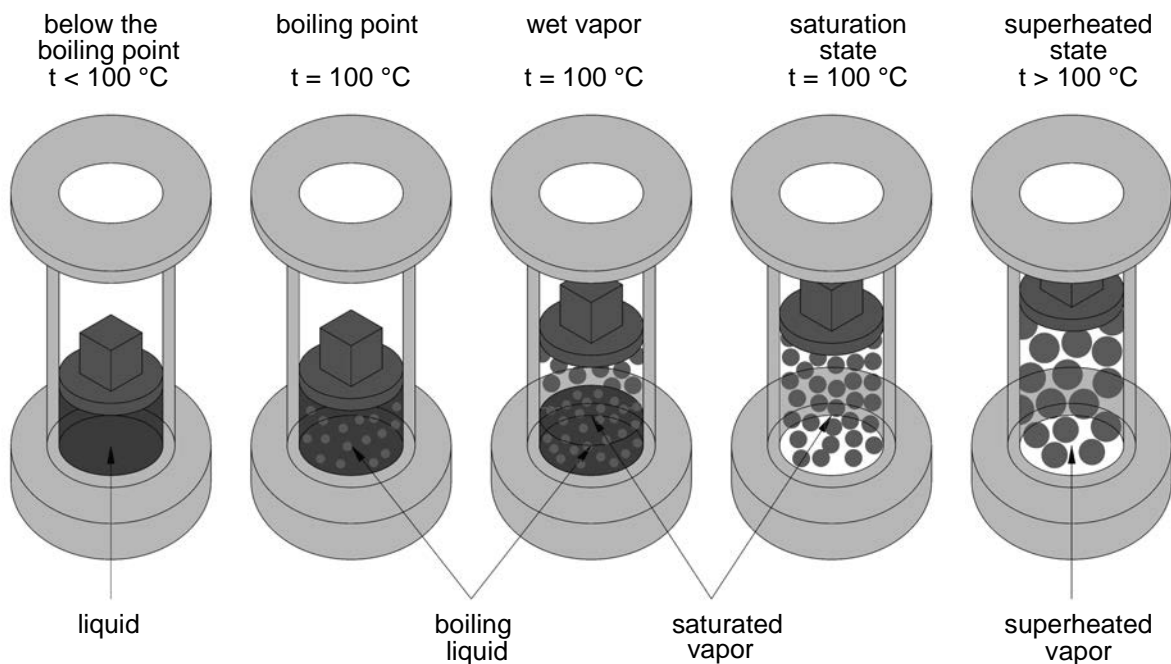


Fig. 1-6. Vaporization of water at constant pressure ($p = 1.013 \text{ bar}$)

1.6.2 Vapor Pressure Curve

The relationship between pressure and boiling temperature is represented in **vapor tables** and **vapor pressure curves** for the most important substances in the field of refrigeration technology. Fig. 1-7 represents a vapor pressure curve for the refrigerant R 134a.

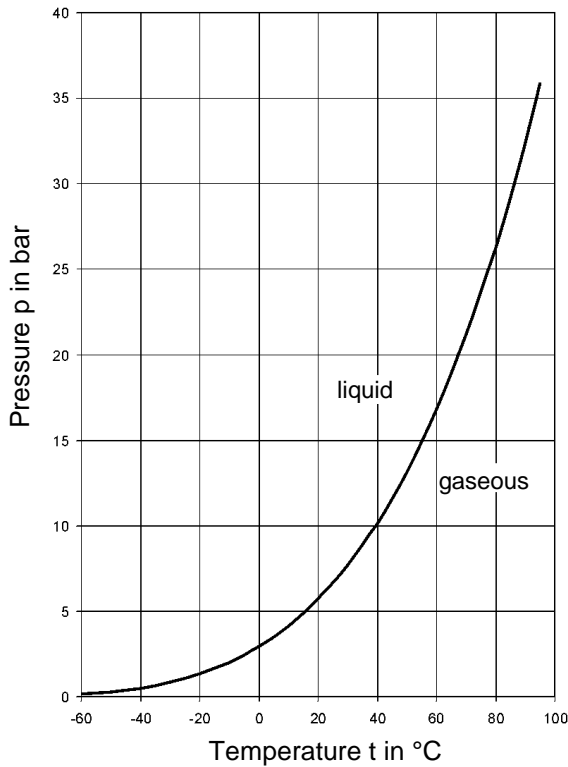


Fig. 1-7. Vapor pressure curve of R 134a

Table 1-1 shows part of a wet vapor table for R 134a:

Table 1-1. Wet vapor table for R 134a

Saturation temperature t in °C	Gauge pressure (pressure gauge readout) p_e in bar	Absolute pressure ($p_{amb} = 1$ bar) p in bar
-20	0.33	1.33
-10	1.01	2.01
0	1.93	2.93
10	3.15	4.15
20	4.72	5.72

These tables are used to determine the vaporization and condensation pressures by means of pressure measurements. **It is imperative to keep in mind that the pressure values indicated are always absolute pressures!**

A detailed vapor table for R 134a is contained in the appendix.

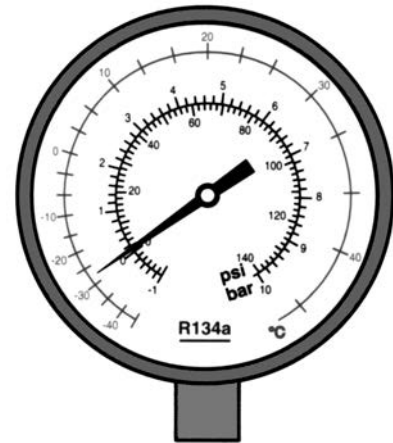


Fig. 1-8. Pressure gauge with saturation temperature scale

Apart from the pressure scale many pressure gauges also feature a saturation temperature scale. This allows the saturation temperature to be read off without any further conversions. The temperature scale applies only at an ambient pressure of one bar and only to the refrigerant indicated! For the pressure gauge illustrated above this is R 134a. Low-pressure gauges are surrounded by a blue frame, high-pressure gauges by a red frame.

Example

If the pressure gauge readout (gauge pressure) at the evaporator outlet indicates a value $p_e = 1.93$ bar, the value of the ambient pressure (at sea level $p_{amb} = 1$ bar) must be added to determine the evaporation temperature. You then obtain an absolute pressure of $p = 2.93$ bar that corresponds to an evaporation temperature of 0 °C, as shown in the table.

Example

A refrigerant cylinder (containing wet vapour) filled with R 134a is connected to a pressure gauge.

What gauge pressure and what absolute pressure prevail in the cylinder at an ambient temperature of 20 °C and an ambient pressure of 1 bar?

Absolute pressure p : 20 °C correspond to 5.72 bar
Gauge pressure p_e (readout):

$$5.72 \text{ bar} - 1 \text{ bar} = 4.72 \text{ bar}$$

The same cylinder will now be transported on top of a mountain with an altitude of $2\,000$ m (ambient pressure 0.8 bar). What gauge pressure and what absolute pressure will now prevail in the cylinder at an ambient temperature of 20 °C?

Absolute pressure p : 20 °C correspond to 5.72 bar
gauge pressure p_e (readout):

$$5.72 \text{ bar} - 0.8 \text{ bar} = 4.92 \text{ bar}$$

1 Fundamentals of Thermodynamics

2 Refrigeration Cycle - Fundamentals

2.1 Utilization of a substance's thermodynamic properties for "Refrigeration Purposes"

As already mentioned, it is not possible to produce "coldness". If a location is to be cooled, "heat" must be transferred from the location to be cooled to another location.

How is this carried out in a refrigeration system?

The fact that the heat content of a substance greatly increases during the fusion and vaporization process, with the temperature remaining constant, is utilized in the refrigeration system. The simplest method to dissipate "heat" from a location would be by means of an ice block, cf. Fig. 2-1. When the ice melts it absorbs the heat flow from the surroundings and the food and rejects it - along with the melted ice flowing out of the "ice box" - to the surroundings.

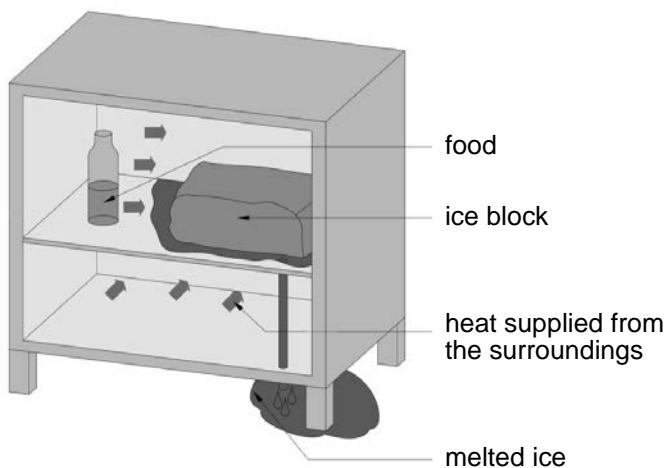


Fig. 2-1. Heat rejection by means of the melt enthalpy of ice: "Ice Box"

Since the enthalpy of evaporation is many times greater than the enthalpy of fusion, a greater heat flow can be absorbed during the evaporation process, at constant temperature. It is therefore recommended that the heat transport should take place at the boiling point of a substance. This is used by the compression refrigeration system to its advantage.

2.2 The basic refrigeration cycle

We would like to explain the functioning principle of the **compression refrigeration system** with its four basic components (compressor, condenser, evaporator and throttling valve) taking a refrigerator as an example.

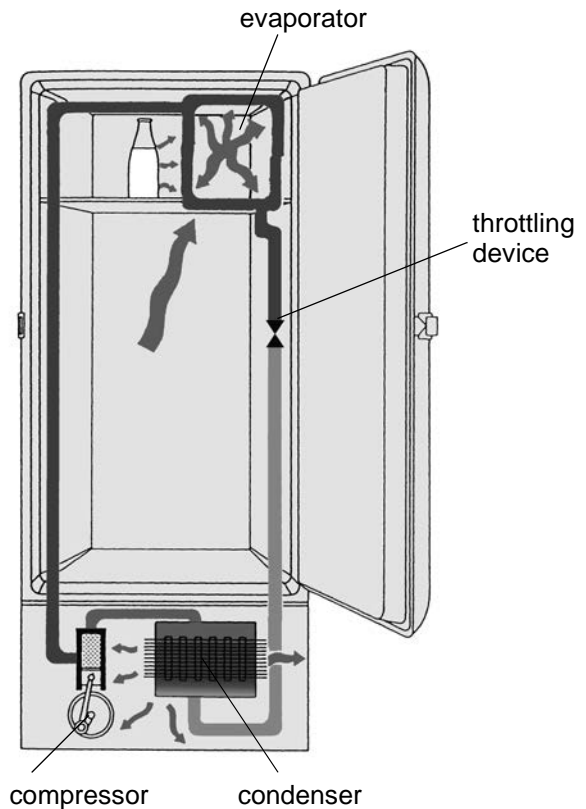


Fig. 2-2. Simplified representation of a refrigeration cycle: "Refrigerator"

A refrigerator contains a **heat exchanger (evaporator)**, into which wet vapor is injected. At a temperature of +5 °C inside the refrigerator, the boiling or evaporation temperature of the refrigerant in the evaporator is approx. -15 °C, which in the case of refrigerant R 134a corresponds to an absolute pressure of 1.7 bar. Heat from the refrigerator interior is absorbed by the much colder evaporator causing the liquid part of the refrigerant to evaporate. The interior of the refrigerator is thus cooled.

The **compressor** draws in the refrigerant vapor from the evaporator and feeds it to a second heat exchanger, the condenser, located on the outside of the refrigerator cabinet.

The refrigerant undergoes a phase change from the vapor to the liquid phase in the **condenser**. In most of the cases, the temperature surrounding the condenser (room temperature) is approx. 20 to 25 °C. To ensure an adequate heat flow from the condenser to the surroundings, the condensing temperature (acc. to values drawn from past experience with refrigerators) needs to be approx. 20 to

2 Refrigeration Cycle - Fundamentals

30 K above the ambient temperature. Based on the refrigerant R 134a and an assumed condensing temperature of 50 °C the absolute pressure in the condenser is 13.2 bar.

The **compressor's** task is therefore not only to remove the evaporated refrigerant from the evaporator, but it must also compress it to a higher pressure.

A **throttling device** (in the case of refrigerators a capillary tube) is located between the condenser and the evaporator which expands the refrigerant from the condensing pressure to the evaporation pressure. This completes the refrigeration cycle.

Fig. 2-3 shows a schematic diagram of the cycle of a simple refrigeration system (e.g. a refrigerator). The four main components and the associated line sections are identified.

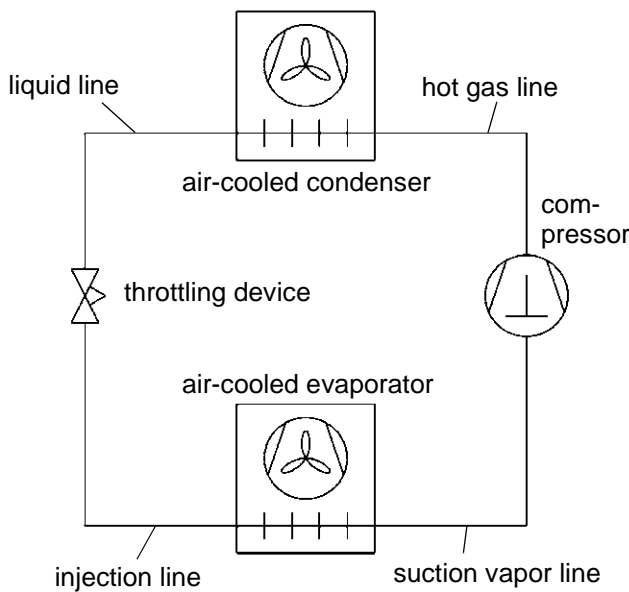


Fig. 2-3. Block diagram of a compression refrigeration system with the four principal components

Brief overview of the main components of a cold vapor compression refrigeration system:

Evaporator

The refrigerant evaporates at a low pressure and a temperature that lies below the ambient temperature of the evaporator thus absorbing heat from the surroundings.

Compressor

The refrigerant vapor removed from the evaporator at low pressure is compressed to a higher pressure and thus a higher temperature. To this end, additional heat is supplied to the refrigerant.

Condenser

The refrigerant is condensed at a temperature that lies above the ambient temperature (higher pressure). When

so doing, the entire heat absorbed in the evaporator and compressor is rejected to the surroundings.

Throttling device

The refrigerant at condensing pressure level expands in the throttling valve to evaporation pressure level.

The processes taking place in the refrigeration cycle can be graphically illustrated in a **temperature enthalpy diagram** (t, h -diagram). In Fig. 2-4 the different phases of the refrigerant as well as the boundary curves (boiling and dew point line) are shown.

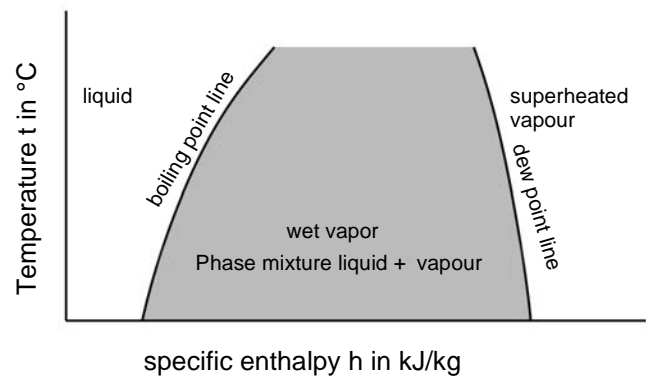


Fig. 2-4. t, h -diagram

The processes taking place at constant temperature (evaporation, condensation) are plotted in the t, h -diagram in horizontal direction, processes at constant specific enthalpy (no heat supply or heat rejection, expansion) are plotted in vertical direction.

In Fig. 2-5 a cyclic process of a compression refrigeration system (without superheating and subcooling) is shown in the t, h -diagram.

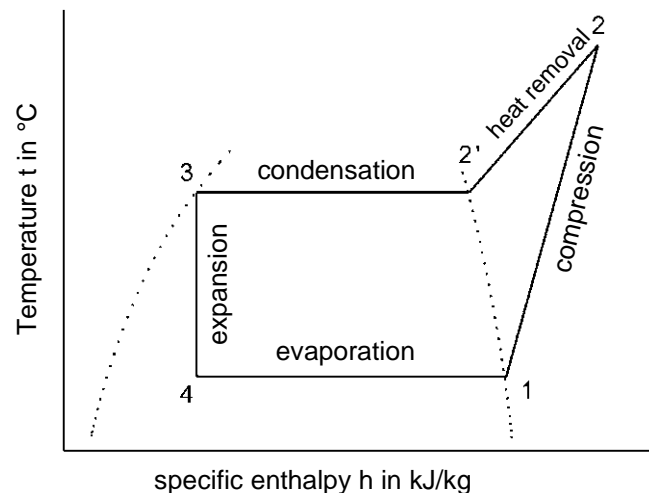


Fig. 2-5. t, h -diagram of a refrigeration cycle

The following processes take place:

4 → 1: In the evaporator the refrigerant is evaporated at a constant evaporation pressure p_o and a constant evaporation temperature t_o . (wet vapor area!)

1 → 2: Dry saturated refrigerant vapor is compressed in the compressor to the condensation pressure level p_c . In point 2 the superheated refrigerant vapor is at the final compression temperature t_{V2h} .

2 → 3: In the condenser the refrigerant is first cooled down to condensing temperature (process 2 → 2' heat removal). Subsequently, the vapor is condensed. Both processes take place at constant condensation pressure p_c .

3 → 4: In the throttle valve, liquid refrigerant expands to the lower evaporation pressure level p_o or the evaporation temperature t_o . The process takes place at constant enthalpy (heat constant).

2.3 Superheat

Due to its construction, a refrigeration compressor can only handle gaseous or vapor substances. Liquids cannot be compressed and therefore must not enter the compression chamber of the compressor.

If the suction state of the compressor lies directly on the dew point line (see item 1, Fig. 2-5), a decrease in evaporator load may result in "wet suction" (drawing in liquid refrigerant). This may result in damage to the compressor. The liquid refrigerant washes off the lubricant film existing between piston and cylinder wall; insufficient lubrication will result in undue wear. The presence of liquid refrigerant in the cylinder causes a direct energy transfer from the piston to the cylinder cover to occur during the compression cycle. The valve plate may be destroyed by so-called liquid slugging.

To prevent the compressor from drawing in liquid, the suction state of the compressor is shifted from the dew line to the right. The refrigerant vapor is "**superheated**". Its temperature t_{o2h} lies somewhat above the evaporating temperature t_o . The **superheat** is calculated as follows:

$\Delta t_{o2h} = t_{o2h} - t_o$	in K
----------------------------------	------

Δt_{o2h}	Superheat at evaporator outlet	in K
t_{o2h}	Refrigerant temp. at evaporator outlet	in °C
t_o	Evaporating temperature	in °C

The letter "h" stands for "superheated".

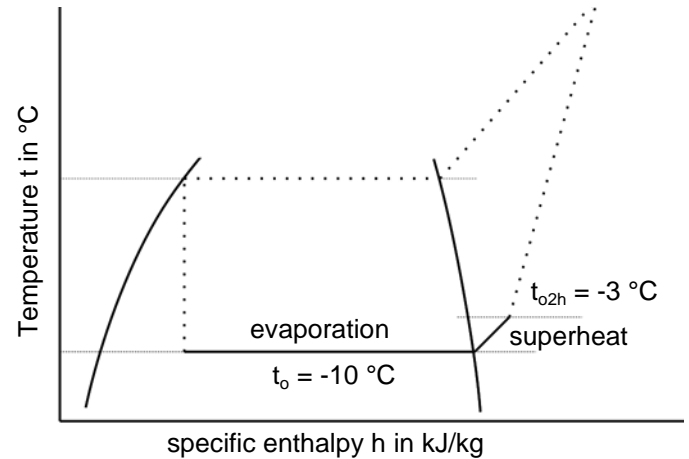


Fig. 2-6. Refrigerant superheat in the evaporator

Superheating: $\Delta t_{o2h} = t_{o2h} - t_o$

$$\Delta t_{o2h} = -3 \text{ °C} - (-10 \text{ °C}) = 7 \text{ K}$$

Apart from protecting the compressor from liquids, superheating has further advantages. Since liquid particles in the suction vapor result in a reduction of the delivery rate of the compressor, it can be increased by superheating. In addition, oil return is improved by superheating as well.

2.3.1 How is superheat produced?

Superheat in the evaporator

Most refrigeration plants with dry vaporization are equipped with thermostat-controlled expansion valves. They serve to regulate superheating downstream of the evaporator in all operating states. Part of the evaporator surface (approx. 20 %) serves for superheating.

Superheat in the suction line

If rather long suction lines are routed through areas with higher temperatures ($t_{amb} > t_{o2h}$) (e. g. machine rooms), additional superheating results from heat absorption from the ambient air. The refrigerant temperature cannot get higher than the ambient temperature. Superheating in the suction line does provide reliable protection of the compressor against ingress of liquids.

Superheat in the compressor

With "suction vapor cooling" the suction vapor is used for cooling the compressor motor. The refrigerant is additionally superheated before it enters the compression chamber. Superheating is also attained by internal heat transfer between hot gas and suction sides.

2 Refrigeration Cycle - Fundamentals

2.3.2 Typical superheat values

The optimum superheat value is approx. 5 - 8 K.

Maximum plant performance is reached at this superheating value. The thermostatic expansion valve, however, cannot accurately regulate this value. Depending on the design and the operating conditions, superheating fluctuates within a range of 4 - 12 K. Heating of the suction line caused by the ambient air results in additional superheating.

Note: Tests have shown that even with a superheating value of 7 K and more small amounts of liquid droplets from the compressor are entrained in the vapor. These, however, have no adverse effect.

2.3.3 Superheat measurement

You are to measure superheating in a refrigeration plant operated with the refrigerant R 134a.

- What measuring instruments do you need?
- Where do you measure which magnitude?
- The readout on a pressure gauge fixed directly to the evaporator is $p_{e0} = 1.7$ bar. What is the vaporization pressure p_o ?
- What is the vaporization temperature t_o ?
- At the evaporator outlet the temperature sensor measures a temperature $t_{o2h} = +3$ °C. What is the evaporator superheat Δt_{o2h} ?
- Evaluate the superheating value determined.

Solution

- Pressure gauge, thermometer, (vapor chart)
- Vaporization pressure p_o and temperature t_{o2h} are measured at the evaporator outlet.
- $p_o = p_{e0} + p_{amb} = 1.7 \text{ bar} + 1 \text{ bar} = 2.7 \text{ bar}$
- With $p_o = 2.7$ bar a vaporization temperature t_o of -2.2°C can be taken from the vapor pressure table for R 134a (see appendix).
- $\Delta t_{o2h} = t_{o2h} - t_o = 3 \text{ °C} - (-2.2 \text{ °C}) = 5.2 \text{ K}$
- The superheat value determined lies within the usual margin of 4 - 12 K.

2.4 Subcooling

The purpose of the expansion valve is to throttle the refrigerant after its condensation to a lower pressure level (vaporization pressure). For an optimum functioning of the valve pure liquid must be present at the inlet.

Figuratively speaking, the refrigerant must “squeeze” through a narrow (throttle gap). If you compare a given amount of refrigerant in the liquid and vapor state (at constant pressure), the vaporous refrigerant has a considerably greater volume. Consequently, the vaporous refrigerant needs more time to “squeeze” through the “narrow”.

Vaporous refrigerant in front of the expansion valve reduces the flow rate resulting in an under-supply of the evaporator with refrigerant. The vaporization pressure and the evaporator performance decrease.

If the refrigeration plant is operated in such a way that the state “inlet of expansion valve” lies directly on the left boundary curve (boiling line) a formation of bubbles in front of the expansion valve may occur at the slightest variation in operating conditions.

For this reason, the state “inlet of expansion valve” is shifted from the boiling line into the liquid region and this is then referred to as **subcooling**. Subcooling guarantees a liquid seal in front of the expansion valve.

Subcooling is calculated as follows:

$$\Delta t_{c2s} = t_c - t_{c2s} \quad \text{in K}$$

Δt_{c2s}	subcooling at condenser outlet	in K
t_{c2s}	refrigerant temp. at condenser outlet	in °C
t_c	condensing temperature	in °C

The letter “s” stands for “subcooled”.

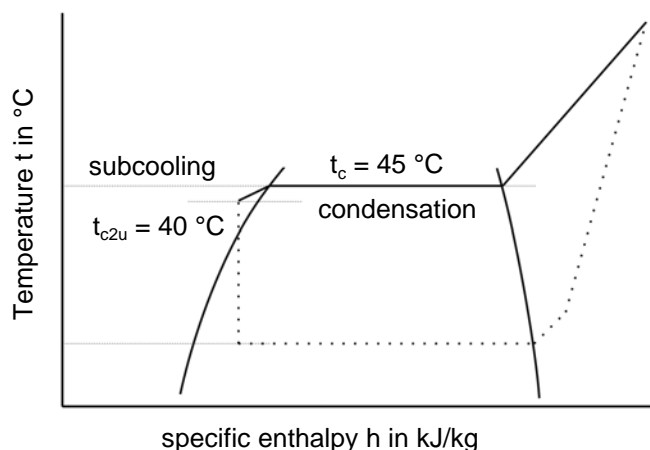


Fig. 2-7. Refrigerant subcooling

Subcooling: $\Delta t_{c2s} = t_c - t_{c2s}$

$$\Delta t_{c2s} = 45 \text{ °C} - 40 \text{ °C} = 5 \text{ K}$$

Apart from ensuring a liquid seal in front of the expansion valve, the cooling capacity may under certain circumstances be increased through subcooling.

2.4.1 How is subcooling produced?

Subcooling in the condenser

If subcooling is to be attained in the condenser, part of the condenser must be filled with liquid refrigerant. The cooling medium (air) entering this area results in subcooling the refrigerant. Since heat transfer in this area is rather difficult, the condensing pressure increases resulting in a decrease of the plant's efficiency. It is therefore not necessarily useful to carry out subcooling in the condenser. With very high condensing temperatures, however, subcooling will yield performance-increasing results despite of the effects mentioned above.

Condenser with special subcooler

Another possibility to achieve subcooling is to install a condenser featuring a subcooler or to install a subcooling loop. In this case, the liquid line downstream of the liquid receiver is passed through the fin stack of the condenser. The cooling medium flowing through this region subcools the liquid refrigerant.

Subcooling in the liquid line

If the liquid line is routed through locations with a low temperature ($t_{amb} < t_{c2u}$) additional subcooling is achieved by heat rejection to the environment.

Liquid suction vapor heat exchanger

The liquid suction vapor heat exchanger (also called internal heat exchanger) serves for the heat exchange between the liquid refrigerant in the liquid line and the refrigerant vapor in the suction vapor line. This brings about subcooling of the liquid on the one hand and superheating of the suction vapor on the other hand. This component is not used in mobile refrigeration.

2.4.2 Typical subcooling values

With plants featuring a receiver, subcooling at the receiver outlet is 0 K (provided the amount of refrigerant charge is correct). In this case, the receiver provides the required liquid seal. **Without receiver, optimum subcooling lies mostly within the range of 2 - 3 K.**

2.4.3 Subcooling measurement

Your task is to measure subcooling in a refrigeration plant operated with refrigerant R 134a.

a) What measuring instruments do you need?

- b) Where do you measure which magnitude?
c) The readout on a pressure gauge fixed directly to the condenser is $p_{ec} = 15$ bar. What is the condensing pressure p_c ?
d) What is the condensing temperature t_c ?
e) At the condenser outlet you measure the temperature $t_{c2u} = 55$ °C. What is the value for subcooling Δt_{c2u} ?
f) Evaluate the subcooling value determined.

Solution

- a) Pressure gauge, thermometer, (vapor table)
b) Condensing pressure p_o and temperature t_{c2s} measured at the condenser outlet are to be measured at the same location downstream of the condenser, if possible.
c) $p_c = p_{ec} + p_{amb} = 15 \text{ bar} + 1 \text{ bar} = 16 \text{ bar}$
d) With $p_c = 16$ bar a condensing temperature t_c of 57.9 °C can be taken from the vapor pressure table for R 134a (see appendix).
e) $\Delta t_{c2s} = t_c - t_{c2s} = 57.9 \text{ °C} - 55 \text{ °C} = 2.9 \text{ K}$
f) The subcooling value determined lies within the usual margin of 2 - 3 K. If, however, a receiver is installed in the plant, subcooling ought to be 0 K.

2 Refrigeration Cycle - Fundamentals

2.5 Refrigeration Cycle

The following illustrations show a schematic refrigeration cycle and the processes occurring in the t, h-diagramm

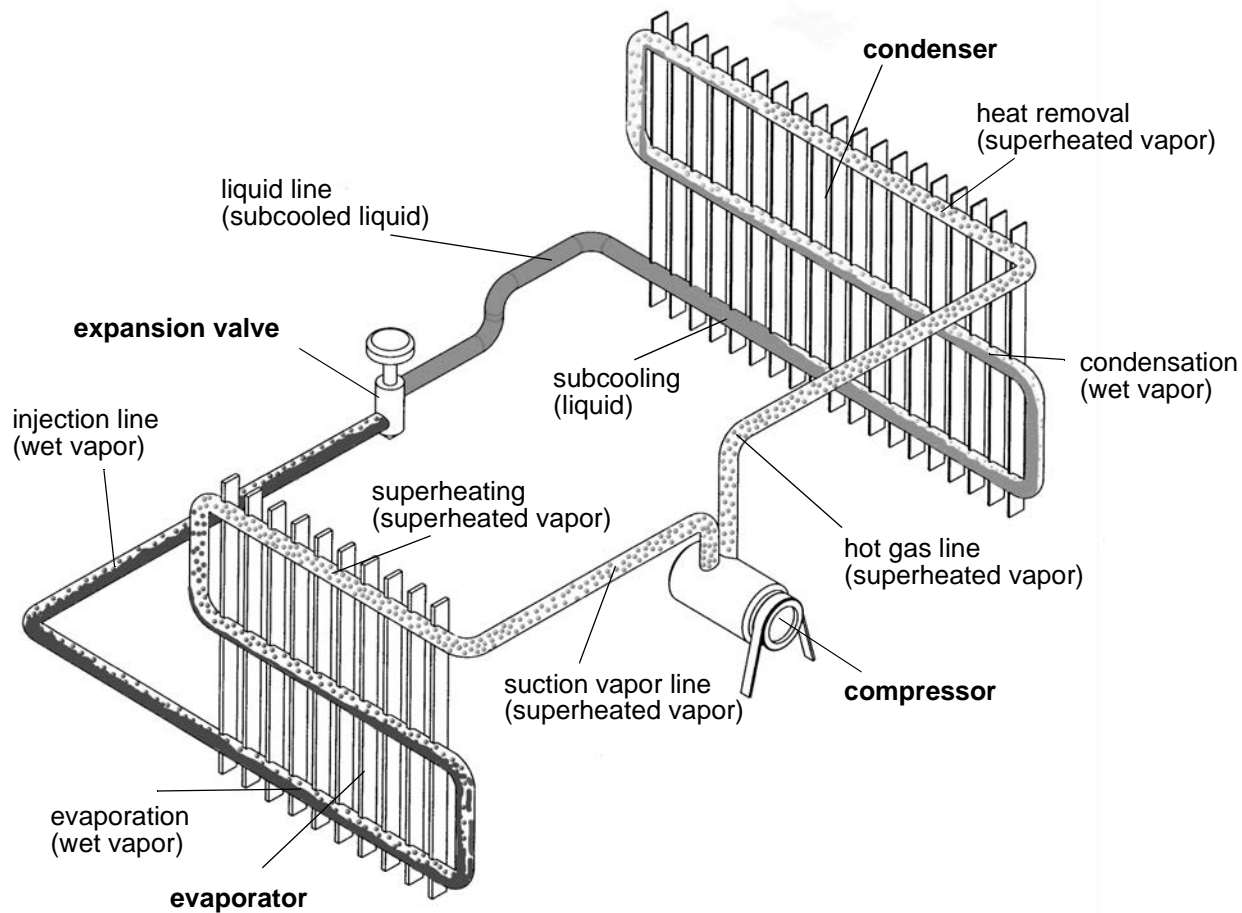


Fig. 2-8. Schematic illustration of the refrigeration cycle

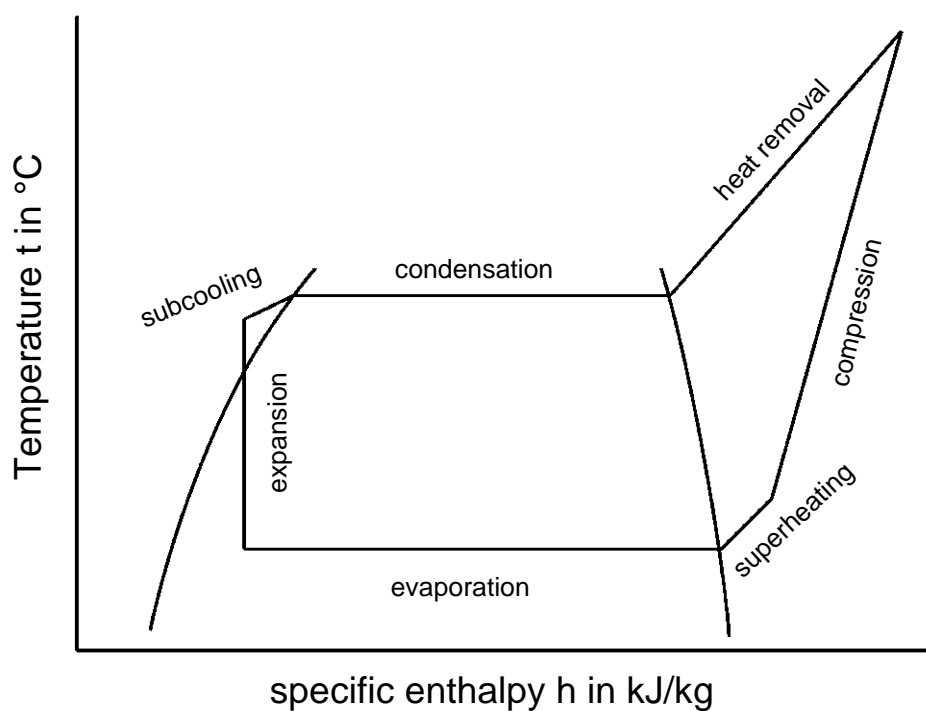


Fig. 2-9. Refrigeration cycle in t, h-diagram

3 Functions of the Main Components of the Refrigeration Cycle

3.1 Evaporator

The *evaporator's* task is to take up heat from its surroundings and to give it off to the refrigerant. The evaporating temperature must be lower than the ambient temperature. The suction effect of the refrigeration compressor and the simultaneous restriction of the expansion valve allows the desired evaporating temperature to be attained in a selective manner. The heat flux between evaporator and surroundings caused by the temperature difference causes the liquid refrigerant supplied by the expansion valve to be evaporated in the evaporator (*evaporating zone*) and possibly superheated (*superheating zone*).

3.1.1 Processes taking place in the evaporator

The liquid (possibly subcooled) refrigerant entering the expansion valve is expanded to evaporating pressure p_0 . Part of the liquid refrigerant evaporates before it reaches the evaporator. This vapor proportion x is the higher the higher the temperature difference between condensing and evaporating temperature is. In Fig. 3-1 this proportion amounts to 20 %.

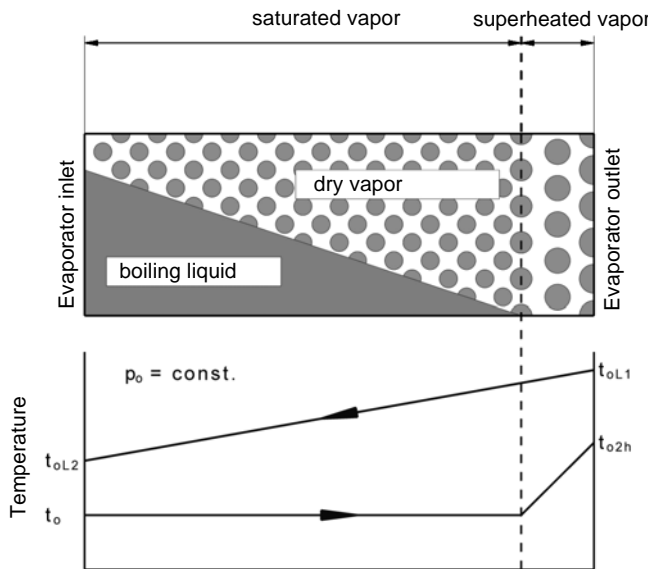


Fig. 3-1. Processes in the evaporator

By the time the refrigerant reaches the evaporator outlet it has completely evaporated and passed into the superheated state. The refrigerant temperature will not rise until the liquid has evaporated completely.

This operating principle of the evaporator is called dry evaporation (dry expansion).

In the superheating zone poor heat transfer properties exist. Also, the temperature difference in relation to the surroundings is lower due to the increased refrigerant temperature. As a consequence, only little heat can be transferred in the area of the superheating zone.

3.1.2 Evaporator capacity

The evaporator capacity primarily depends on the following conditions:

Surface A

The larger the heat transfer surface the more thermal energy can be transferred. Here, the entire outside surface involved in the heat transfer process is to be taken into consideration.

Driving temperature gradient Δt_1

The driving temperature gradient is defined with respect to the inlet temperature difference = air inlet temperature - evaporating temperature

$$\Delta t_1 = t_{oL1} - t_o \quad \text{in K}$$

If the heat exchanger is viewed by itself, the following applies: The higher the driving temperature gradient the higher the heat exchanger capacity.

k-value

The k-value (thermal conductivity coefficient) stands for the quality of heat transfer. It measures the heat transfer from the medium to be cooled (air) to the refrigerant. This value considers the following influences:

- Evaporator material (copper, aluminium)
- Pipe spacing and pipe diameter
- Piping arrangement
- Evaporator depth
- Fin spacing
- Fin shape
- Surface finish
- Contamination and ice formation
- Flow against or through fins/plates
- Flow rates of air and refrigerant
- Flow pattern of air and refrigerant
- Temperature-dependent properties of cooling medium (air) and refrigerant
- Distribution of both media
- Refrigerant state (vaporous, saturated vapor, liquid)
- Oil proportion in refrigerant

Ideally, the driving temperature gradient should not exceed a value of approx. 10 K on the evaporator side. Greater temperature differences (i.e. lower evaporating temperature) as they are common in mobile refrigeration applications (15 - 20 K under normal operating condi-

3 Functions of the Main Components of the Refrigeration Cycle

tions) due to space, weight and cost considerations, have a negative effect on the efficiency of the plant. If the refrigeration plant is to be used for the dehumidification of air, greater temperature differences are necessary. From approx. -5° evaporating temperature, frost will start to form on the evaporator. An anti-freeze control must be provided to protect the evaporator from icing up in that it shuts off the compressor or activates a power control device.

For the evaporator capacity (refrigeration capacity) Q_o the following applies:

$$\dot{Q}_o = A \cdot k \cdot \Delta t_1 \quad \text{in kW (kJ/s)}$$

This equation is used in the development of heat exchangers. The k-value takes into account both the refrigerant and cooling medium.

The refrigeration capacity can be determined by means of the air or the refrigerant.

The **refrigeration capacity determined by means of the air** can be determined using the following equation:

Note: This equation only applies to dry air. If water vapor condensation occurs, the performance may well be higher by 40 - 50 % !

$$\dot{Q}_o = \dot{m}_L \cdot c_L \cdot \Delta t_L \quad \text{in kW (kJ/s)}$$

m_L mass flow of the air in kg/s
 c_L specific heat capacity of air in kJ/(kg · K)
 Δt_L cooling of the air in K
 $\Delta t_L = t_{oL1} - t_{oL2}$

The **refrigeration capacity determined on the basis of the refrigerant** can be determined using the following equation:

$$\dot{Q}_o = \dot{m}_R \cdot \Delta h_o \quad \text{in kW (kJ/s)}$$

m_R mass flow of the refrigerant in kg/s
 Δh_o enthalpy difference in the evaporator in kJ/kg

In practice, this approach for determining the refrigeration capacity is hardly practical. The refrigerant mass flow would have to be measured. This approach is used by official testing facilities.

3.1.3 Evaporating pressure

The evaporating pressure p_o is the result of the "restriction" of the expansion valve and the suction effect of the

compressor. But also the surface, its contamination or icing, and the volume flow and/or the inlet temperature of the cooling medium (air, water or brine) are factors influencing the current evaporating pressure.

The following relationships apply:

Table 3-1. Influence of various parameters on the evaporating pressure

Influences	Effect
Increase of air or brine temperature	p_o rises
Decrease of air or brine temperature	p_o falls
Increase of surface	p_o rises
Reduction of surface	p_o falls
Contamination, icing up (reduction of k-value)	p_o falls
Increase of air, water, brine flow rate (increase of k-value)	p_o rises

A lower evaporating pressure (evaporating temperature) reduces the refrigeration capacity. If the evaporating temperature is lowered by one Kelvin the refrigeration capacity will be reduced by approx. 4 %.

3.2 Compressor

The **compressor's** function is to compress vaporous refrigerant from the low evaporating pressure level to the higher condensing pressure level. In addition it must ensure the flow rate (refrigerant mass flow) required for the specified refrigeration capacity.

3.2.1 Delivery behaviour of the compressor

The **delivery behaviour** of the compressor is governed by the characteristics described below.

Geometric displacement

The geometric displacement of a compressor is defined by the dimensions of the piston capacity (stroke length, number of cylinders, piston diameter). The larger the geometric displacement the larger the attainable refrigerant mass flow rate.

Geometric volume flow

If the volume flow rate (the time-based delivery behaviour) is to be determined, the compressor speed must be taken into account. The flow rate is usually expressed in m^3/h . The higher the compressor speed the higher the delivery rate. With increased speeds, however, the losses will increase as well.

3 Functions of the Main Components of the Refrigeration Cycle

Intake volume flow

For manufacturing tolerance reasons and compressor ruggedness (e.g. resistance to liquid particles in the intake vapor) a residual volume (**dead space**) is located above the top dead center (TDC). After the compression, a high-pressure gas is present in this dead space.

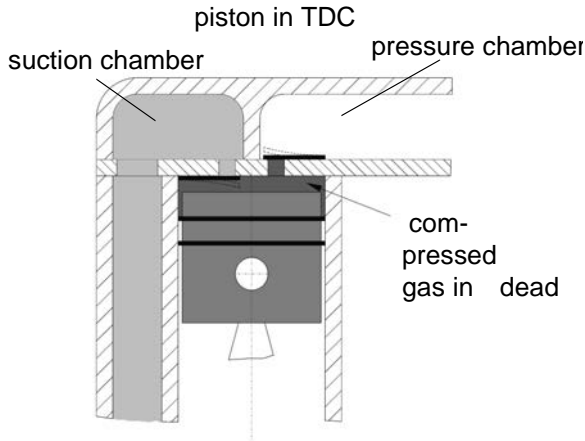


Fig. 3-2. Dead space of piston compressor

While the piston is moving down again to perform another suction stroke, this gas must be expanded to intake pressure before the inlet valve can open. This causes the effective suction volume to be reduced, and the piston chamber is therefore not utilized to full capacity anymore. The volume flow actually delivered by the compressor is lower than the geometric piston-swept volume flow.

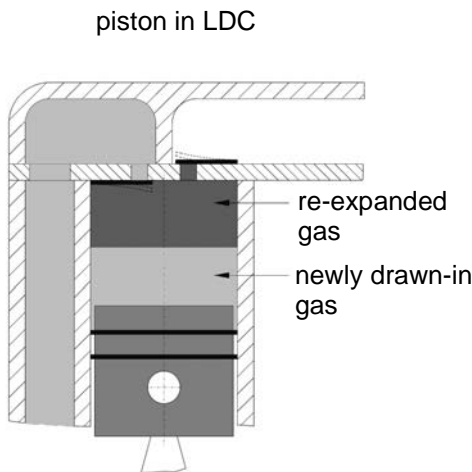


Fig. 3-3. Effect of the dead space

Volumetric efficiency

The ratio of intake volume flow to geometric piston-swept volume flow is referred to as **volumetric efficiency** of the compressor.

$$\lambda = \frac{\dot{V}_{V1}}{\dot{V}_g} \quad \text{in m}^3/\text{m}^3$$

λ	volumetric efficiency	in m ³ /m ³
\dot{V}_{V1}	actual intake volume flow rate	in m ³ /s
\dot{V}_g	geometric volume flow	in m ³ /s

Apart from various other influence factors, e.g. the temperature of the refrigerant, oil proportion, etc., the pressures prevailing upstream and downstream of the compressor have a great influence on the volumetric efficiency. For this reason, it is useful to plot the volumetric efficiency against the **pressure ratio**. It should be noted that absolute pressures have to be used for the calculation of the pressure ratio.

$$\pi = \frac{p_{V2}}{p_{V1}}$$

π	pressure ratio	-
p_{V2}	final compression pressure in bar	
p_{V1}	intake pressure	in bar

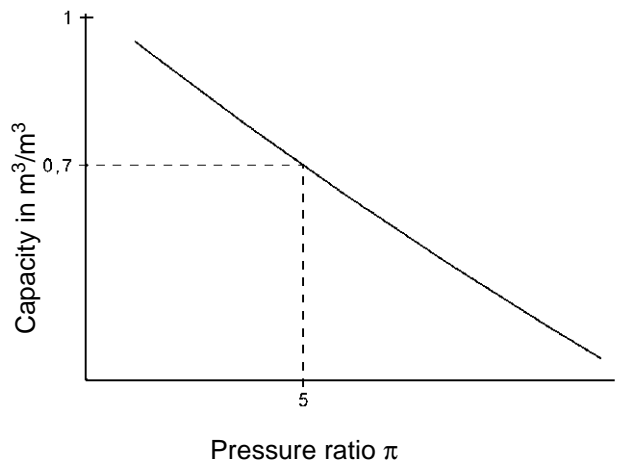


Fig. 3-4. Volumetric efficiency as a function of the pressure ratio

As the pressure ratio increases the volumetric efficiency decreases steadily. Volumetric efficiency is primarily dependent on the dead space and the re-expansion taking place there. Very small compressors with a relatively large dead space and several pistons have therefore a poor volumetric efficiency behaviour.

Intake density

The flow rate of a compressor is also dependent on the refrigerant density at the intake fitting. If the compressor draws in gas with a low density the swept volume contains only a small amount of refrigerant. As a consequence, only a small amount of refrigerant is delivered per stroke.

As the superheat of the suction vapor increases, the refrigerant density and thus the flow rate of the compressor

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sor decreases.

With decreasing intake pressure the refrigerant density decreases as well and hence the flow rate.

3.2.2 Refrigeration capacity

Since the compressor is simply a refrigerant delivery unit, the specification of a refrigeration capacity is dependent on the state of the refrigerant upstream and downstream of the heat exchanger (evaporator) flow rate.

Apart from the compressor and the refrigerant itself, the following factors are of importance to achieve a **high refrigeration capacity**:

- high evaporating pressure (suction pressure) and thus a high evaporating temperature
- low condensing pressure and thus a low condensing temperature
- low refrigerant temperature at the inlet of the expansion valve
- little superheat (suction line)

Fig. 3-5 represents the influence of evaporating and condensing temperature on the refrigeration capacity.

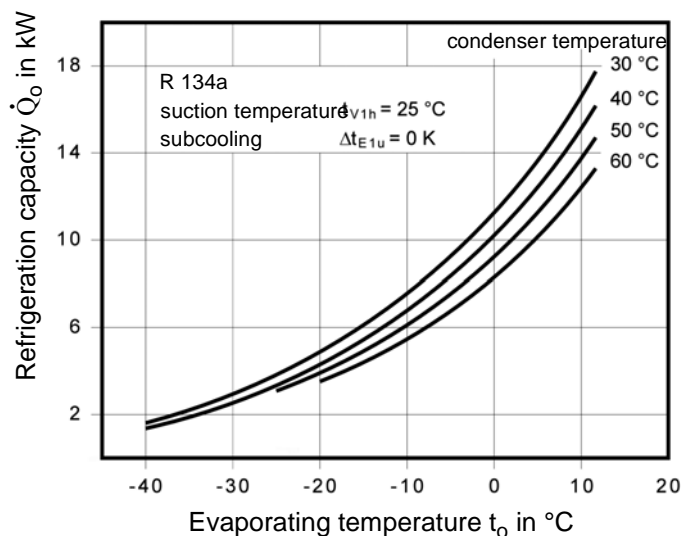


Fig. 3-5. Refrigeration capacity plotted against the evaporating temperature at various condensing temperatures

A change in evaporating temperature (evaporating pressure) has a greater effect on the refrigeration capacity of the compressor than a change in condensing temperature. It is therefore of primary importance to ensure proper sizing of the evaporator of a refrigeration plant and to keep pressure losses in the suction vapor line to a minimum. The refrigeration capacity in the lower speed range can be significantly increased by a higher speed. In the upper range limits are imposed by increasing losses.

3.2.3 Compressor power input

The **power requirement** of the compressor is dependent on the following influencing variables:

- compressor type and design
- compressor speed
- refrigerant
- density of the refrigerant at the compressor inlet
- pressure of the refrigerant at the compressor inlet
- pressure of the refrigerant at the compressor outlet

The compressor power consumption is stated in the technical data of the compressor manufacturers. Very little superheating and liquid refrigerant entering the compressor result in considerably higher power consumption. With open-type compressors the transmission losses (belt efficiency), mechanical losses and external motor losses must be taken into account.

Influence of compressor speed

Since more refrigerant is delivered at higher speeds also a higher compressor driving power is required.

Influence of the refrigerant

The energy required for delivering the refrigerant depends on the suction density and the pressure ratio. There is a significant difference between some of the various refrigerants. If the compressor delivers a given volume flow, the resulting refrigeration capacity can vary greatly. This is due to differing evaporating heat values. The same compressor, for example, can yield a refrigeration capacity of 9 kW with the refrigerant R 134a and a capacity of 15 kW with the refrigerant R 502. The power requirement of a compressor with refrigerant R 22 is approx. 50 % higher than with refrigerant R 134a.

Influence of the condensing temperature (pressure)

A higher condensing temperature (pressure) primarily results in a higher compressor power input. At a constant evaporating temperature the pressure ratio increases. This results in a lower refrigerant flow rate (refrigeration capacity).

Influence of the evaporating temperature (pressure)

The refrigerant density decreases with decreasing evaporating temperature. At a constant condensing temperature the pressure ratio increases at the same time. Consequently, the refrigerant mass flow rate and thus the compressor driving power decreases.

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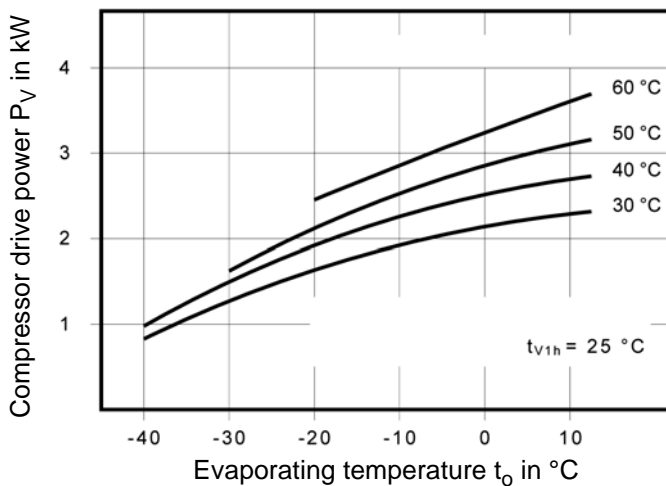


Fig. 3-6. Compressor power requirement as a function of the evaporating and condensing temperature

3.2.4 Compressor operating limits

Apart from the refrigeration capacity and the required drive power, the compressor operating limits are of utmost importance to the practitioner.

Interpretation of the compressor operating limits diagram (Fig. 3-7)

① The compressor can be used up to a evaporating temperature $t_0 = 25$ °C. Above this temperature the motor would experience an overload condition due to the high gas density (high drive power). Also, refrigeration is not practical in this temperature range. To protect the com-

pressor and drive motor, a MOP valve is frequently used. The latter limits the evaporating pressure in upward direction.

② The compressor can be used up to a condensing temperature $t_c = 70$ °C. Firstly, this limit is imposed by the maximum permissible operating pressure on the high pressure side (e.g. $p_{zul} = 25$ bar) and secondly by the critical hot gas temperature t_{V2h} .

Depending on the individual manufacturer and the compressor type, the hot gas temperature - measured at the pressure pipe of the compressor - is limited to 120 °C to 140 °C. In the compression chamber a gas temperature must be expected that is still higher by 20 to 30 K. Coking is likely to occur. In addition, high temperatures at the pressure joints may promote chemical reactions in the mixture of refrigerant-lubricant-water-dirt. High hot gas temperatures have a negative effect on the life of the compressor.

③ At a suction vapour temperature t_{V1h} of over 20 °C above line ③ the compressor may only be used in conjunction with an additional cooling unit as otherwise a danger of coking or excessive thermal load of the compressor exists. It is recommended that a heat protection thermostat be used. Also, a highly thermostable oil must be used.

④ The compressor is approved for operation up to an evaporating temperature $t_0 = -30$ °C. Below this temperature the hot gas temperature would be too high. In addition, the delivered refrigerant volume flow is dramatically reduced.

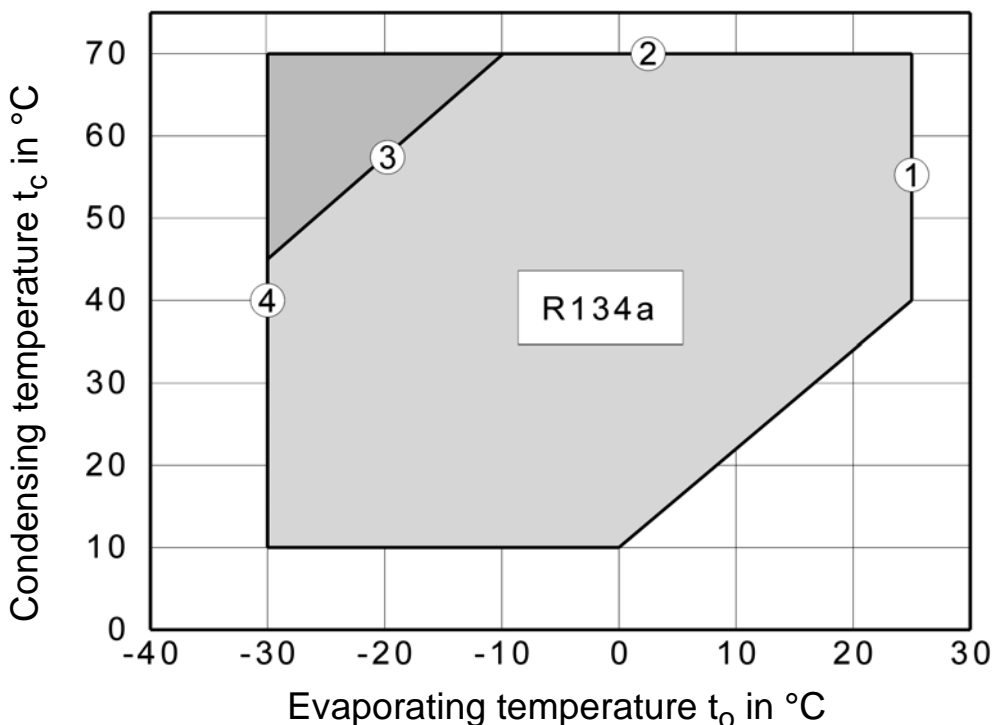


Fig. 3-7. Operating limits of an open reciprocating compressor

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Additional operating limits are the maximum permissible operating pressures on the high and low pressure sides as well as the maximum and minimum compressor speeds. At higher speeds, the compressor overheats; if the speed is too low, the bearings are not sufficiently lubricated due to an inadequate flow rate of the oil pump.

Failure to comply with these operating limits may result in damage to the compressor!

3.3 Condenser

The **condenser** serves to reject heat from the refrigeration cycle. This heat is given off to the surrounding air. Since a heat flow can only flow when there is a temperature difference, the condensing temperature must always lie above the inlet temperature of the ambient air. The heat to be rejected from the condenser is the heat absorbed by the evaporator and the heat of the compressor motor as well as any other heat absorbed (e.g. suction vapour line).

3.3.1 Processes taking place in the condenser

The condenser can be subdivided into three zones:

- heat rejection
- condensation
- subcooling

Pressure drops are not taken into account in the following considerations. The same pressure is therefore assumed to prevail in all three zones.

Heat rejection zone

The refrigerant enters the condenser in a vaporous (superheated) state. As a first step, heat is removed from the refrigerant, the temperature of the refrigerant drops to condensing temperature. A pure gas flow occurs in the heat rejection zone. Despite a large driving temperature gradient and high flow velocities, heat transfer is not as good as with two-phase flows due to the low refrigerant density. The surface area of the heat rejection zone is approx. 10 - 15 %.

Condensing zone

As soon as the refrigerant has cooled to condensing temperature, the condensation process starts. The condensing temperature results from the prevailing pressure. During condensation, this temperature remains constant. Heat removal results in a phase change, not in a decrease in temperature. Heat transfer is best in this region.

Subcooling zone

As soon as there is no more vapor contained in the refrigerant, subcooling starts through further heat removal. As the phase change from the gaseous to the liquid state is

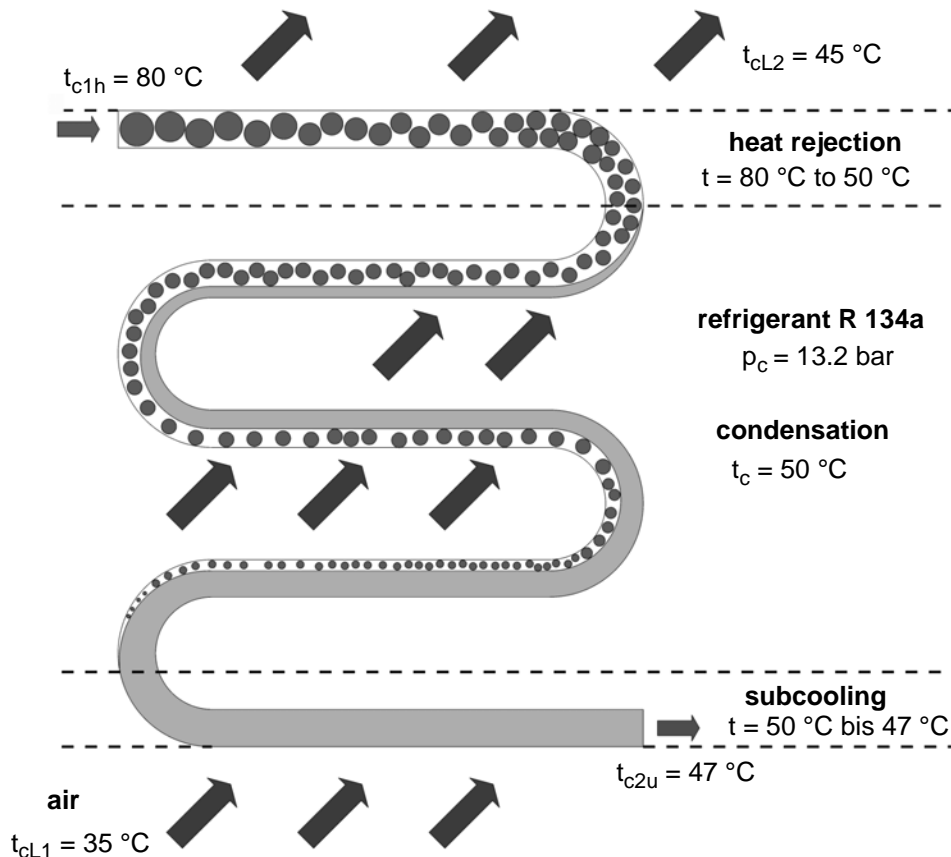


Fig. 3-8. Condenser zones

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now complete, any additional heat removal results in a temperature drop. Due to the low flow rate of the refrigerant heat transfer in this region is very poor. Subcooling may under certain circumstances be necessary to ensure that no vapor bubbles can form at the expansion valve inlet. The refrigerant must be completely liquid at the inlet of the expansion valve.

Since heat transfer is very poor in the subcooling zone, it is useful to dispense with a subcooling zone in the condenser. This can be achieved by installing a liquid receiver. The latter separates the liquid refrigerant from the remaining vapor bubbles thus ensuring a liquid seal in front of the expansion valve. The entire condenser surface can thus be used for heat removal and condensing. If no receiver is installed, subcooling would have to occur due to backed-up refrigerant in the condenser. "Surface loss" along with a higher condensing pressure would be the result.

In the event that the liquid line has a lot of built-in parts causing pressure drops, is routed through a warmer environment or must overcome great differences in height, subcooling measures in the form of a subcooler coil have to be taken. Also in this case, a receiver must be installed to provide a separation between the gaseous and liquid phase. This separation must be achieved before subcooling takes place as otherwise the condensing zone moves out of the condenser and no subcooling is possible.

3.3.2 Condenser capacity

The **condenser performance** is - just as the evaporator performance - dependent on the factors: area, k-value and driving temperature gradient.

The driving temperature gradient is defined as the inlet temperature difference = condensing temperature - air inlet temperature.

$$\Delta t_1 = t_{cL1} - t_c \quad \text{in K}$$

The higher the driving temperature gradient the higher the heat exchanger performance.

The driving temperature gradient should not exceed a value of approx. 15 K on the condenser side. Higher temperature differences (i.e. a higher condensing temperature), as they are common in mobile refrigeration applications (10 - 30 K under normal operating conditions) for reasons of space, weight and costs, have a negative effect on efficiency of the plant.

3.3.3 Condensing pressure

In the case of air-cooled condensers the temperature of the cooling medium will vary greatly. High outside temperatures result in an increase, low temperatures in a decrease of the condensing pressure. For a refrigeration plant to be able to operate economically, the condensing

pressure must be maintained within specified limits. High condensing pressures result in a lower refrigeration capacity and an increased power consumption of the compressor; insufficiently high condensing pressures adversely affect the function of the expansion valve. The refrigerant supply to the evaporator is impaired.

For the condensing pressure the following interrelationships apply

Table 3-2. Effect of various parameters on the condensing pressure

Influences	Effect
Increase in ambient temperature	p_c rises
Decrease in ambient temperature	p_c falls
Increase in surface	p_c falls
Reduction in surface	p_c rises
Contamination (reduced k-value)	p_c rises
Increase in air flow rate (increased k-value) e.g. additional fan	p_c falls
Fan failure	p_c rises

If the condensing temperature is increased by 1 Kelvin the refrigeration capacity will be reduced by approx. 1.5 %.

3.4 Throttling device

The **throttling device** in the refrigeration cycle serves to expand liquid refrigerant from a higher pressure and a higher temperature to a lower pressure and a lower temperature. Expansion starts immediately downstream of the smallest cross section at the valve seat of the throttling device. Part of the refrigerant evaporates extracting heat from the still liquid part. As a result, the liquid refrigerant is cooled down to evaporating temperature.

Another task of the throttling valves is to feed the evaporator only so much liquid refrigerant as can evaporate under the respective operating condition. If the evaporator is fed too much refrigerant, the liquid refrigerant that has not evaporated will reach the compressor. If the evaporator is supplied with an insufficient amount of liquid refrigerant, the evaporator surface is not utilized to full capacity. As a result, excessive working heat may occur already in the evaporator so that the final compression temperature is unacceptably high. A high efficiency of the refrigeration plant can be achieved when the refrigerant evaporates completely and leaves the compressor under slight superheat conditions.

The expansion process of the refrigerant in the throttle valve is isenthalpic, which means that the heat content neither increases nor decreases when the refrigerant

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flows through the throttle valve.

The refrigerant at the inlet of the throttling valve is in a liquid state at a high pressure. After it leaves the throttling valve the refrigerant is at a low pressure in the saturated vapour region. Partial evaporation has taken place. 20 to 50 % of the liquid refrigerant have evaporated before the refrigerant enters the evaporator.

refrigerant migrating into the evaporator and/or compressor. Liquid slugging is a frequent cause of defective compressors.

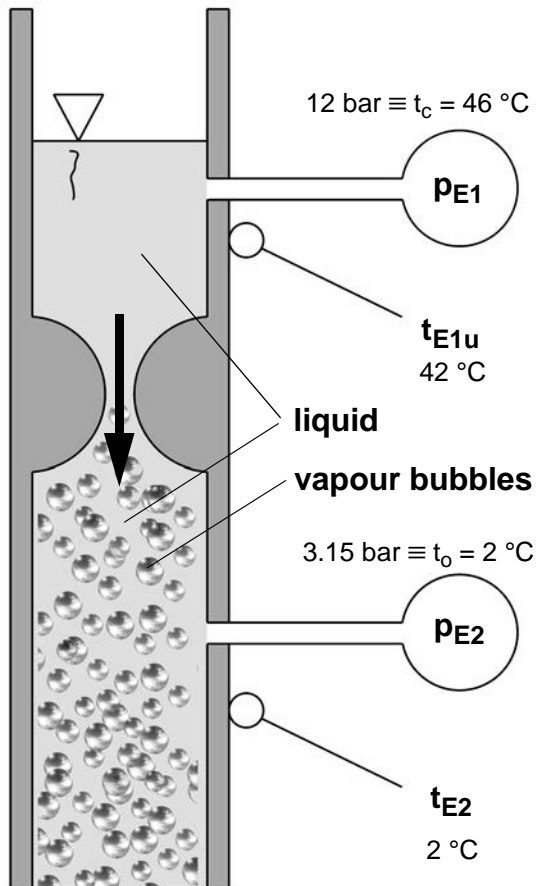


Fig. 3-9. Expansion process of liquid refrigerant

An appropriate throttling valve has to be selected depending on the operating mode and type of the refrigeration plant. Plants that always operate under constant conditions can be equipped with a fixed throttling element (expansion tube or capillary tube). If major load changes occur during operation it is recommended that a regulating throttling element such as a temperature-controlled expansion valve be used.

A specific setting of the expansion valve would be required for every operating point of a refrigeration plant. As it is not possible to continuously readjust the valves (and they should not be constantly readjusted), the basic adjustment of the regulated valves during commissioning is a task that has to be carried out very responsibly and requires a certain know-how and experience.

When the plant is not in use, incorrectly adjusted expansion valves (too little superheating) frequently result in the

4 Components of the Refrigeration Cycle

4.1 Compressor

4.1.1 General

Few compressors of stationary refrigeration plants are subjected to more extreme external conditions as in mobile refrigeration applications. Apart from ambient temperatures of approx. $-40\text{ }^{\circ}\text{C}$ to $+120\text{ }^{\circ}\text{C}$ prevailing in the vehicle and speeds of approx. 700 to $9\,000\text{ min}^{-1}$ in the case of passenger cars as well as 500 to $3\,500\text{ min}^{-1}$ in the case of trucks and buses, non-load-regulated compressors are greatly strained by frequent on/off cycles and "start-ups" in fractions of a second.

In mobile applications nearly all known compressor designs have been tried and tested and some of them put to practical use. Apart from the - still most widely used - piston compressors (swash-plate compressors or plunger compressors) also vane-cell, scroll (spiral compressors) and screw-type compressors are used.

For about 15 years compressors with capacity control have been developed which constitute technologically advanced products. Given the fact that before 1980 users had only a choice between the pulsation-efficient 2-cylinder YORK compressor, the 4-cylinder DELCO radial flow compressor and the 16 kg heavy, 6-cylinder FRIGIDAIR swash-plate compressor, then it becomes apparent that very significant technological progress has been achieved.

The most essential demands on refrigeration compressors for mobile applications are as follows:

- high cooling capacity at low speeds (idle requirement)
- small and light-weight, yet capable of delivering large volumes
- uniform torque characteristic, low starting torques, low pulse rate and quiet running
- burst-proof up to approx. $9\,000\text{ min}^{-1}$ (passenger car) or up to approx. $3\,500\text{ min}^{-1}$ (bus)
- resistant to higher ambient temperatures (up to $120\text{ }^{\circ}\text{C}$)
- insensitive to drawn-in liquid refrigerant ("liquid slugging")
- infinitely variable output, remote ("external") control best choice
- cost-efficient and long life
- low power consumption and therefore high coefficient of performance
- insensitive to re-condensed refrigerant in the cylinder head

As these requirements are partly contradictory no compressor model can meet all of the specifications.

A piston compressor under idle conditions (idling speed $700 - 800\text{ min}^{-1}$) yields a higher refrigeration capacity than a scroll compressor.

A scroll compressor, on the other hand, can be operated without any problems at high speeds, has a good volumetric efficiency, a relatively uniform torque characteristic and is more resistant to non-vaporated liquid refrigerant.

A high cooling capacity in the idling mode requires a high refrigerant flow rate as it can only be attained by compressors with a large piston capacity or by means of higher speeds (transmission ratio $> 1 : 1$). The large-capacity compressor is either overdimensioned at higher speeds or else the compressor will rev up extremely at larger transmission ratios, which will impair the fatigue strength.

In general it can be said that the choice of a compressor is always a compromise between several objectives.

4.1.2 Reciprocating Compressor

A **disadvantage** with this type of construction are the oscillating masses of the driving mechanism (piston- and connecting rods). And there is also the dead space in the area of the top dead center commonly found in all models. Firstly, this affects the volumetric efficiency and secondly restricts the available space when liquid refrigerant is drawn in. Another disadvantage present the valves on the intake and discharge sides in that they offer flow resistance and a spring tension that needs to be overcome (adverse effect on volumetric efficiency).

The **advantage** of the reciprocating compressor is its relatively high volumetric efficiency at low speeds (at high speeds the valves have a negative effect). At the same time the highest possible cooling capacity under idling conditions can be achieved. This aspect is the strongest point in favour of the piston compressor and it is for this reason that it is often given preference over other compressor types.

In the area of air-conditioning systems for passenger cars and commercial vehicles **axial piston compressors** (swash-plate compressors), for buses or other installations requiring a larger cooling capacity **plunger piston compressors** are used.

In plunger piston compressors the piston is directly connected to the crankshaft via the connecting rods. In axial piston compressors the connecting rods are attached to a swash plate.

The figure below shows a cutaway view of an open-type piston compressor (plunger piston compressor). The compressor in the above figure features a drive shaft brought to the outside. The specific type of drive can be selected depending on the application making this design suitable for universal use.

4 Components of the Refrigeration Cycle

4.1.2.1 Plunger piston compressor

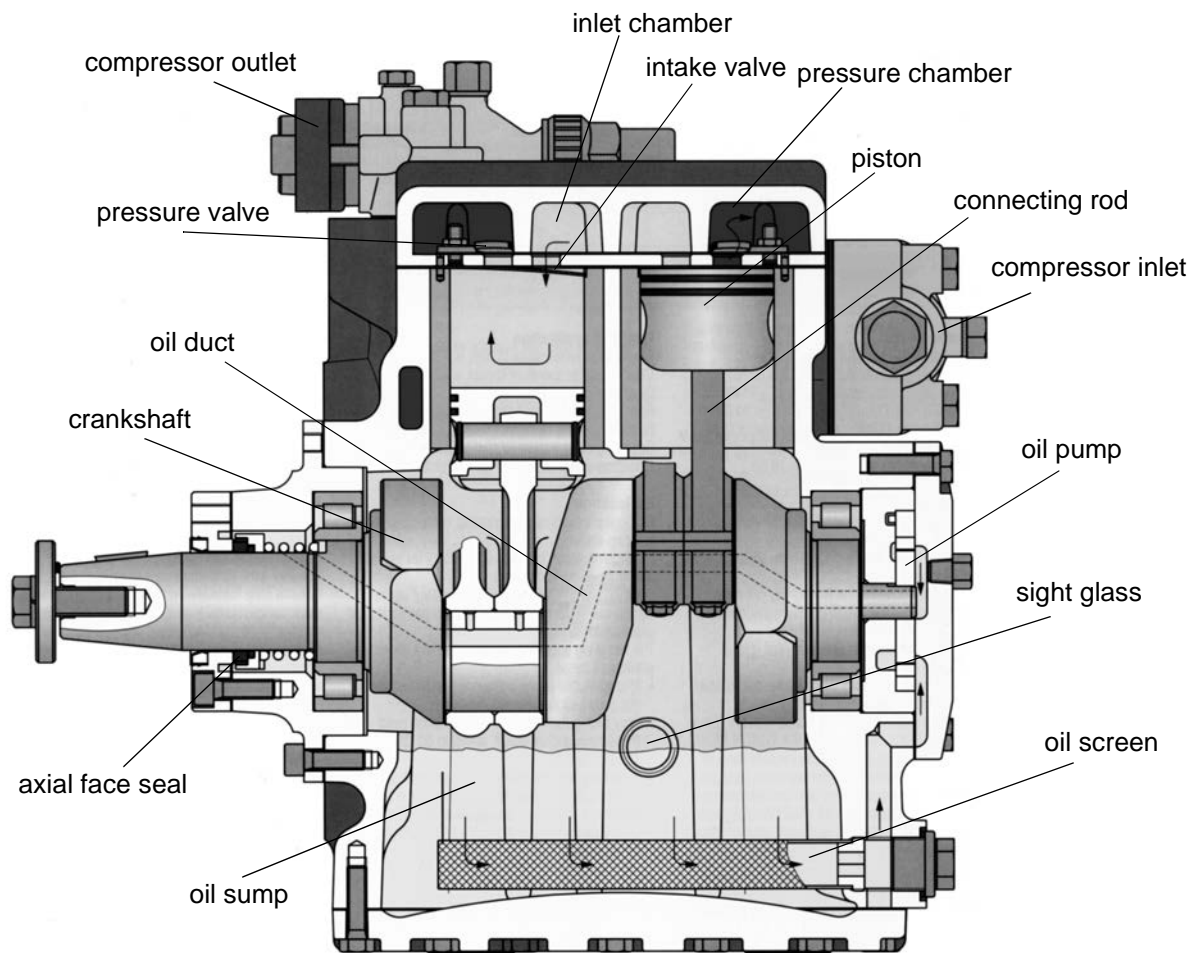


Fig. 4-1. Open-type plunger piston compressor, schematic (Bock)



Fig. 4-2. Plunger piston compressor FK 40 (Bock)

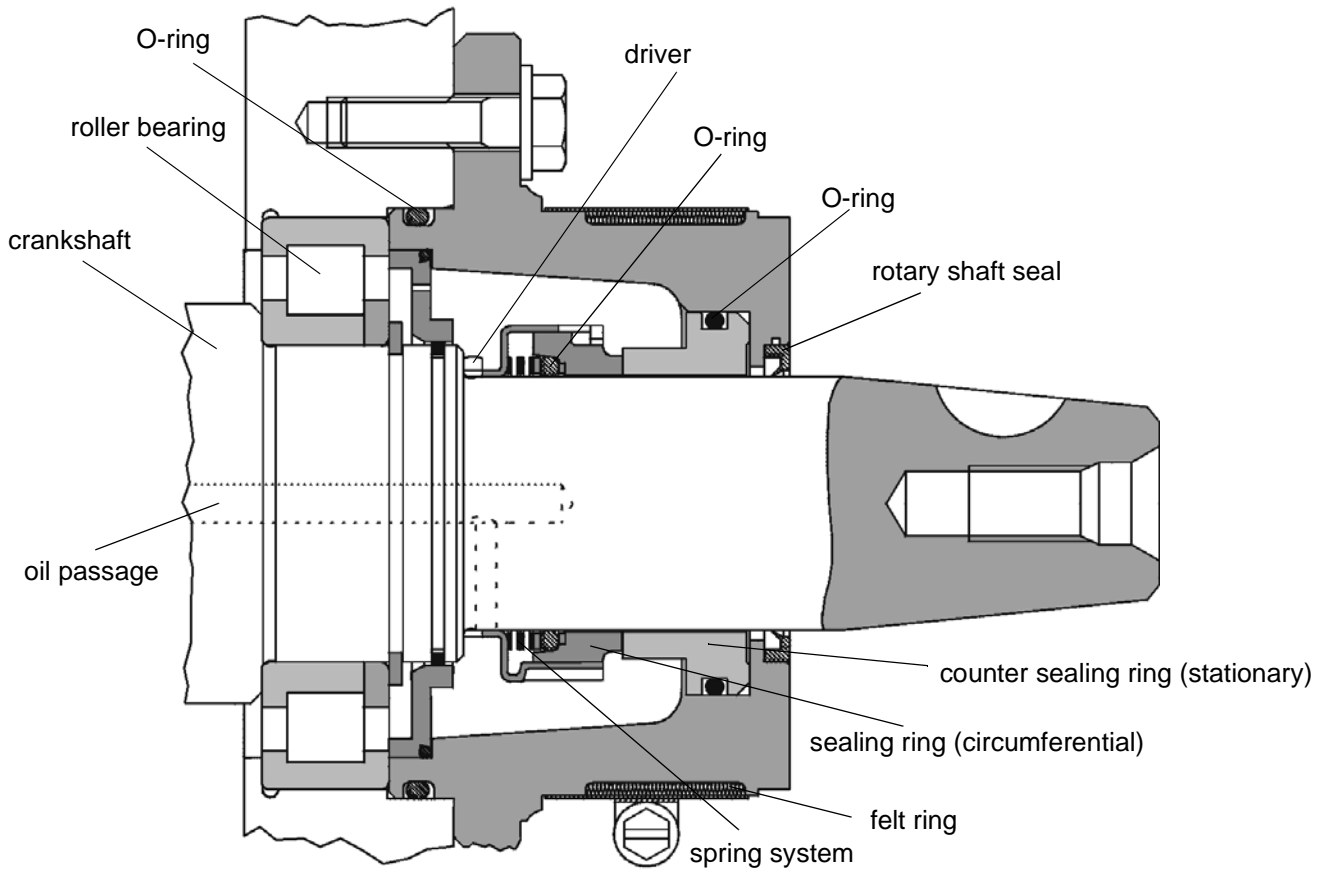


Fig. 4-3. Axial face seal of shaft leadthrough of an open Bitzer compressor

A disadvantage is the axial face seal of the drive shaft since as a rule small leaks of both oil and refrigerant will occur. The above figure depicts the construction of the shaft seal of an open-type compressor.

Axial face seals for refrigeration compressors consist of a spring system, a circumferential sealing ring and a stationary counter sealing ring. In lieu of a spring system also metal bellows are used.

The spring system and the sealing ring are arranged on the circumference of the shaft. The counter sealing ring is located in the housing cover, an O-ring provides a seal against the outside. For sealing action within the spring system as well as between the spring system and the shaft is accomplished by both profiled rings and O-rings.

For the sealing action of the sealing rings a sufficient oil supply. The sealing ring is pushed against the counter sealing ring by the action of the spring system. The pressure inside the compressor casing causes oil to be forced into the gap between the sealing rings (hydrostatic bearing). In addition, the tangential sliding motion of the undulated sealing surfaces results in additional oil to be transferred to the bearing (hydrodynamic bearing). The sealing rings are separated from one another and sealed by a thin oil film. Since a slight oil flow is required for the sealing action, no perfect seal can be achieved. The emerging oil is retained by the radial shaft sealing ring and flows

through a hole underneath the felt ring. The latter absorbs the emerged oil thus preventing any contamination of the compressor.

4.1.2.2 Axial piston compressor

Due to its low number of cylinders, the plunger piston compressor creates strong pulsations in the refrigerant lines. In addition, the speed is limited by the high oscillating masses. Due to the increasing demand in the passenger car sector for quiet operation of the refrigeration system attempts have been made to increase the number of cylinders of the compressor. Owing to the design of axial piston compressors (swash-plate compressors) it is possible to produce very compact, burst-proof compressors with a high number of cylinders (up to 10).

In most cases, the axial piston compressor is driven by the vehicle engine by means of a V-belt. In non-regulated compressors the belt pulley is connected to the compressor shaft via an electromagnetic clutch. In the case of more advanced compressors with capacity control this clutch is sometimes dispensed with.

4 Components of the Refrigeration Cycle

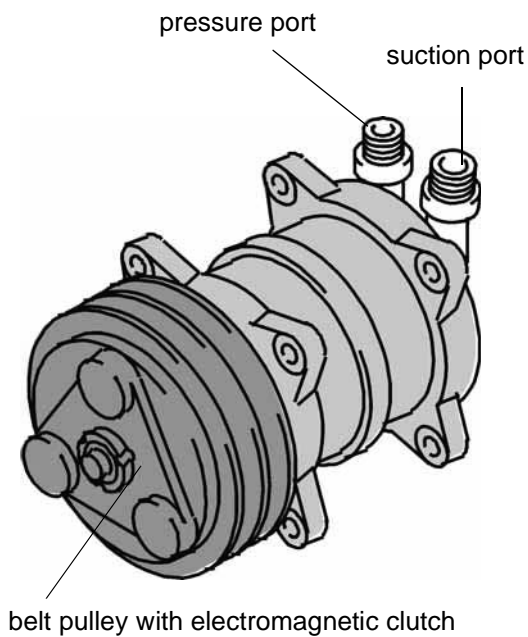


Fig. 4-4. Axial piston compressor

Fig. 4-5 shows a single-acting (the pistons only pump in one direction) axial piston compressor with stationary swash plate.

The compressor comprises several pistons whose axial motion is created by means of the swash plate connected to the compressor shaft. On the cylinder head two automatically operated valves per cylinder are provided. These valves are designed so that they regulate the refrigerant flow automatically, both in the suction and discharge cycle.

Lubrication is effected through the pressure difference between the pressure prevailing in the compressor casing and the pressure resulting from the suction process, the oil slinging caused by rotating component parts and the oil entrained in the refrigerant flow.

4.1.3 Vane compressor

The standardized designation is "rotary vane compressor". Rotary vane compressors are more suited for smaller cooling capacities, swash-plate compressors for higher cooling capacities. Rotary vane compressors are available with a displacement of 50 to 150 cm³, swash-plate compressors with displacements between 150 to 200 cm³. Rotary vane compressors may have 3, 4 or 5 vanes. The 5-vane version has the same running behaviour as a 10-cylinder reciprocating compressor, a 3-vane version corresponds to a 6-cylinder reciprocating compressor. These compressors are burst-proof up to approx. 8 000 min⁻¹, briefly also up to 9 000 min⁻¹ and offer quiet running properties.

Mode of operation

A rotor arranged eccentrically in a cylinder rotates around an axis that is eccentric in relation to the cylinder axis. For creating subdivisions of the crescent-shaped swept space the rotor is provided with movable slides (vanes) that are held outwardly against the cylinder wall by centrifugal force. In order to prevent the compressor from running backward upon switching off, a check valve is frequently installed.

Fig. 4-6 shows a double-inlet rotary vane compressor with 5 vanes and oval enclosure.

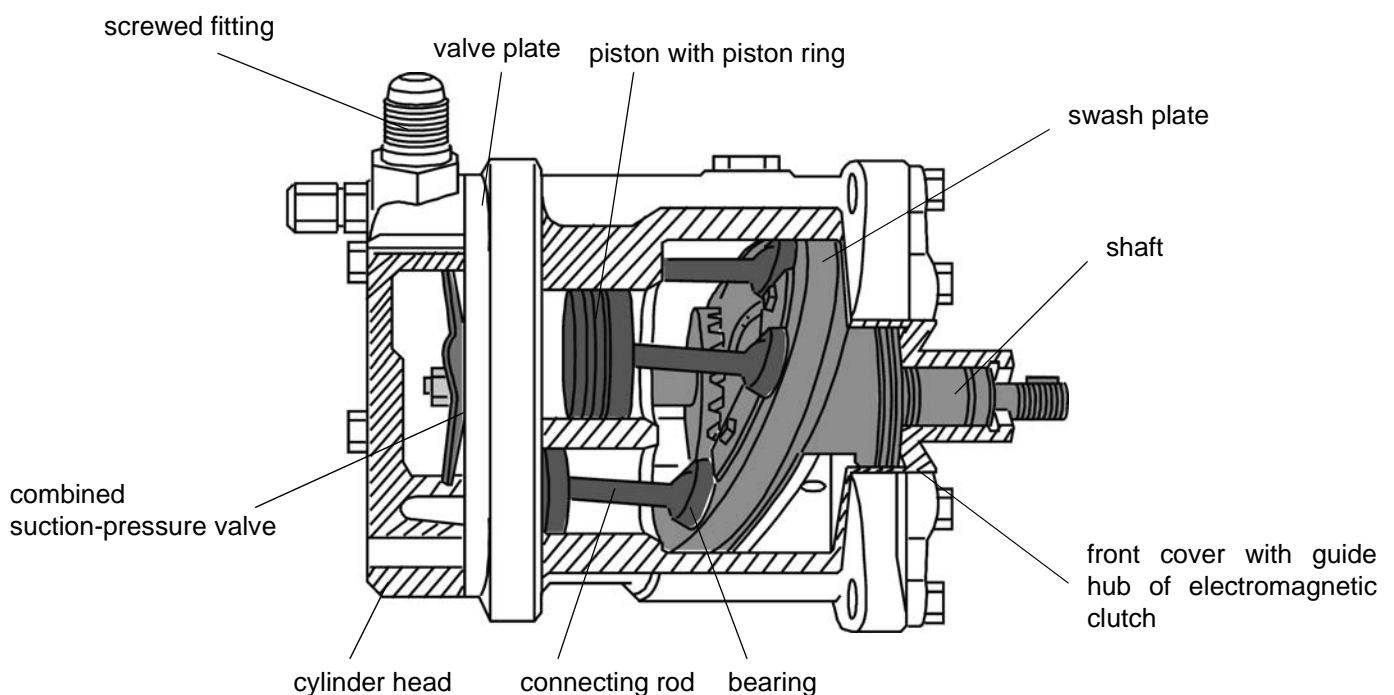


Fig. 4-5. Cut-away view of an axial piston compressor

4 Components of the Refrigeration Cycle

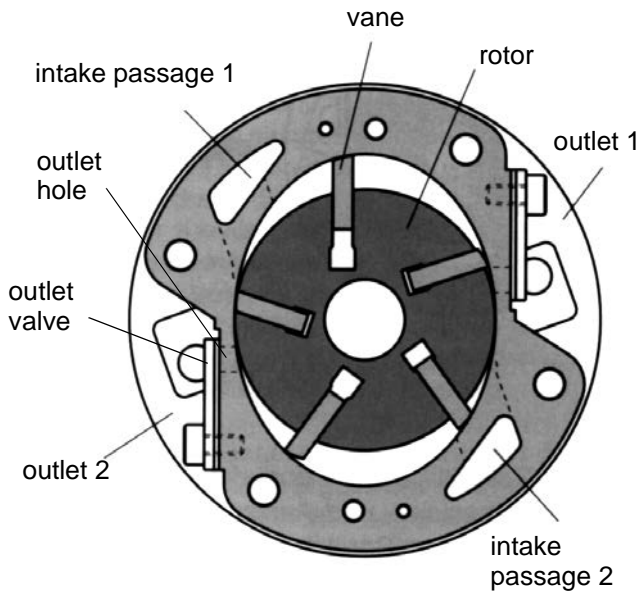


Fig. 4-6. Rotary vane compressor with 5 vanes

An advantage of the rotary vane compressor is the very good mass balance enabling high speeds and quiet running. Since the vanes are not able to seal properly until higher speeds are reached, the compressor starts up at reduced load. Moreover, this type is more compact than comparable swash-plate compressors.

A drawback is the higher drive energy (higher friction and leakage losses) required on account of the particular construction principle. Since these losses are converted into heat, the final compression temperature is higher than in swash-plate or scroll compressors. To achieve good sealing action, rotary vane compressors are operated with a higher oil content (up to 10 % oil in the refrigerant) and more viscous lubricants. A high proportion of oil in the cycle results in a deterioration of the heat transfer properties in the heat exchangers. In order to reduce this oil content, an oil separator is often integrated in the compressor.

4.1.4 Scroll compressor

The scroll technology is a simple compression concept that was patented in 1905 for the first time. A scroll is an involute spiral that when paired with a mating scroll member causes a series of crescent-shaped gas pockets between the two elements to be formed.

Method of operation

Upon compression, one spiral remains stationary (fixed scroll), while the other spiral orbits (not rotates!) around this first member. As the spiral motion continues, the gas pockets created between the two members are slowly moved toward the center of the scroll as the pockets become smaller in volume. When the pocket reaches the center of the fixed scroll member, the gas which is now under high pressure is forced through the discharge point

provided there. Since several gas pockets are compressed at the same time, a highly uniform process is the result. Suction (at the outer portion of the spirals) and discharge (at the inner portion) are almost continuous.

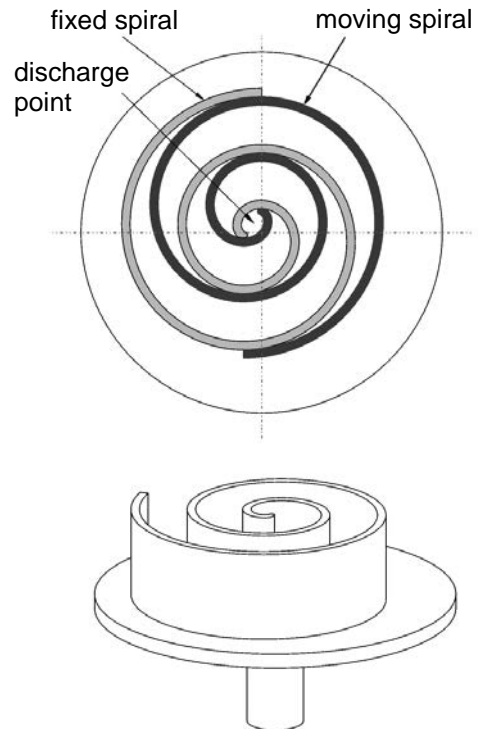


Fig. 4-7. Scroll compressor

Compression process

The compression process is shown in detail in Fig. 4-8.

- 1 Compression is achieved through the interaction of an orbiting and a stationary spiral. While one spiral orbits around the other, gas is drawn in from the outside of the scroll.
- 2 The inlet passages are sealed off while gas is drawn into the spirals.
- 3/4 As the spiral motion continues, the gas is compressed in two gas pockets that become smaller and smaller in volume.
- 5/6 When the compressed gas finally reaches the opening at the center of the scroll, it is fully pressurized and discharged.

During operation, all six gas pockets are in various stages of compression at all times, which is why the suction and discharge process is nearly continuous.

At the start-up a metallic noise can be heard for a brief moment, which is caused by the initial contact of the spirals and is totally normal. When the unit is switched off the compressor may run backward for a very brief time while the internal pressures are balanced. To prevent this from

4 Components of the Refrigeration Cycle

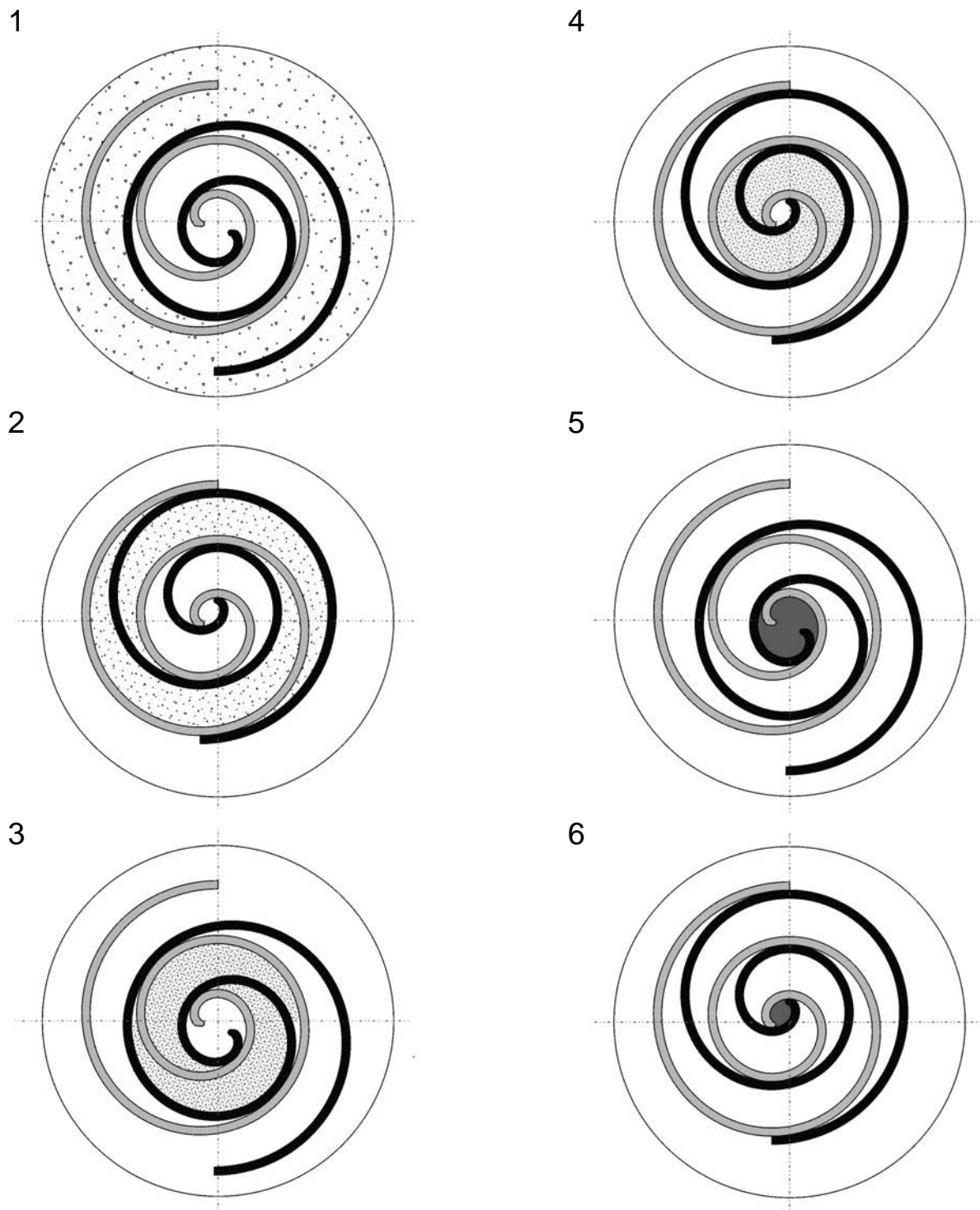


Fig. 4-8. Compression process in a scroll compressor

happening, a check valve is installed in many cases.

Advantages

Due to the design of the scroll, the internal compression elements always start up under no load conditions therefore exhibiting excellent start-up properties. The scroll does not require any operating valves and permits considerably higher speeds than reciprocating compressors. For predefined cooling capacities this results in smaller compressors that are more favorably priced requiring less mounting space. Good efficiency can be attained at higher

evaporating temperatures and speeds.

Owing to the absence of operating valves and - in some versions - the possibility to displace the spiral in axial and radial directions, this compressor type is more insensitive to drawing in moisture and dirt particles.

Disadvantages

A decisive disadvantage of the scroll compressor lies in a reduced volumetric efficiency at lower speeds. To compensate for the associated drop in cooling capacity, the

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scroll compressor must be operated at a higher transmission ratio than 1.5 when the engine is idling. At a high engine speed, compressor speeds of more than 10 000 min^{-1} are attained.

4.1.5 Screw compressor

In the last few years a dramatic development has taken place in the area of screw compressors. This type is increasingly used for air conditioning applications with a cooling capacity of approx. 20 kW and higher. This compressor is also used for applications in buses.

The two-shaft screw compressor comprises two differently shaped rotors (rotary pistons), that rotate in an enclosed casing without making contact with one another. The main rotor has convex lobes (teeth) in the transverse plane, whereas the gate rotor has concave lobes. The typical design today consists of 4 lobes on the main and 6 lobes on the gate rotor. In one revolution of the rotors the gaps between the lobes change as a result of the screw shape in such a way that they increase on the one end (suction) and decrease (compression and discharge) on the other end. The compression ratio is designed in by fixed control edges. As the screws are oil-sealed with respect to each other, a great amount of oil enters the cycle. In most of the cases it is retained by an internal oil separator.

Another design uses single-shaft screw compressors consisting of a single screw rotor and two discs with teeth.

Advantages

- high speed possible
- pressure ratio of 25 - 30 possible
- few moving parts
- no valves
- no unbalanced mass, little vibration
- low weight
- more compact design
- maximum coefficients of performance in applications with optimum pressure ratio

Disadvantages

- oil injection necessary for sealing action, hence high energy requirement
- poor part-load behaviour

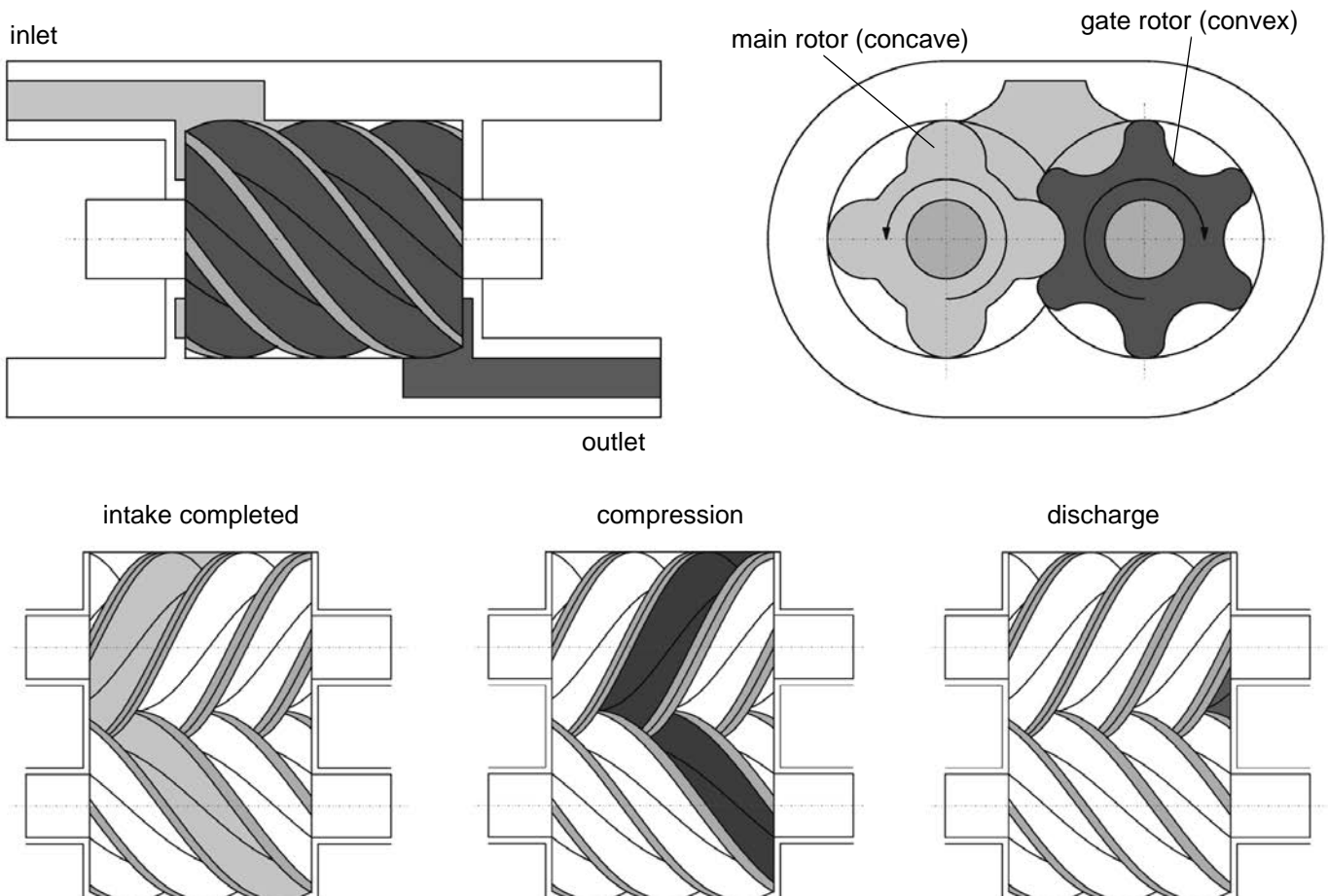


Fig. 4-9. Screw compressor, schematic view

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4.1.6 Compressor drives

Open compressors are driven by electric motors, combustion engines or hydraulic motors via a coupling or belt drive. For a reliable interaction between compressor and motor/engine great importance need to be placed on the proper **alignment** of the shaft ends with respect to one another (in true alignment with drive coupling, parallel with belt drives). In reciprocating compressors a flywheel effect is to be provided depending on the number of cylinders, type of power control and particular application. In the case of V-belt pulleys this flywheel effect is in most of the cases adequate on account of the required grooved pulley face.

As belt pulley either V-belt pulleys (1 or 2 grooves) or V-ribbed belt pulleys (typically 3 - 8 grooves) are used.

V-belt pulley with 1 or 2 grooves



V-ribbed belt pulley with 6 grooves

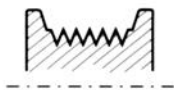


Fig. 4-10. Types of V-belt pulleys

For the **initial belt tension** it is imperative that the values specified by the compressor manufacturer are complied with. In the example shown in Fig. 4-11 the force resulting from the belt tension at the point of application of force must not exceed the value $F_{\max} = 2\,750\text{ N}$. If the point of application of force is displaced toward the front (L_1) the maximum permissible force diminishes depending on the distance from the point of application of force.

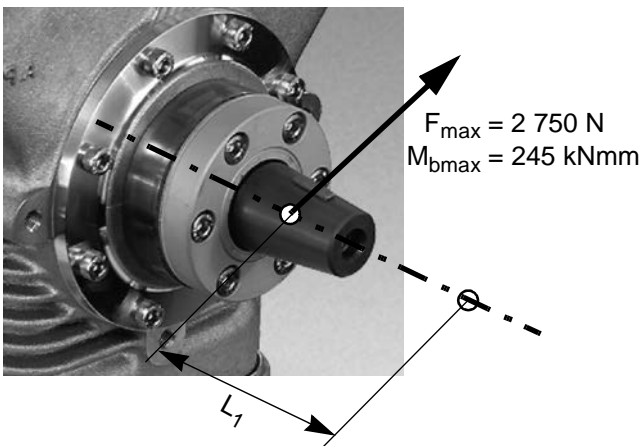
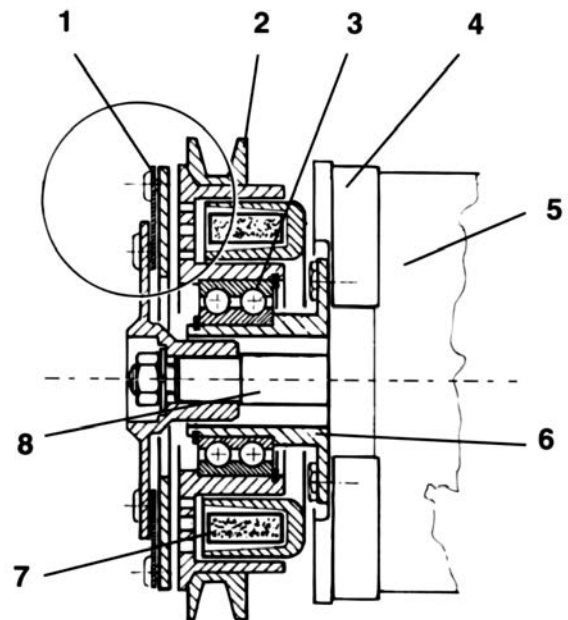


Fig. 4-11. Initial belt tension (Bock)

Belt tension needs to be rechecked after a given run-in period. Any load on the compressor shaft may only be

applied in the directions specified by the manufacturer. The belt pulley should be mounted as closely to the compressor bearing as possible to minimize bearing load. Damage to the compressor may be caused by improperly designed belt drives, untrue belt running or extremely high tension forces.

Fig. 4-12 shows a cutaway view through a non-actuated **magnetic clutch**. If no voltage is applied to the clutch field coil (7) there is no contact between the clutch front plate (1) and the belt pulley (2). The clutch plate is pushed away from the belt pulley by spring action. In the idling mode, the belt pulley turns with the bearing (3), the compressor is not operating. If voltage is applied to the solenoid (12 or 24 V), a magnetic field is created drawing the clutch plate toward the pulley. The clutch plate and hence the compressor shaft (8) turn with the pulley, the compressor starts up.



- | | |
|----------------------|--------------------|
| 1 Clutch front plate | 5 Compressor body |
| 2 Belt pulley | 6 Bearing support |
| 3 Bearing | 7 Coil |
| 4 Compressor head | 8 Compressor shaft |

Fig. 4-12. Magnetic clutch

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While the A/C is in use, the magnetic clutch is controlled by the freeze protection thermostat or pressure regulators.

The magnetic clutch is generally designed to last over the entire service life of the compressor. Insufficient supply voltage (insufficient coil power), excessive pressure in the A/C system (high cycling rate of the pressure regulators), oil-contaminated surfaces or incorrect adjustment of the clearance between belt pulley and driving plate, result in slipping and premature wear of the magnetic clutch.

4.2 Condenser

The condenser consists of tubes and fins that are tightly bonded together in order to achieve a large heat exchanger surface and optimum heat transfer. Ventilation is provided by a fan or ambient airflow at vehicle speed.

4.2.1 Condenser without subcooler

A frequent condenser type is the **tube-and-fin condenser**. Depending on the application, the tubes are made of aluminium or copper, the fins of aluminium.

The refrigerant inlet into the condenser is always located at the top so that the liquid refrigerant created during condensing can collect at the bottom of the heat exchanger and the expansion valve is supplied with pure liquid. The outlet port is mostly somewhat smaller than the inlet port due to the fact that the liquid refrigerant has a higher density.

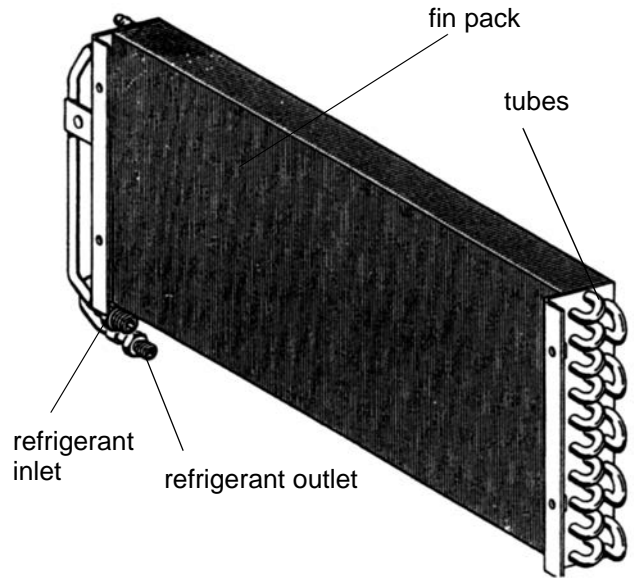


Fig. 4-13. Tube-and-fin heat exchanger

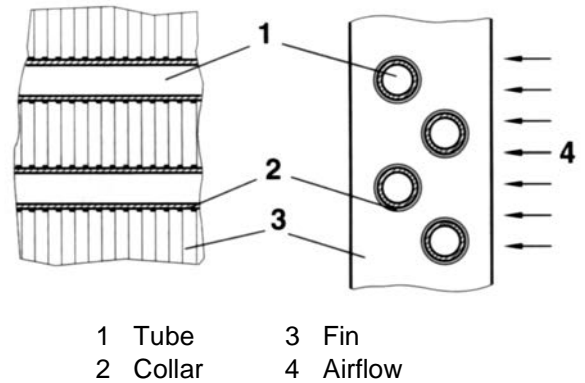


Fig. 4-14. Construction of the tube-and-fin condenser

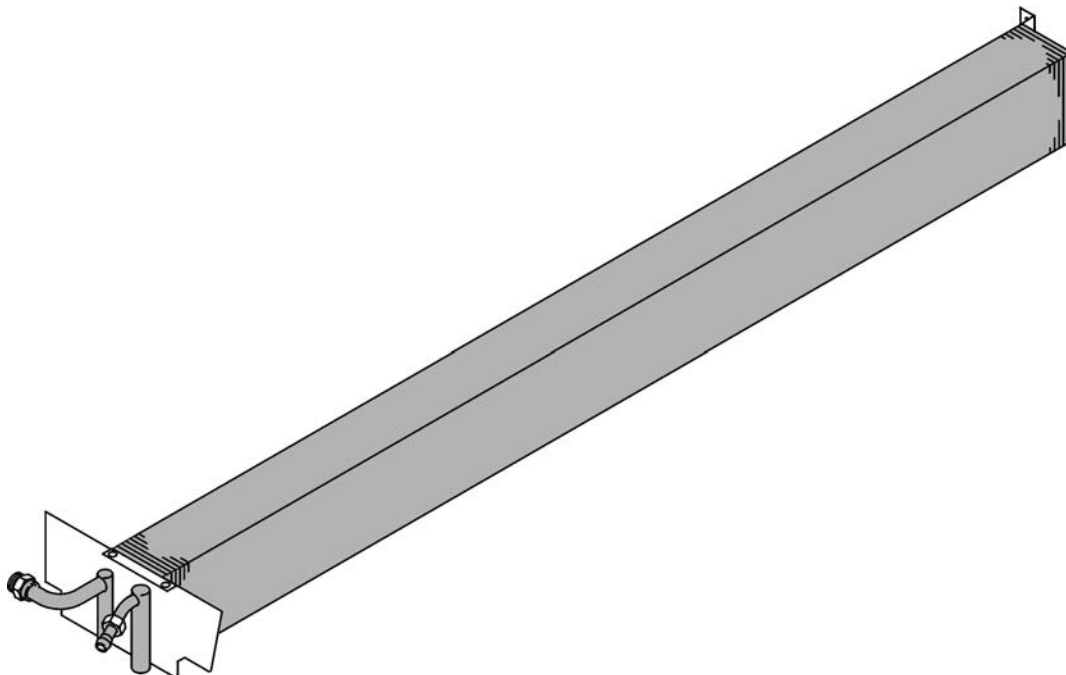


Fig. 4-15. Example of a bus-installed condenser

4 Components of the Refrigeration Cycle

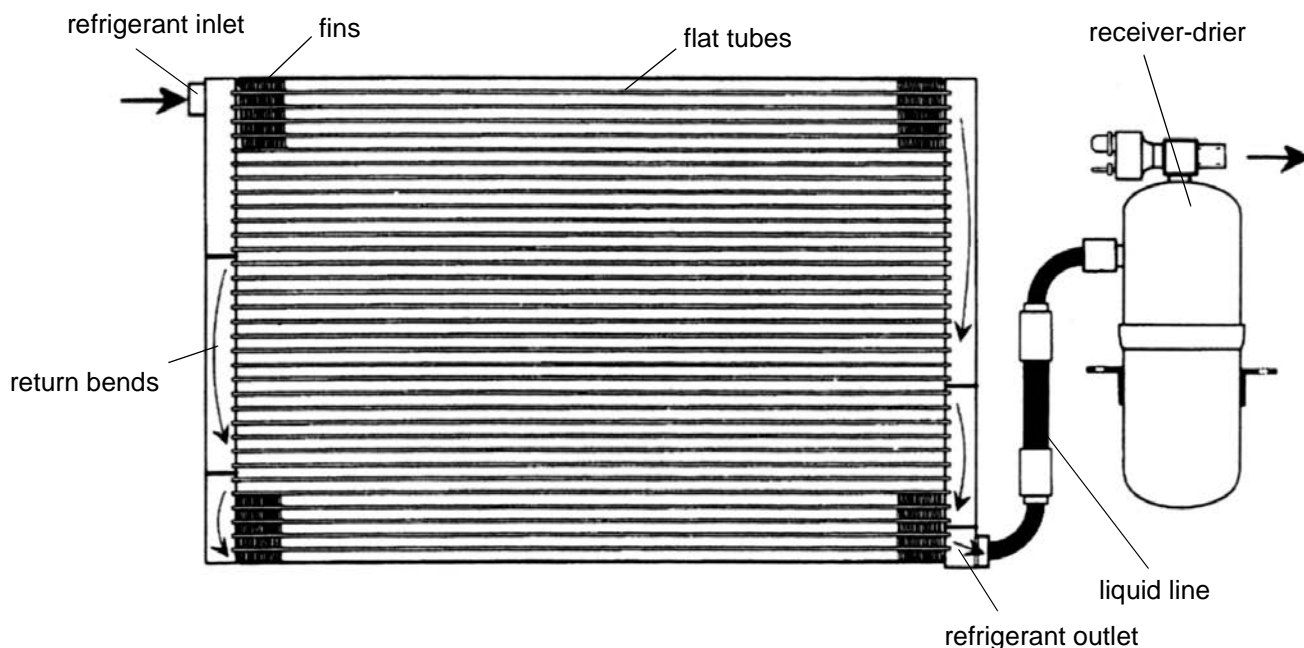


Fig. 4-16. Parallel flow condenser

Nowadays, **parallel flow condensers** are used in passenger car A/C systems (see Fig. 4-16). They are constructed entirely of aluminium.

The refrigerant is first distributed and then flows through multiple flat tubes arranged in parallel. In return bends the refrigerant is collected, diverted and distributed to the next flat tubes. To ensure a high flow velocity, the flow cross-section becomes smaller and smaller toward the condenser end.

the condenser, it is good practice to eliminate the subcooling zone in the condenser. This can be accomplished by installing a high-pressure receiver-drier (see Fig. 4-16). The latter separates the liquid refrigerant from the residual vapor bubbles thus ensuring a liquid seal in front of the expansion valve.

Without a receiver-drier the liquid seal would have to be provided by the refrigerant backing up into the condenser (subcooling). However, the backed-up liquid refrigerant reduces the heat exchanger surface that can be utilized for condensing, the condensing pressure rises, system output and efficiency decrease.

Frequently, systems equipped with a receiver-drier are overcharged to such an extent that subcooling takes place despite the receiver. The receiver in this case only functions as a buffer reservoir for any refrigerant leaks that may occur. This subcooling does not result in any improvement of the system performance in most of the cases.

The condenser equipped with subcooler offers the possibility to improve the system performance by subcooling. At the condenser outlet, the liquid refrigerant must be separated from the remaining refrigerant vapor by means of a receiver or receiver tube. The liquid then passes again through the heat exchanger assembly and is subcooled in so doing. The phase separation is necessary as otherwise the condensing zone would move to the subcooler and any subcooling would be impossible.

The complete condenser surface can be utilized for the heat rejection and condensing process enabling a higher cooling capacity and improved system efficiency to be achieved.

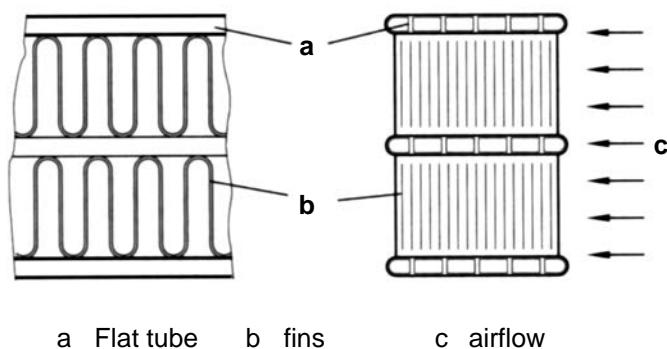


Fig. 4-17. Construction of the parallel flow condenser

This technology offers a higher heat exchange rate at identical space requirements as compared to traditional tube-and-fin heat exchangers.

A sufficient cooling air flow and a clean surface are prerequisite for a trouble-free operation of the condenser.

4.2.2 Condenser with subcooler

As heat transfer is rather poor in the subcooling zone of

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If no separate receiver is to be installed, phase separation can be effected in a somewhat larger header of the condenser with the liquid then being passed through the heat exchanger assembly.

4.2.3 Fan

Depending on the particular application, axial or centrifugal fans are used as condenser fans.

Axial fans can only be used if low resistances have to be overcome on the air side; this is the case if the air is drawn in and discharged freely, if the fin pack is not too deep and the tubes are not spaced too closely.



Fig. 4-18. Axial fan

The installation instructions with a view to a sufficient distance with respect to the fin pack, the intake port as well as the touch guard of favorable fluidic design must be complied with.

Centrifugal fans are suitable for greater pressure differentials on the air side (air filter, air distribution system). Compact evaporators with deeper fin packs (more than 4 - 5 tube rows) can be used.



Fig. 4-19. Centrifugal fan assembly with 2 fans

The impact of contamination of the fin pack on the air flow rate delivered is not as high as with axial fans.

4.3 Receiver

The cooling capacity of a plant decreases significantly if the liquid refrigerant contains vapor bubbles in front of the expansion device. In order to prevent this from happening the refrigerant must either be subcooled by some Kelvin degrees or else the remaining refrigerant vapor must be separated from the liquid refrigerant in a receiver.

In refrigeration plants with varying operating conditions, e.g. different evaporating temperatures and compressor speeds, the refrigerant charge in the evaporator may change considerably under certain circumstances. In particular with expansion valves with a flat characteristic curve, e.g. MOP valves, a load-dependent charge of the evaporator can be observed. Any refrigerant in the evaporator that is currently not required should - without any harmful effect - be stored on the high-pressure side. Also this situation speaks in favor of using a high-pressure receiver.

Another function the receiver may have to fulfill is that of a reservoir in the case of leaks.

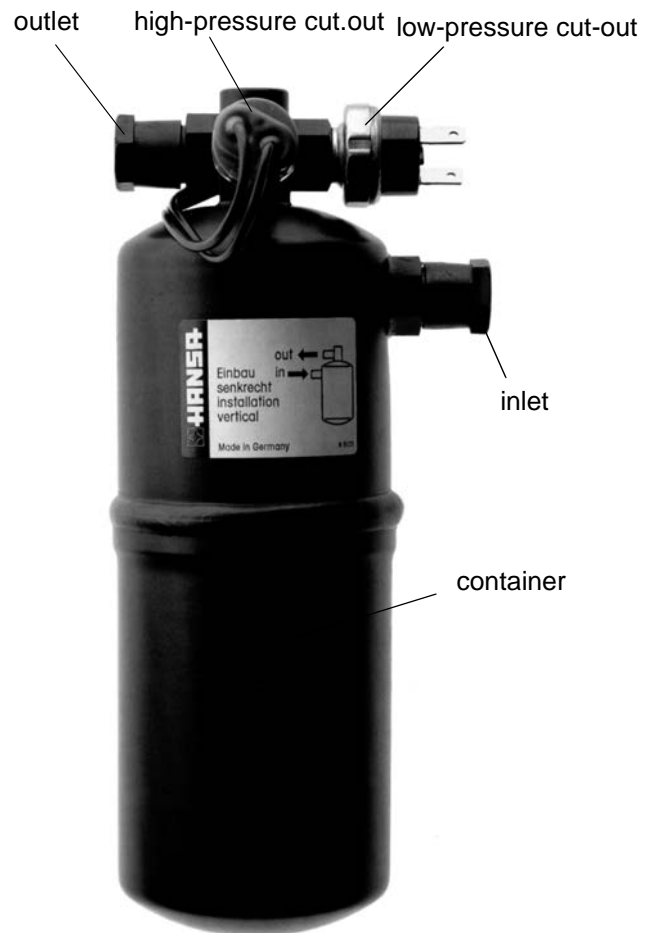


Fig. 4-20. Receiver in a passenger car application

When a receiver is installed, subcooling will be nearly 0 K provided the refrigerant charge is correct. In the case of a pressure drop in the liquid line, this will quickly result in the formation of vapor bubbles. If the system is operated with

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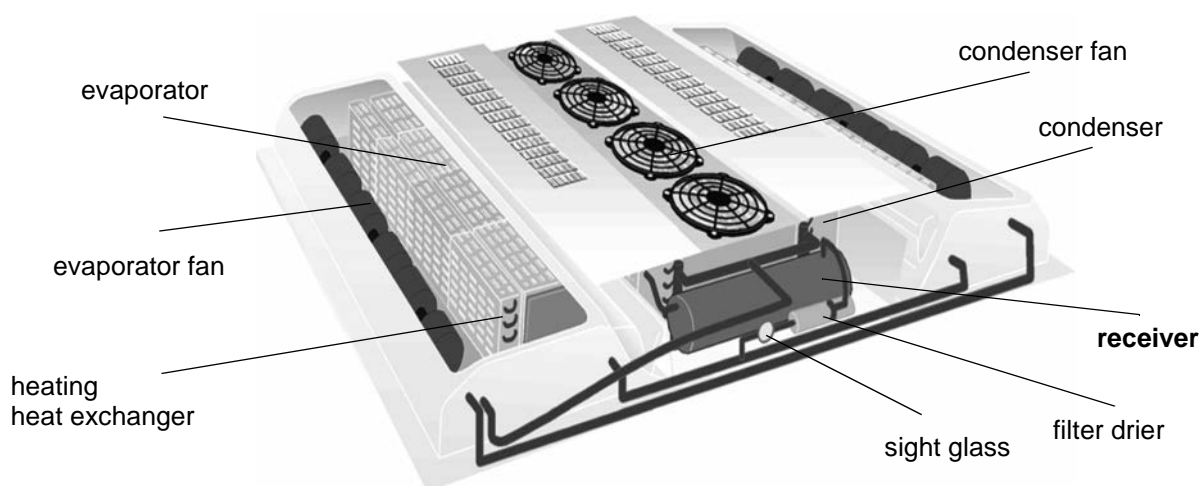


Fig. 4-21. Sample installation of a receiver in a bus-installed refrigeration system

a receiver, subcooling is only possible by installing an additional subcooling coil in the condenser, or by overcharging.

Function

Saturated vapor enters the side of the container. The liquid settles at the bottom and then flows to the expansion valve through a riser. Depending on the particular type, additional functions are integrated in the receiver container. The receiver may contain a drier insert (moisture adsorbance), a sieve (dirt filter), possibly two pressure cut-outs (one for shut-down if pressure is too high and one for shut-down if pressure is too low as a result of leaks). Some receivers also incorporate a sight glass or the service valve on the high-pressure side.

4.4 Filter drier

4.4.1 Functions of the filter drier

The functions of the filter drier in the refrigeration cycle are as follows:

- removal of water contained in the refrigerant
- removal of acid contained in the refrigerant
- filtering out dirt and other impurities

Even if utmost care is exercised during installation, commissioning or repair, moisture will still get into the system by way of air, refrigerant, oil and parts containing moisture (hoses). Acids can form resulting in damage to the compressor and lubricant. In addition, water promotes catalytic corrosion (copper plating effect) within the system. Dirt results in clogging of the throttle valve and increased abrasion in the compressor. Dirt in combination with moisture encourages the dangerous formation of acids.

4.4.2 Installation of the filter drier

The water adsorption property of a filter drier depends on the temperature. The lower the temperature the higher the water adsorption. With respect to the desiccation of refrigerants,

an installation in the intake vapor line would therefore be ideal. Here however, the flow velocity due to the oil return is relatively high, so that high-volume filter driers are required to keep the pressure loss within reasonable limits. Filter driers should therefore predominantly be installed in the liquid line, i.e. between the condenser and the expansion valve.

To achieve a better performance the liquid refrigerant should flow through the filter drier from top to bottom. The arrow indicated on the housing must always point into the direction of flow. The lower the flow velocity the higher the dwell time of the refrigerant in the filter drier and the moisture removal performance.

Apart from pure filter driers also combinations featuring an integrated sight glass and/or liquid receivers are available. Apart from the drier core, the enclosure also contains a given volume of empty space designed to store liquid refrigerant.

Always install filter driers that are compatible with the refrigerant being used.

With the system in the open condition, the drier material will immediately extract moisture from the ambient air and will possibly already be saturated before it is installed in the system. The filter drier should therefore be the last component to be fitted, i.e. directly before commissioning. The protective caps provided on either side must not be removed until just before installation.

4.4.3 Design of the filter drier

The desiccant material is either provided as a loose fill or in the form of a sintered solid block. With smaller filter driers the desiccant is contained, inaccessibly, in metal capsules (see Fig. 4-22). In large-capacity refrigeration systems and in the case of cleaning filters, screw-type capsules are used so that saturated or contaminated cores can be replaced.

4 Components of the Refrigeration Cycle

Three desiccants are primarily used for the drying of CFC refrigerants:

- Activated alumina Al_2O_3
- Silica gel SiO_2
- Linde molecular sieve LMS

They differ in their acid and/or water adsorption capability. Aluminium oxide has a high acid adsorbing capacity, silica gel a higher water adsorbing capacity. LMS has a high water adsorption and a medium acid adsorption capacity. For this reason, the solid matter core often contains two and even all three of these substances.

As a rule, the “filter drier” component consists of a cylindrical housing. In many cases, the ports for refrigerant inlet and outlet are located on the face of the unit.

A pressure spring inside the housing is designed to retain in place either a solid matter core (sintered) or a bed of loose fill beads. To prevent drier particles from entering the refrigeration cycle, a filter cloth reinforced by a perforated steel ribbon is provided at the outlet of the drier. For this reason, the arrow indicated on the housing must always point into the flow direction of the refrigerant, i.e. into direction of the throttling device.

The filter drier shown below uses a combination of all three desiccants. The main desiccation process is effected by a hollow consisting of sintered silica gel beads. In this hollow molecular sieves and aluminium oxide are embedded in an abrasion-resistant manner. These substances accomplish the final desiccation and the binding of acids. Filtration is carried out through the fine-porous solid block with a very large surface.

The drier size depends on the capacity of the refrigeration plant (refrigerant charge). Depending on the design, the drier may have a holding capacity of 6 - 10 g of water.

Vehicle shocks and engine vibrations greatly affect the desiccant material. Under no account must the material decompose under these conditions.

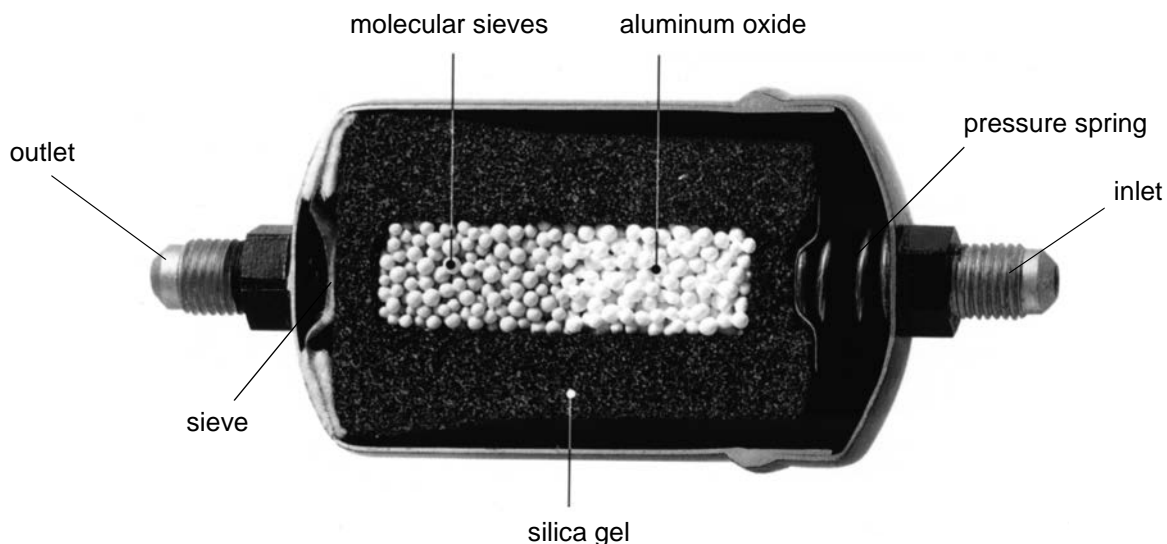


Fig. 4-22. Sectional view of a filter drier with solid core block

4.5 Sight glass

Trouble-free operation of throttling devices can only be ensured if liquid refrigerant is supplied for expansion. If no liquid receiver is installed, this is only the case if the refrigerant is sub-cooled. A sight glass is provided enabling the refrigerant flow to be checked. It is expedient to install it in the liquid line directly ahead of the expansion device.

Due to the fact that the refrigerant is colorless, no liquid flow can be seen in the sight glass if the refrigerant is adequately sub-cooled. If the refrigeration system is completely evacuated, no refrigerant can be seen in the sight glass either. A pressure test also serves to verify the refrigerant charge.

If no subcooling exists before the expansion valve, i.e. if the refrigerant contains vapor bubbles, this can be identified by bubbling in the sight glass

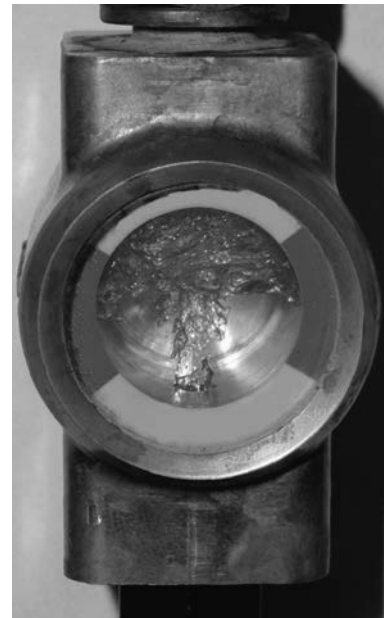


Fig. 4-23 Bubbles in the sight glass due to refrigerant shortage

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As a rule, vapor bubbles are an indication of a low refrigerant charge in the system. However, vapor bubbles can also be caused by pressure loss in the lines, shut-off devices, filters, driers, etc. In a severely contaminated filter drier, the pressure drop may be so significant that even though subcooling exists ahead of the filter drier, bubbles can be seen in the sight glass. Vapor bubbles can also be produced by too much heat from the surroundings entering the liquid line.

If the sight glass turns black from the inside, this is a sign of deterioration of the lubricant due to excessively high operating temperatures.

Usually sight glasses contain moisture indicators. You can tell by the color of the indicator whether the moisture content (water content) of the refrigerant is unacceptably high. Every manufacturer uses different colors. If a color change occurs, the filter drier is saturated with water and needs to be replaced.

For the various refrigerants specific indicators must be used, as otherwise the color change will not take place at the defined water content.



Fig. 4-24. Sight glass with moisture indicator

In typical passenger car refrigeration systems the sight glass is often dispensed with due to cost reasons.

4.6 Throttling devices

4.6.1 Temperature-controlled expansion valves

These valves, commonly referred to as **thermostatic expansion valves** are some of the most widely used expansion devices. Thermostatic expansion devices are superheat regulators keeping the superheated refrigerant vapor at the evaporator outlet constant by changing the valve lift. Thermostatic expansion devices can be used for all evaporating temperatures and all evaporator types.

With thermostatic expansion valves a distinction is made between thermostatic expansion valves with internal and external pressure equalization.

4.6.1.1 Thermostatic expansion valve with internal pressure equalizer

The main components of a temperature-controlled expansion valve (TEV) are the housing including seat and pipe connections, the valve cone, the regulating device (metal diaphragm or metal bellows), the capillary tube, the temperature sensor, the regulating spring (setpoint value spring), and the set screw.

The temperature or pressure system (including the sensor, the capillaries and the regulating device) is filled with a medium that responds to changes in temperature with a corresponding change in pressure. In the simplest case this medium is a refrigerant.

The temperature sensor must be fitted firmly and according to manufacturer instructions as close to the evaporator outlet as possible.

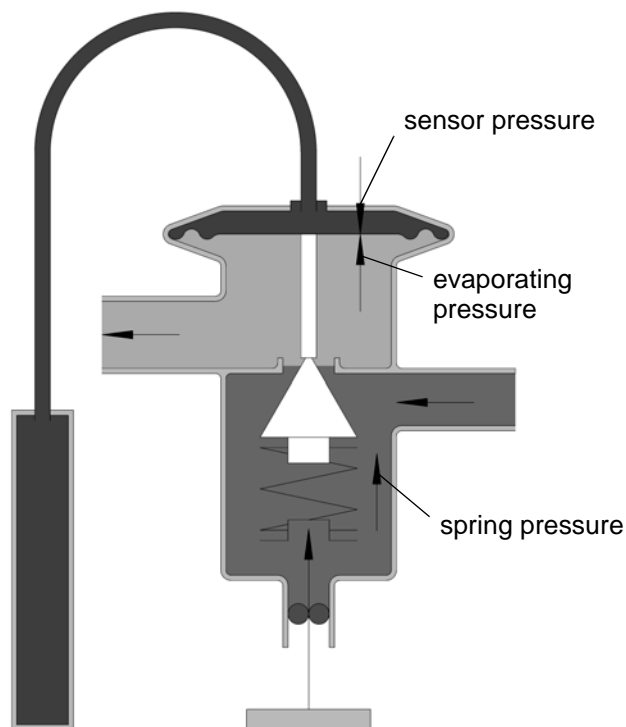


Fig. 4-25 TEV with internal pressure equalizer

The **operating principle** is governed by the interaction of three pressures acting on the control device (see Fig. 4-25):

The sensor pressure is an **opening force** and depends on the temperature of the evaporated refrigerant at the evaporator outlet and the sensor charge.

The evaporating pressure p_0 at the evaporator inlet as well as the pressure exerted by the regulating spring (setpoint value) are a **closing force**.

For as long as these three pressures are in equilibrium,

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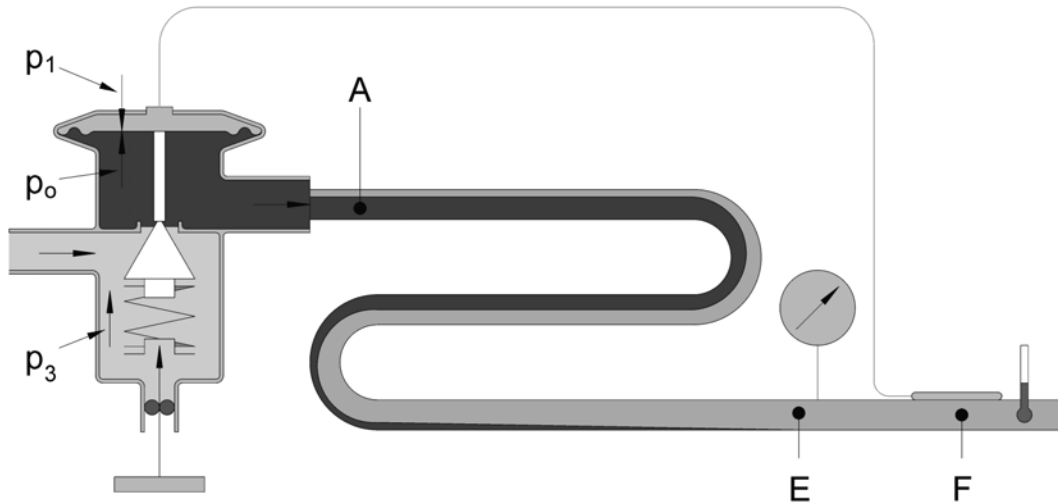


Fig. 4-26. Thermostatic expansion valve (internal pressure equalization) with evaporator

the open position and thus the free cross-section of the valve remain unchanged. If too little refrigerant flows into the evaporator (too much superheat), the sensor heats up, the sensor pressure rises resulting in a larger opening of the valve. A decrease in evaporating pressure has exactly the same effect.

A decrease in sensor temperature and an increase in evaporating pressure result in closing of the valve. If the compressor switches off, pressure p_o quickly rises and the valve closes. This applies only until associated heating causes the feeler pressure to dominate over closing pressures p_o and spring pressure p_3 .

The thermostatic expansion valve is a proportional con-

troller. Its controlled variable is the superheat of the refrigerant at the evaporator outlet.

Saturated vapor enters the evaporator at point A and should be completely evaporated by the time it reaches point E. Between E and the sensor mounting point F the refrigerant vapor (within the evaporator) is superheated, i.e. heated beyond its saturation temperature. Although this superheating segment reduces the evaporator performance, it is necessary for the stable operation of the regulating valve. Pressure p_3 of the regulating spring defines at which pressure difference existing between sensor and evaporating pressure the valve will start opening. This value is referred to as **static superheating**.

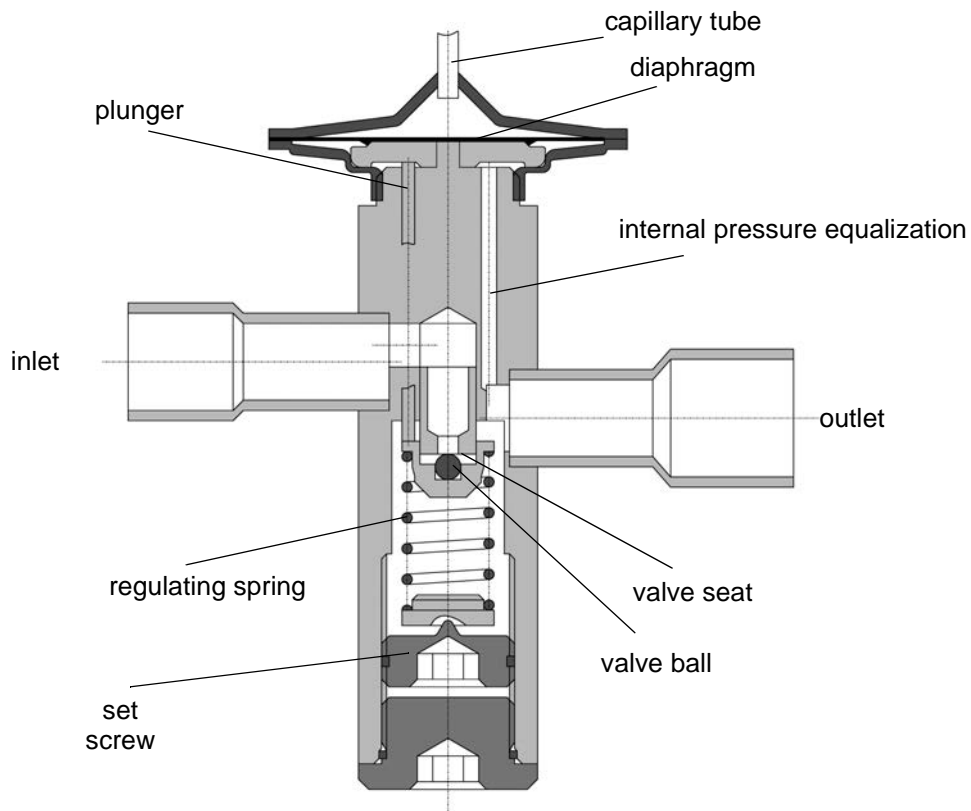


Fig. 4-27. Schematic illustration of an internal equalizing thermostatic expansion valve

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To operate the valve through its full operating range - from the start of the opening process through to the valve's rated capacity - a further increase in sensor pressure (at constant suction pressure), i.e. additional sensor heating (superheating) to overcome the increased spring pressure is required. This additional amount of superheat is referred to as **opening superheat**.

The sum of static superheat and opening superheat is called **operating superheat**. The superheat measured at the evaporator outlet is the operating superheat. Fig. 4-28 shows these relationships by means of the performance curve of a thermostatic expansion valve.

The superheat curve of the valve is so designed that the factory-set static superheat nearly constantly lies above the evaporating temperature. As a rule, thermostatic expansion valves should be operated at preset factory values. If a readjustment becomes necessary, adjustment may only be effected in small increments. Each readjustment results in a change of the superheat curve as a function of the evaporating temperature.

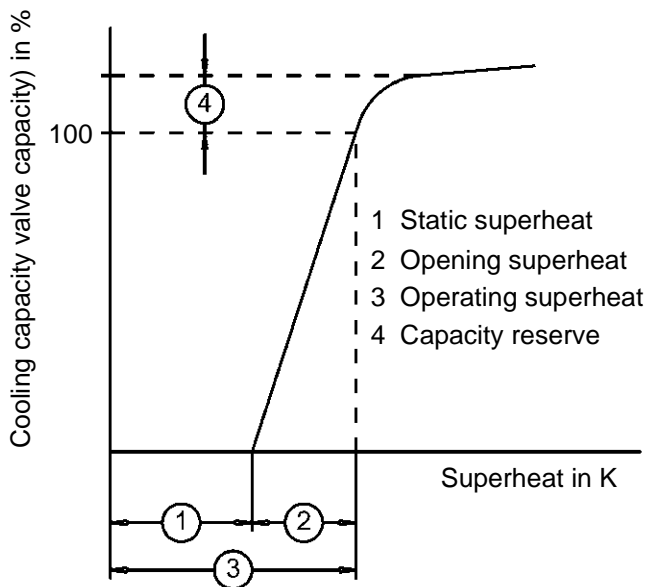
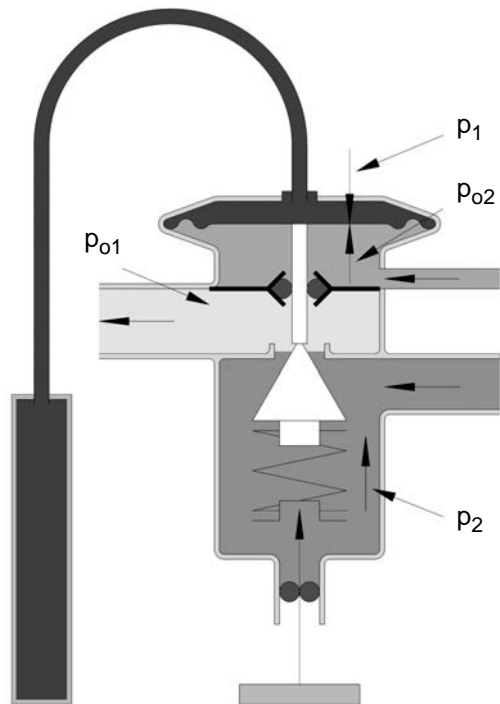


Fig. 4-28. Performance curve of a thermostatic expansion valve

If the operating superheat of a thermostatic expansion device (at high evaporator load) is, for example, changed from 10 K to 6 K, the danger of too little or no operating superheat in the part-load mode of the evaporator exists. The consequence may be damage to the compressor. In the worst case, the valve will not close anymore when the compressor is switched off. Refrigerant migration from the high pressure side to the low pressure side must be reckoned with. A critical point in this respect is an accumulation of refrigerant in the compressor or in the suction vapor line.

4.6.1.2 Thermostatic expansion valve with external pressure equalizer



- p_{o1} Evaporating pressure (evaporator inlet)
- p_{o2} Evaporating pressure (evaporator outlet)
- p_1 Sensor pressure
- p_2 Spring pressure

Fig. 4-29. TEV with external pressure equalization

Valves in which the pressure prevailing at the evaporator inlet acts on the control element (underneath the diaphragm) are referred to as valves with internal pressure equalization. They are primarily used in low-capacity systems where the pressure drop within the evaporator is relatively small. In the case of valves with internal pressure equalization, a pressure drop in the evaporator results in a higher superheat at the evaporator outlet and a performance drop in the system.

In **external equalizing thermostatic expansion valves** pressure p_{o2} prevailing at the evaporator outlet acts underneath the diaphragm. Pressure p_{o1} is separated by a partition wall with a sealed plunger lead-through. The superheat is regulated in accordance with the pressure at the evaporator outlet. Pressure drops in the evaporator or in a refrigerant distributor are thus compensated for.

Fig. 4-30 shows the schematic view of such a valve. In this example of a valve a coil-shaped temperature sensor is installed. It contains a liquid exhibiting very similar or identical properties as the refrigerant used.

The sensor pressure is "compared" - via a diaphragm - with the evaporator outlet pressure. The resulting pressure is transferred - via pins - to the valve setting unit

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which in turn adjusts the refrigerant flow rate. Throttling takes place in the calibrated borehole, atomization of the refrigerant is effected underneath.

The pressure equalizing tube must always be fitted in the direction of flow, after the temperature sensor, as in the case of leaks in the pin seal the valve cannot properly perform its regulating action. Leaks cause liquid droplets to

enter the suction vapor line through the pressure equalization tube. If the droplets hit the temperature sensor, the valve will regulate upon every single drop. The order of installation of the sensor is designed to prevent this from happening.

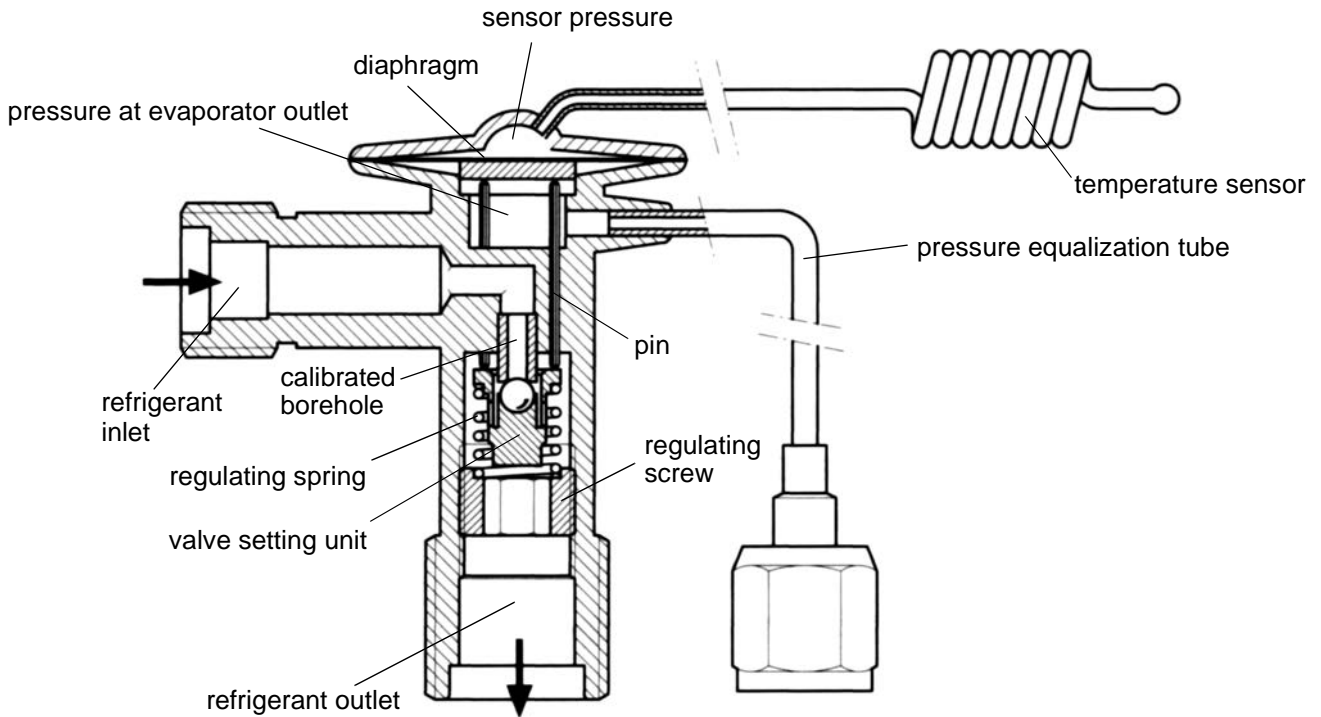


Fig. 4-30. TEV with external pressure equalization (corner valve)

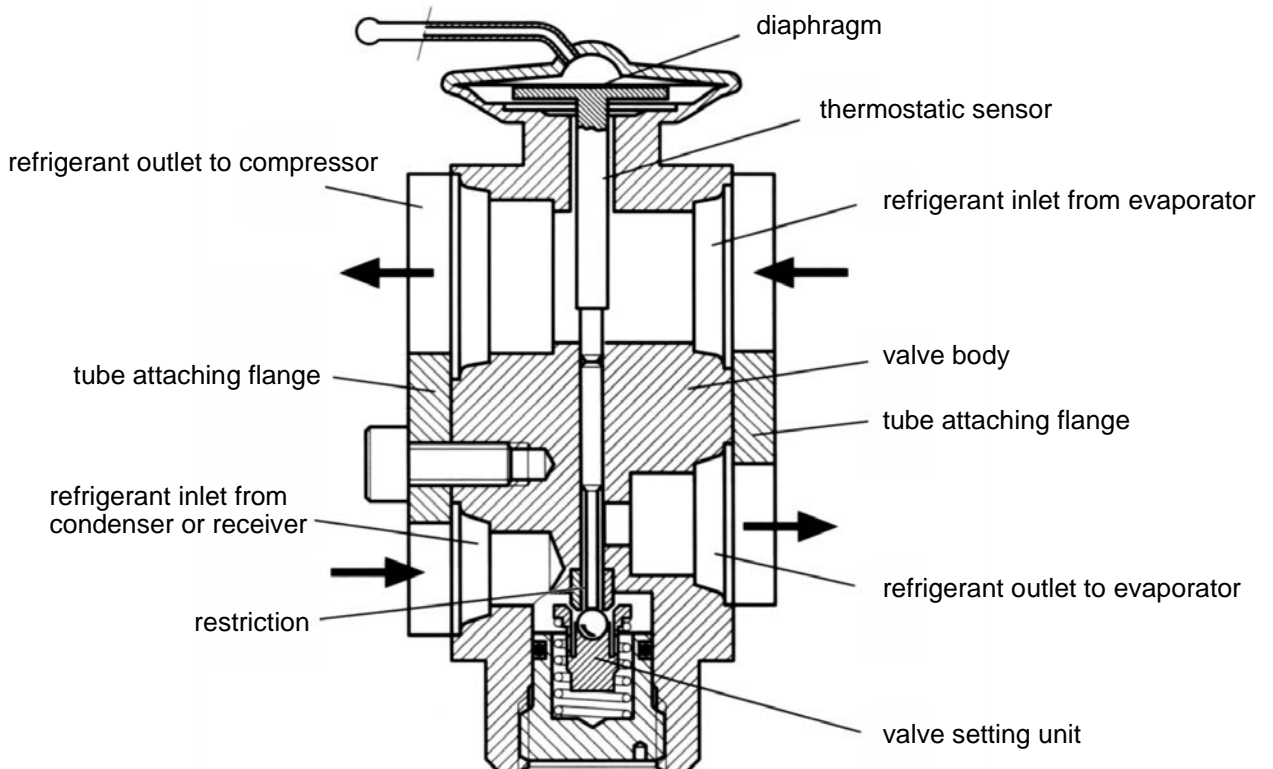


Fig. 4-31 TEV with external pressure equalization (block valve)

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4.6.1.3 Block valve

The demand for compact and robust components has led to a special version of the thermostatic valve with external pressure equalization for refrigeration systems in cars - the block valve (Fig. 4-31). The latter is fixed directly to the evaporator. The suction vapor line is routed directly underneath the diaphragm through the valve housing. The suction pressure underneath the diaphragm acts as a closing force. Through heat transfer the suction vapor temperature is transmitted to the diaphragm via the valve body and the thermostatic sensor.

A higher suction vapor temperature results in a higher pressure in the valve head causing the valve to open. The static superheat can only be adjusted in the lower part of the valve body by means of a special tool.

4.6.1.4 MOP valve

MOP stands for **Maximum Operating Pressure**. This refers to a thermostatic expansion valve limiting the evaporating pressure to a maximum value. As the heat load at the evaporator rises, the evaporating temperature rises and hence also the sensor temperature of the valve. Expansion devices with MOP contain a defined saturated vapor sensor charge which is completely evaporated at a predefined sensor temperature. If the temperature rises further, the pressure increase in the sensor is negligible.

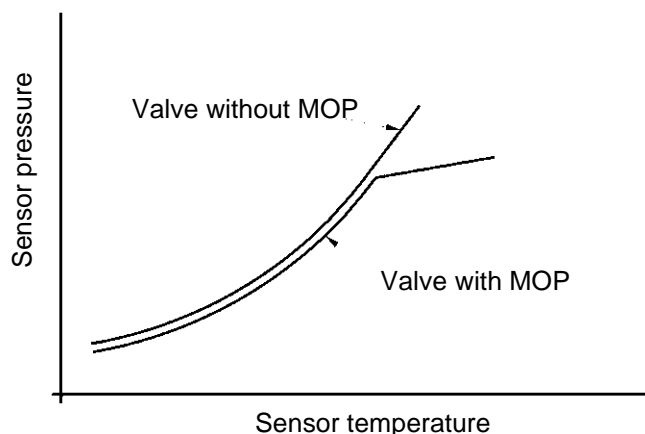


Fig. 4-32. Sensor pressure as a function of the sensor temperature

As a result, the valve cannot open any further. The evaporator is no longer supplied with additional refrigerant and

the evaporating temperature remains constant. If additional heat is supplied, only the suction vapor superheat increases further.

Pressure-limited valves are used whenever the power consumption of the compressor must not exceed a predetermined value. If a vehicle is left standing in the sun for a prolonged period, the interior temperature may possibly rise to 60 - 70 °C. Activating the A/C system would produce evaporating pressures of nearly 10 bar ($t_0 = 40$ °C). This leads to very high refrigerant flow rates and hence to a very high performance demand on the compressor. Apart from extreme loads on the mechanical driving elements of the belt drive the danger of stalling the combustion engine exists in the worst case scenario.

Therefore, MOP valves are always installed in refrigeration systems in cars. The MOP value is approx. 5 bar, which corresponds to a maximum evaporating temperature of 15 °C.

4.6.2 Expansion tube (orifice tube)

The **orifice tube** represents the simplest version of a throttle device. A short nozzle (e.g. with an inside diameter of 1.2 to 1.84 mm; length 38.8 mm) is embedded in a plastic housing. At the inlet and outlet one fine mesh strainer each is located functioning as a dirt trap. The cartridge - sealed with O-rings - is fitted into the refrigerant line leading to the evaporator.

In connection with the orifice, the term “bubble point“ is often used. This is the point at which the first vapor bubbles occur as a result of the pressure drop. The flow rate and the location of the bubble point is dependent on differential pressure, condensing pressure and subcooling. High pressure and high subcooling result in an increase in the refrigerant flow rate. A refrigeration cycle using an orifice can be compared with capillary tube injection in a refrigerator. In these systems the evaporator charge is not regulated but adjusted by the refrigerant charge. With appropriate dimensioning of the diameter/length relationship of the short nozzle any additional refrigerant supplied from the outside would accumulate in the evaporator. The refrigerant charge is therefore correctly adjusted if the heat load of the evaporator is such that no unevaporated residual liquid refrigerant enters the suction line. If the

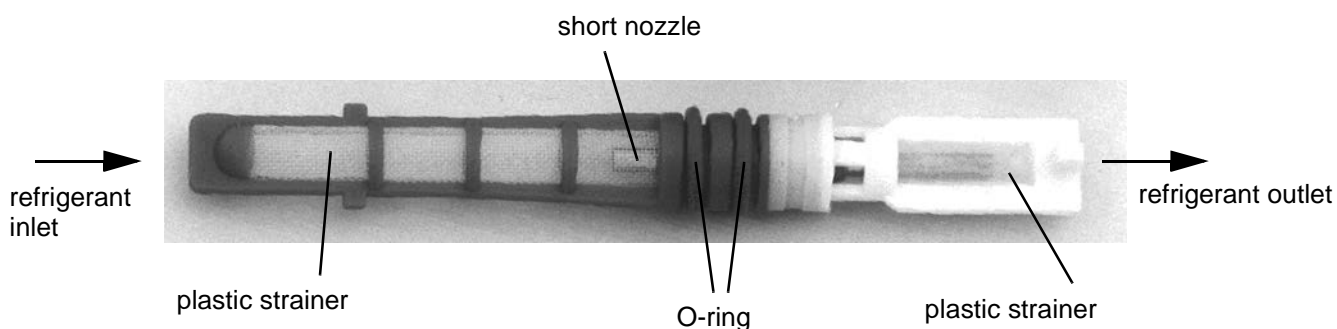


Fig. 4-33. Orifice tube

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heat load of the evaporator increases, the refrigerant will be evaporated before the evaporator outlet is reached. The refrigerant superheats at the remaining surface. Apart from the fact that the evaporator is not utilized to 100 % and the temperature at the pressure joint of the compressor increases as a result of the higher suction superheat, no further drawbacks are to be expected. However, operating phases are critical in which the heat load of the evaporator decreases (recirculated air mode/interior largely cooled). Liquid refrigerant from the evaporator enters the suction line. To prevent this liquid from being drawn into the compressor, a liquid collector (accumulator) needs to be installed downstream of the evaporator.

Regulation by means of an orifice is also referred to as throttle control.

4.7 Evaporator

4.7.1 Air-cooled evaporators

Apart from plate evaporators only **finned evaporators** are used as air-cooled evaporators in mobile refrigeration applications. These consist of copper-, aluminium- or steel tubes to which aluminium fins are bonded to increase the outer surface.

The fin spacing must be adapted to the specific application. In so doing it must be taken into consideration that air always contains a small amount of water in the form of vapor. If the air is cooled below the dew point, this water vapor condenses on the cold evaporator surface and is removed from the air. This water must be able to drain via a condensate line.

If the evaporating temperature lies below approx. $-3\text{ }^{\circ}\text{C}$, the condensed water will freeze on the evaporator surface. The ice covering the evaporator reduces the evaporator performance. The longer the evaporator is operated at such low temperatures the larger the fin spacings must be. The fin spacings range between 2.4 mm (car A/C evaporator) and 12 mm (evaporator for refrigerated vans). If the layer of ice is too thick the compressor must be switched off by a built-in freeze protection device.

The evaporator shown in the illustration consists of 5 tube circuits. The refrigerant leaving the expansion valve is distributed among multiple tubes. This is useful as the heat transfer surface is thus increased and the tube lengths are reduced (to minimize pressure drop). Moreover, less installation space is required. The vaporous refrigerant is collected again at the evaporator outlet and passed to the

compressor.

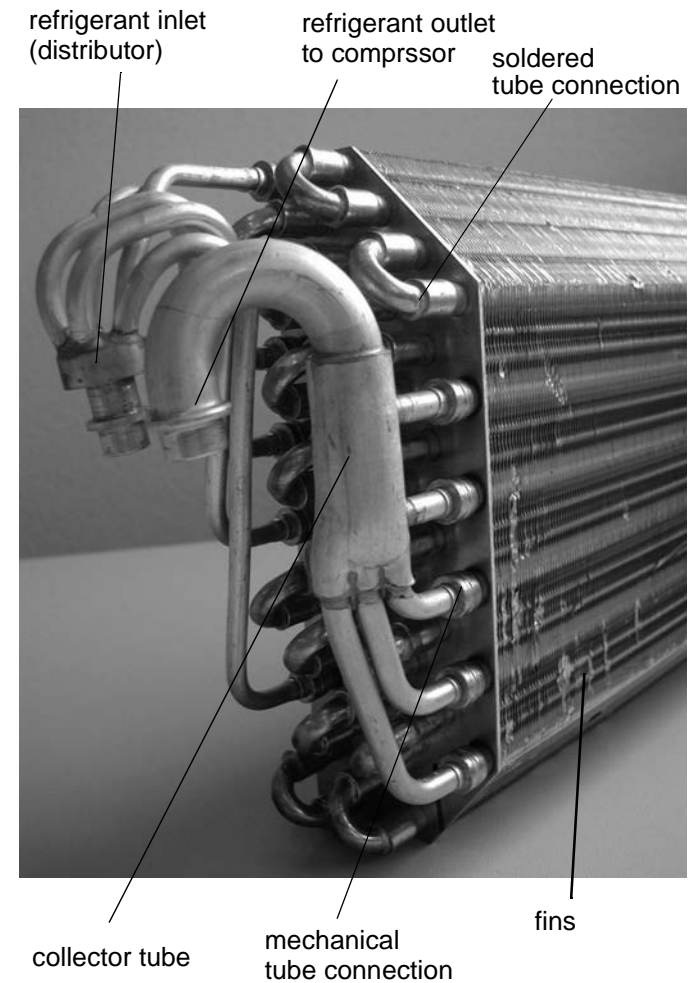


Fig. 4-34. Finned evaporator

The dimensions of this unit vary depending on the required capacity and the installation space available. U-shaped bent tubes are passed through the fin packs and then expanded to ensure proper positioning of the fins. Finally, the pipe bends and connections are soldered or pressed together.

For an improved heat transfer, the **plate** or **disc evaporator** (Fig. 4-35) have been developed, especially for A/C systems in cars.

The evaporator consists of an aluminium plate assembly whose plates feature a U-shaped cavity through which the refrigerant flows. The cavity is provided with a structure resulting in turbulences in the refrigerant flow and thus to an improved heat transfer. The refrigerant flowing through the inlet port is distributed by the distributor to the first plates. Once the refrigerant has passed through these plates it is collected and distributed to the subsequent plates. At the outlet the refrigerant is collected in the collector tube and then passed to the outlet port.

Ribs are located between the individual plates resulting in an increased surface and creating air turbulences (improved heat transfer).

4 Components of the Refrigeration Cycle

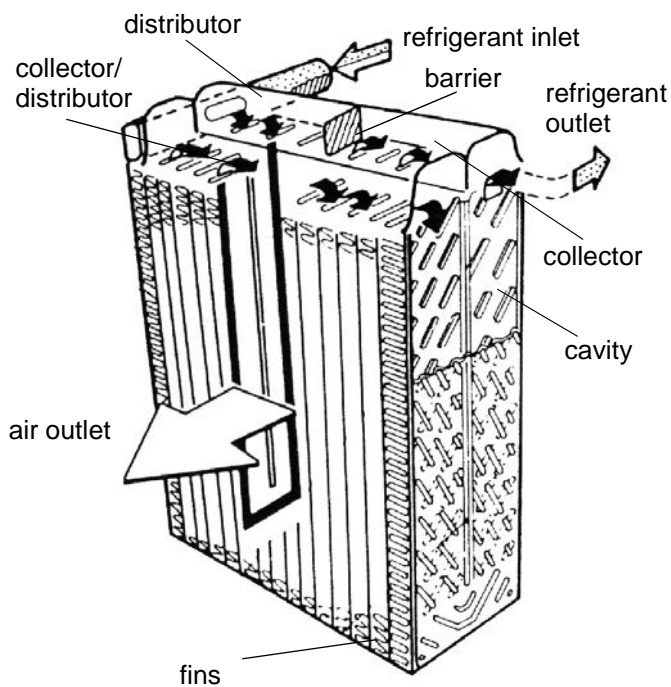


Fig. 4-35 Plate evaporator installed in a car

A common problem in vehicle A/C systems relates to “odors from the air-conditioning system”. The evaporator can under certain circumstances become a breeding ground for microorganisms whose metabolisms can produce offending odors. As the evaporator in an air-conditioning system is mostly inaccessible and therefore cannot be cleaned, it is provided with a special coating. This anti-microbial will inhibit the growth of microorganisms and also - owing to a water-repelling effect - improve condensate drain.

4.7.2 Liquid-cooled evaporators

In A/C systems for bus applications, systems with **indirect cooling** are also used. Here, a liquid (mostly a water/glycol mixture) serving as cooling medium is cooled in the evaporator of the refrigeration system. This mixture is then supplied to the cooling system heat exchangers by a pump.

Arguments in favor of this system are the simple regulation at the units to be cooled, leaks are less critical, the same heat exchangers can be used for heating and cooling, smaller refrigerant charges and lower pressure losses due to shorter refrigerant lines.

Drawbacks, however, are the additional expenditure for the system and the lower evaporating temperature (reduced refrigeration capacity) required.

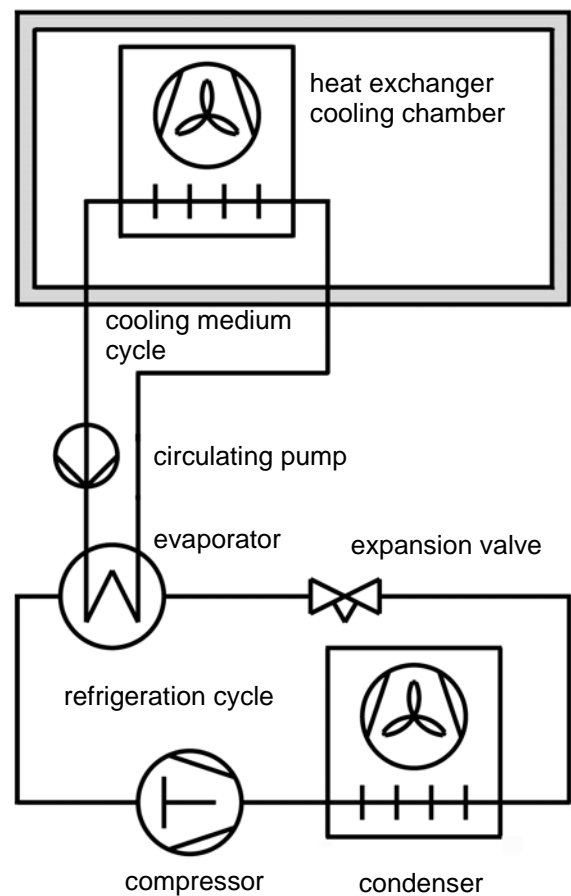


Fig. 4-36 Indirect cooling

In refrigeration systems for mobile applications, plate evaporators (Figs. 4-37 and 4-38) are used for indirect cooling. These evaporators consist of individual plates that are soldered or screwed together. The refrigerant flows through every second gap that is created between the individual plates, the cooling medium through the others.

Since high flow velocities are achieved in plate evaporators and a large heat transfer surface is provided, high performances can be attained with little space requirements.

A plate evaporator should always be installed in upright position. The coolant mixture consisting of liquid and vapor should enter the lower port. The evaporating process takes place in the vertical flow channels.

Larger and longer plates are better suited for evaporators because their longer flow path ensures that no liquid refrigerant emerges from the evaporator.

For the trouble-free operation of the system it is necessary that the brine circuit be completely vented. Any air trapped in the cycle results in a reduced cooling capacity and in addition produces running noise.

4 Components of the Refrigeration Cycle

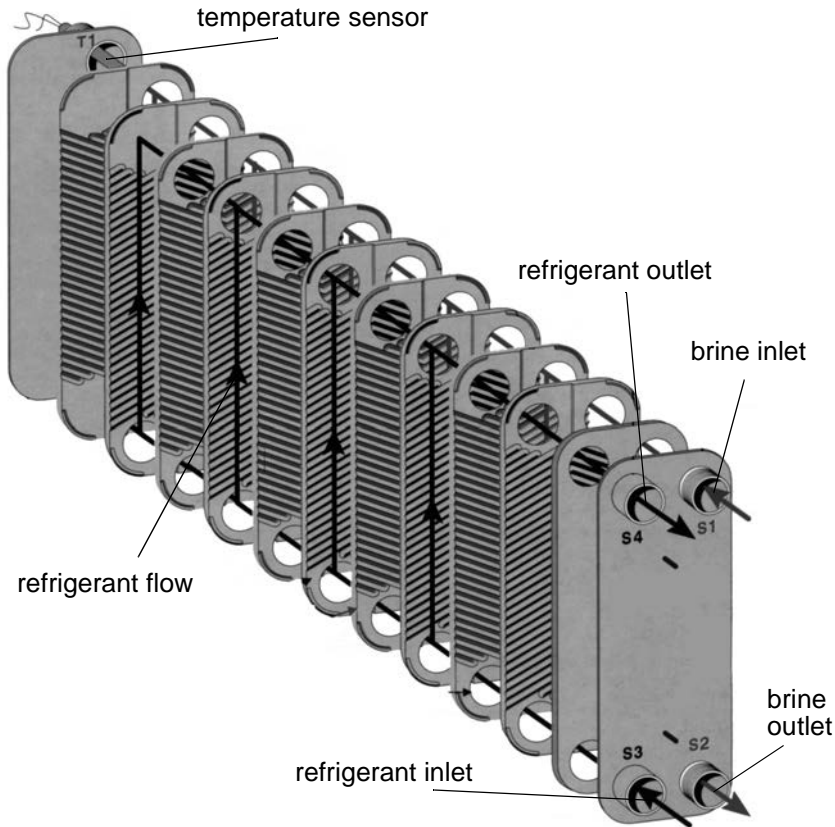


Fig. 4-37 Construction of a plate evaporator



Fig. 4-38. Plate evaporator

4.7.3 Refrigerant distribution

Proper distribution of the refrigerant in the evaporator is crucial for the trouble-free operation of the evaporator and the expansion valve. Correct refrigerant distribution ensures that the evaporator can be operated at a lower superheat. This improves the effective area for evaporation and thus the cooling capacity.

A uniform distribution of vaporous and liquid refrigerant among the individual circuits of an evaporator assembly can be achieved by installing an appropriate distributor.

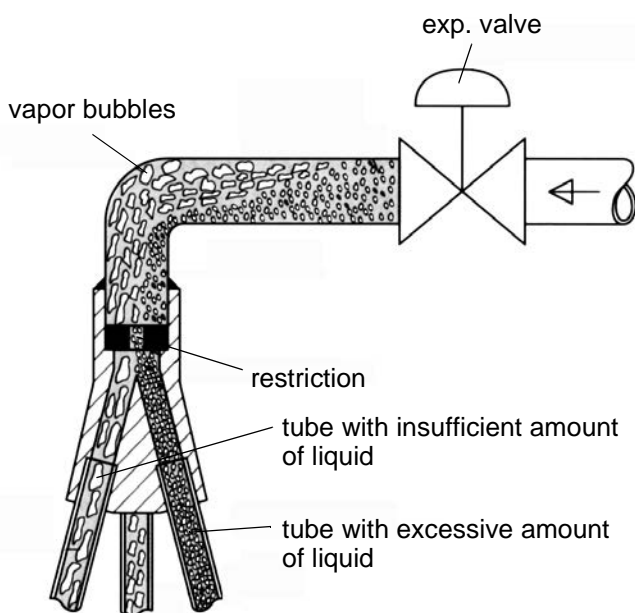


Fig. 4-39. Refrigerant distributor

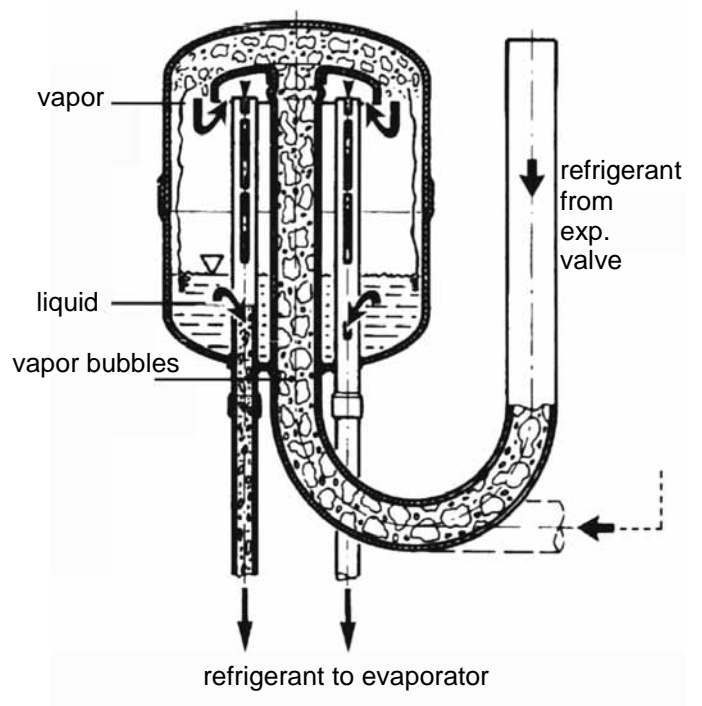


Fig. 4-40. Refrigerant distributor

4 Components of the Refrigeration Cycle

Fig. 4-39 shows a schematic view of a refrigerant distributor as used in a finned evaporator. Poor refrigerant distribution is experienced resulting from an incorrect installation position.

For optimum distribution of the refrigerant it is imperative that the distributor be installed in vertical position. Although this requirement was complied with in Fig. 4-39, the distributor is, however, located too close to a tube bend, which results in non-uniform refrigerant distribution.

Another distributor type is shown in Fig. 4-40. This type which serves as a buffer storage owing to its large capacity - is better suited for handling greater load fluctuations of the refrigeration system than the venturi type distributor.

In the case of plate heat exchangers a non-uniform distribution of the refrigerant among the individual flow channels may occur. Since the liquid refrigerant has a higher density, the centrifugal force causes more liquid refrigerant to reach the rear channels of the heat exchanger.

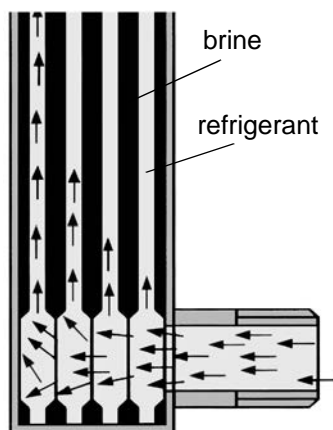


Fig. 4-41 Refrigerant distribution in the plate evaporator

This problem can be eliminated by using a special distributor. One possibility, for example, is an atomizer. This is a thin disc consisting of small metal spheres. The cavities between the individual spheres form a kind of sieve. The 'mister' is installed in the evaporator port on the refrigeration side and soldered to the pipe duct. The mixture of liquid particles and vapor leaving the expansion valve is therefore finely atomized before entering the evaporator and uniformly distributed among all flow channels.

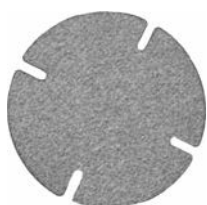


Fig. 4-42. Mister

4.8 Liquid separator

In systems with orifice, critical operating phases exist in which the heat load of the evaporator decreases (recirculating air mode/interior largely cooled). Liquid refrigerant escapes from the evaporator. To prevent this liquid from being drawn into the compressor, the liquid separator (accumulator) is installed downstream of the evaporator.

The flow velocity is reduced in that the volume is expanded. Liquid drops contained in the stream settle at the bottom. Together with the refrigerant also the lubricant is circulated. Oil return to the compressor, however, must not be interrupted. A "U" tube is integrated in the accumulator for this reason. At the lowest point of this "U" tube an oil sniffing hole (diameter approx. 1 mm) is provided. One open end of the U-tube within the accumulator extends into the vapor chamber, the other end into the suction line. Adequate flow velocities within the tube must be maintained to ensure that the oil or oil/liquid refrigerant mixture can be 'sniffed' from the lower portion of the accumulator.

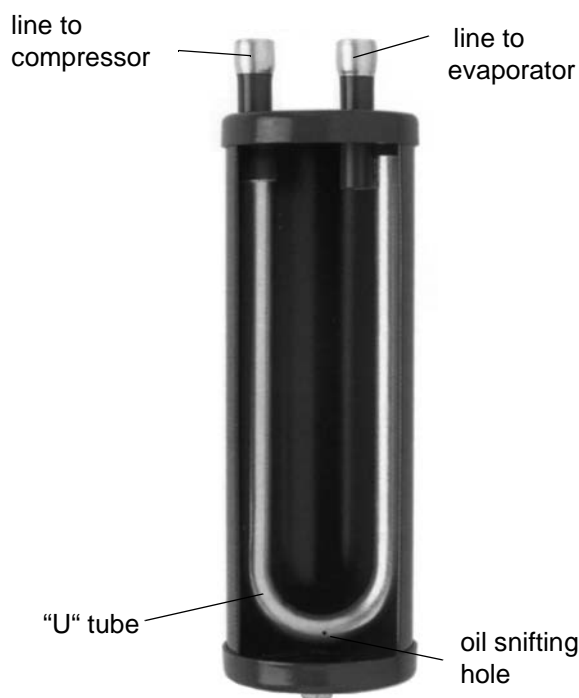


Fig. 4-43. Liquid separator

Another job of the accumulator is the storage of refrigerant in the case of overcharging or at part-load operation. In the lower portion of the liquid separator the desiccator material is often located and a strainer for the filtering of contaminants is provided above the oil sniffing hole.

As a rule, the size of the accumulator is such that it can hold the entire refrigerant charge.

4.9 Hoses

For easier mounting, compensation for manufacturing tolerances and for noise absorbing purposes, hoses are often used as flexible connections in mobile refrigeration applications.

Pressure pulsations are dampened through the volume elasticity of hoses. In addition, vibrations generated by the vehicle engine and the compressor and dampened and compensated for.

Hoses for refrigeration applications in vehicles basically consist of the following components:

- Inner layer (core)
- Pressure holding reinforcement (braiding)
- Outer layer (covering)

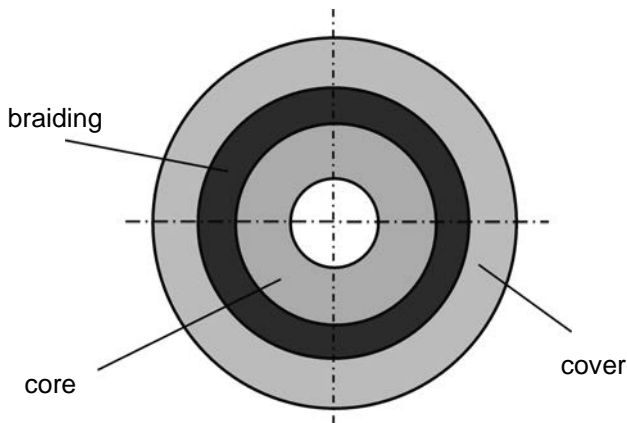


Fig. 4-44 Tube construction

The individual layer must fulfill the following functions:

Inner layer

Barrier function and resistance against R 134a and PAG or ester oil.

Pressure holding braiding

Resistance against inside pressure, provides dimensional stability of hose, absorption of external forces (e.g. elongation, compression).

Outer layer

Protective function against media influence from the outside, resistance against abrasion

Inner and outer layers of vehicle refrigeration hoses must be resistant to various media (liquids and gases). The material must not change its structure through chemical reactions. Also, the elastomer must not become brittle, shrink or swell.

The pressure holding reinforcement (braiding) is mostly braided onto the inner layer of the hose.

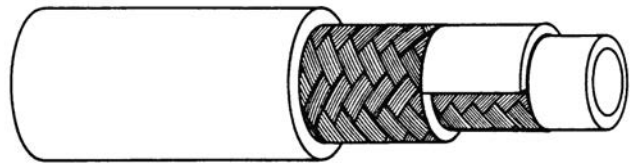


Fig. 4-45 Tube with braided pressure holding reinforcement (2-layer)

A rubber and/or waterproof hose is not a closed system. It always has a given gas permeability that depends on the materials used and the construction of the hose. In the hose assembly the core material basically assumes the barrier function. Refrigerant and ester and/or PAG oil are highly hygroscopic. If inadequate hoses are used, water can be absorbed from the surroundings through the hose material, which can lead to problems in the system in the long term.

For a quick and reliable installation of refrigerant hoses it is required that the torsional angle and the maximum bending of the hose be taken into consideration. When installing the hoses ensure that there is no strain on them.

Hoses that were used with refrigerant R 12 are not suited for use with R 134a. The R 134a molecules are smaller and are therefore more likely to penetrate through the pores of the hose material.

4.10 Pipework

The refrigeration capacity of the system is influenced by pressure drops in the pipework. Such pressure loss not only results in a decrease in cooling capacity but also in an increase in the compressor's power consumption.

Sizing of the piping system is determined by the following factors:

- pressure drops
- flow velocity
- oil return

Pressure drops in the lines cause a reduction in system capacity. Pressure losses are always stated in Kelvin (K) depending on the saturation temperature drop.

Table 4-1. Refrigeration capacity loss owing to pressure drops

Pressure loss in K	Refrigeration capacity in %
2 K in the suction line	92.2 %
2 K in the hot gas line	98.8 %

4 Components of the Refrigeration Cycle

On account of the lower cooling capacity, the pressure drop in the suction vapor and hot gas lines should be kept to a minimum. In mobile refrigeration applications a pressure drop of 4 to 7 K is normal, depending on the operating state. In the liquid line maximum 0.5 K are permissible since pressure drops may cause the formation of vapor bubbles in the liquid thus rendering reliable superheat regulation impossible.

With respect to flow velocities, the empirical values shown in Table 2 shall apply. Flow velocity is limited in upward direction through the pressure differences and noise problems that will be experienced.

Since the oil is carried along in the suction vapor and hot gas lines by the flow velocity, it is imperative that a minimum flow velocity be maintained. This is especially important at part-load operation in the case of capacity-regulated compressors. Insufficient oil return results in compressor failure. The flow velocities recommended in Table 2 are empirical values at which correct oil return is ensured.

Some refrigeration systems in cars operate in the regulated state without superheat. Here, the oil and refrigerant mix completely in the liquid line and the oil is thus swept along also at lower velocities.

Table 4-2. Recommended flow velocities

Line	Recommended velocity with CFC refrigerants
Suction line	5 - 15 m/s
Hot gas line	5 - 20 m/s
Liquid line	0.3 - 1.2 m/s

Correct routing of the lines is crucial for the proper functioning of the refrigeration system. All refrigerant lines should always be as short and should be run as directly as possible.

To facilitate oil return, all refrigerant lines should be pitched downward in the direction of refrigerant flow.

The suction line should be routed in a downward pitch in the direction of the compressor (Fig. 4-46). A siphon or a liquid separator fitted ahead of the compressor protects the compressor against refrigerant liquid that could migrate from the evaporator into the suction line when the system is shut down. Systems susceptible to refrigerant migration (e.g. railroad air-conditioning system with condenser and receiver mounted on the roof, and compressor at the bottom of the vehicle) should be equipped with a pump-down circuit.

It is recommended that the suction line is pitched downward away from the compressor (Fig. 4-47).

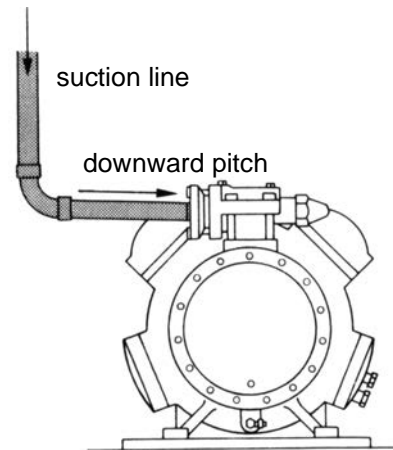


Fig. 4-46 Suction line routing at compressor

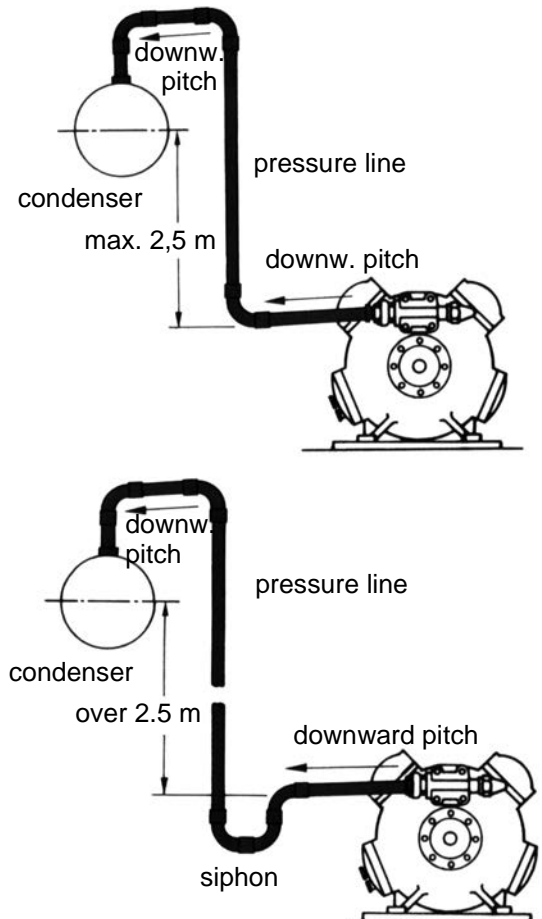


Fig. 4-47 Pressure line routing in case of condenser at higher level

In the case of condensers mounted at a higher level a section of pipe pitched in downward direction should always be provided downstream of the pressure stub. If the condenser is located more than 2.5 m above the level of the compressor, a siphon or a check valve must be installed

4 Components of the Refrigeration Cycle

in the pressure line. The siphon aids in the oil return along the higher-level riser. After the system has been operated for a given time, oil will collect in the siphon, the flow cross-section will then be restricted and the oil is swept along. During shutdown periods no oil or liquid refrigerant flows into the cylinder head which could result in problems upon start-up.

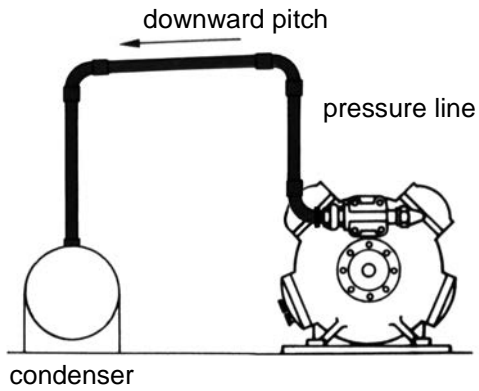


Fig. 4-48. Pressure line routing, condenser at same level or lower

If the condenser is located at a lower or the same level as the compressor, the pressure line can be routed in an ascending pitch directly behind the compressor port.

4 Components of the Refrigeration Cycle

5 Refrigerants, Lubricants and Heat Transfer Media

5.1 Introduction

In the DIN 8960 standard a refrigerant is defined as a heat transfer medium that - in a refrigeration machine process - takes up heat (evaporator) at a low temperature and a low pressure and rejects it at a higher temperature and a higher pressure (condenser).

Previously, in mobile refrigeration applications the only refrigerant used was HCFC R 12. In 1974, one started to suspect that CFC compounds destroy the protective ozone layer around the earth. The ban on CFC and Halon production therefore prohibits the use of CFCs in new plants. Substitutes for the refrigerants concerned had to be found. In mobile refrigeration systems the refrigerant R 134a is now being used for new installations.

5.2 Requirements of an ideal refrigerant

5.2.1 Physical properties

5.2.1.1 Evaporating pressure

The evaporating pressure should be at least 1 bar (absolute) over the entire operating range. This prevents air and water from entering the system in the case of small leaks or in systems with open-type compressors.

5.2.1.2 Condensing pressure

The condensing pressure should be as low as possible in order to keep construction complexity and energy expenditures within reasonable limits. The operating pressures that are acceptable in a system depend on the refrigerant used and the type of condenser. Details are described in DIN 8975 Part 1.

5.2.1.3 Pressure difference

Compressor motor sizing is governed by the pressure difference $p_c - p_o$. It should be as small as possible.

5.2.1.4 Pressure ratio

A pressure ratio as small as possible should be strived for. With an increase in the pressure ratio p_c/p_o , the volumetric efficiency λ of the compressor decreases resulting in a lower compressor capacity. It is therefore advantageous to use a refrigerant with a flat vapor pressure curve.

5.2.1.5 Final compression temperature

Considering the thermal stability of the lubricant used, the final compression temperature should be as low as possible. This temperature is dependent on the refrigerant, the

suction superheat and the condensing pressure of the system, as well as on the compressor.

The critical temperature measured at the outside wall of the pipe lies between 120 and 140 °C. A decisive factor in this respect is the temperature at the valve plates of the compressor, which is approx. 160 °C. At higher temperatures the oil will start to coke.

5.2.1.6 Solubility in water

Water in a refrigeration system will cause problems. The higher the water solubility of a refrigerant is the more moisture it can bind without any malfunctions of the system occurring.

In particular with respect to the high water absorbance of ester and PAG oils, moisture in the system plays a crucial role. Refrigerants are delivered with a residual moisture of max. 20 ppm.

5.2.1.7 Evaporation enthalpy and suction density

To be able to make a statement on the refrigeration capacity that can be attained with a given refrigerant, these two variables must be considered together. If a refrigerant has a high evaporation enthalpy, the same refrigeration capacity can be achieved with a lower compressor displacement. For the compressor to be able to deliver as much refrigerant per stroke as possible, the refrigerant at the intake fitting of the compressor should have a density as high as possible.

5.2.1.8 Miscibility/solubility of lubricants

For proper oil return in refrigeration systems it is desirable that the liquid refrigerant is completely miscible with the lubricant. Complete insolubility, as is the case with ammonia, can also be coped with in a satisfactory manner by fitting suitable oil return features.

Some refrigerants, however, are not miscible with refrigeration oil at every temperature and at every mixing ratio. This "**miscibility gap**" is of crucial importance for the oil return. If the oil/refrigerant ratio lies within the miscibility gap, malfunctions in the operation of the system may occur owing to problems with the oil return. The curve of the miscibility gap depends on the refrigerant type and the oil type.

The figure below shows an example of a miscibility gap diagram as it can be obtained from oil manufacturers.

5 Refrigerants, Lubricants and Heat Transfer Media

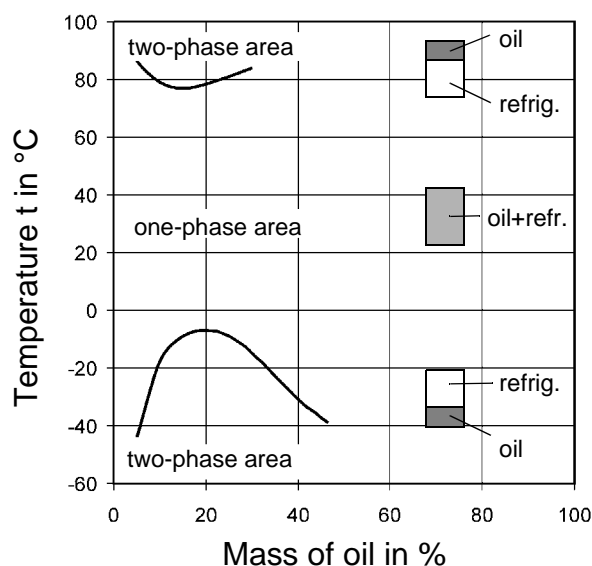


Fig. 5-1. R 134a-oil-solubility characteristic

The two-phase areas represent miscibility gaps of this specific refrigerant-oil pair. The oil-rich and refrigerant-rich phases are separate. In these areas oil return is only possible in the case of adequate flow velocities. The one-phase area represents good miscibility of refrigerant and oil. Oil return is effected in that the oil is “dissolved” in the refrigerant.

5.2.2 Chemical properties

Under all perceivable operating conditions, the refrigerant should not chemically react with the materials and lubricants used. The refrigerants themselves are only mildly aggressive towards the materials used. The refrigerant-oil mixture must be taken into consideration in this case.

5.2.3 Physiological properties

High physiological compatibility (non-toxicity) is desirable. For R 134a the MAK value (threshold limit value) is 1 000 ppm. This means that no detrimental effect on anybody’s health is to be expected if less refrigerant is inhaled during an 8-hour period. With higher concentrations there is a danger of asphyxiation owing to a reduction in oxygen especially near the bottom (R 134a is heavier than air). Moreover, headache, nausea and unconsciousness may be experienced.

Poisonous decomposition products will form under the influence of open flames, hot or incandescent metal surfaces or ultraviolet light (arcing).

5.2.4 Environmental compatibility

The use, production and disposal of the refrigerant should be without any negative impact on our environment. The refrigerants and plants are assigned specific indices by

means of which environmental compatibility can be assessed with respect to one another.

5.2.4.1 Ozone Depletion Potential (ODP)

The natural concentration of stratospheric ozone has been depleted over the last few decades through harmful influences and the protective shield against UV radiation has been reduced. This is caused by halogenes such as chlorine, fluorine and bromine which are released from ascending chlorofluorocarbons (CFCs) that are broken down by UV radiation.

At the international conference in Montreal in 1987, the “Montreal Protocol” was signed in which the signatory states agreed to phase out production of ozone-depleting substances by the end of 1995. Since 1995 no more CFCs are allowed to be produced in the signatory states. Unfortunately, not all states have ratified the Montreal Protocol as of yet.

As it takes approx. 15 - 20 years for the long-lived CFCs to reach the ozone layer, depletion of the ozone layer will continue over the next few years.

The most severe ozone depletion (over 50 %) takes place over the pole regions. Over the Antarctic one can therefore watch the so-called ozone hole in the period between September to November, the Antarctic spring. In the northern hemisphere an increased depletion occurs in the winter and spring months. Between 1968 and 1992 the ozone reduction over Europe amounted, on average, to three percent per decade. In the last few years an ozone reduction of five percent per decade could be observed. Higher UV levels will cause increased skin cancers and cataracts.

Refrigerants with the highest **ozone depletion potential**, i.e. R 11 and R 12, were assigned the value ODP 1.0 (100 %). The other chlorine-based refrigerants are assigned their ODP values in accordance with their depletion potential in comparison to R 11.

5.2.4.2 Global Warming Potential (GWP)

Life on earth depends on the radiated energy emitted by the sun. However, the largest part of it is either reflected back or given off by evaporation. The natural green house gases, i.e. water vapor and CO₂, prevent the complete reflection of the thermal energy and retain heat in the earth’s atmosphere. This effect can be compared with that of a glass surface of a green house. Without the natural green house gases the average temperature of the earth’s surface would not be 18 °C but -15 °C.

This green house effect, although beneficial in itself, increases continuously through green house gases (CO₂, refrigerants, methane from the agriculture) that are emissions from human sources. Since the beginning of industrialization in the Middle Ages the proportion of CO₂ in the

atmosphere has increased steadily.

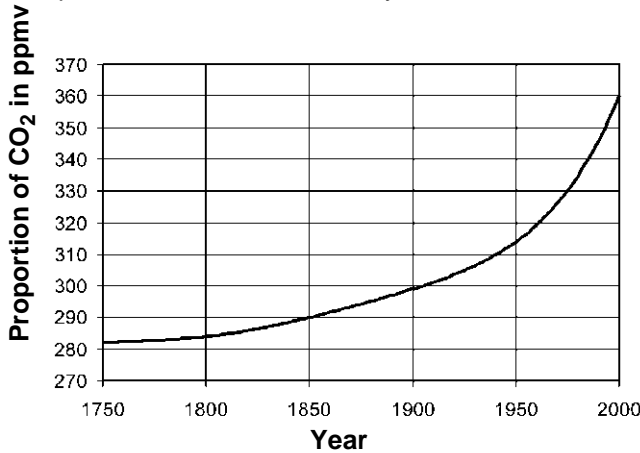


Fig. 5-2. Increase of CO₂ concentration

This increased green house effect has resulted in a mean annual temperature rise of 1 – 1.5 K. This temperature increase will in the long term cause a rise of the sea level, variations and shifts in the distribution of precipitation and an increase in extreme weather phenomena.

The **global warming potential** of the refrigerants is identified by the GWP (direct global warming effect based on CO₂) or the H-GWP (direct global warming effect based on R 11).

- R 12 GWP 7 100
- R 134a GWP 1 200

The ODP and GWP values are not absolute, measurable properties. They are determined by means of model computations of the reactions taking place in the atmosphere. This leads to numerically differing results. Both values take only into account the properties of a refrigerant, not the production process or the energy requirement of the refrigeration system.

5.2.4.3 TEWI

The TEWI value (total equivalent warming impact, direct and indirect global warming effect) not only refers to the impacts caused by the refrigerant itself, but also to the system in which the refrigerant is used. The global warming impact caused by the energy requirement of the plant, the release of refrigerants during disposal, and leaks are also taken into account. A distinction is made between the **direct greenhouse effect** caused by the refrigerant (leaks, repair losses, scrapping losses) and the **indirect greenhouse effect** (CO₂ emissions during energy generation).

A weak point of the TEWI evaluation is that the production of the individual refrigerant is neglected.

5.3 Prohibition on the production of HCFC-Halon

The ban on the production of CFCs and Halon of 6 May 1991 provides regulations for the phase-out and use of CFC and the handling of these refrigerants.

Contents

- Since 1995 no more HCFC (R 12) refrigerants are allowed to be used in new installations.
- It is prohibited to release refrigerants - contrary to the state of the art - into the atmosphere during the operation, repair and decommissioning of refrigeration systems.
- Any work on refrigeration systems may only be performed by persons having the required competence and technical equipment.
- The use of refrigerants must be documented.
- Older plants must be converted to refrigerants with as little ozone depletion potential as possible (replacement refrigerants).

The Federal Environmental Agency must announce suitable alternative refrigerants, which in the case of R 12 was done at the end of 1995. As a consequence, no more R 12 is allowed to be filled into older systems. As soon as the system is opened within the context of a service call, the system must be converted to a suitable alternative or service refrigerant. Existing plants may continue to be operated for as long as they do not develop any leaks. R 134a was named as an alternative refrigerant for R 12.

Within the European Union, "EU Regulation 2037/2000 on Ozone Depleting Substances" contains controls on the production, use and availability of CFC and HCFC.

5.4 Refrigerant R 134a compared with R 12

For R 12 a single component replacement substance R 134a has been found with suitable properties and which is exclusively used in today's automotive air-conditioning systems.

Comparison R 134a/R 12

R 134a is more environmentally friendly than R 12 since, firstly, it does not attack the ozone layer (it contains no chlorine), and secondly has a lower global warming potential.

The vapor pressure curves of R 134a and R 12 are very similar, as can be seen in Table 5-1. At lower saturation temperatures R 12 exhibits somewhat higher pressures,

5 Refrigerants, Lubricants and Heat Transfer Media

at higher saturation temperatures R 134a. Owing to the altogether similar pressure behaviour nearly identical components can be used. The remaining thermodynamic values are also very similar. The values for cooling capacity, energy requirement and temperature behaviour are comparable.

Only as far as the lubricant and the sealing materials were concerned did significant changes have to be made upon the conversion to R 134a. R 134a is a smaller molecule and therefore requires seals with smaller pores. Sealing materials or hoses used with R 12 must therefore not be used in conjunction with R 134a.

Mineral and synthetic oils that were commonly used with R 12 are not miscible (soluble) with R 134a and are only insufficiently transported in the refrigeration cycle. Non-miscible oils may deposit in the heat exchangers and impede heat transfer so significantly that no proper operation of the system is possible. Therefore, ester (bus) and PAG oils (cars) are used together with R 134a.

5.5 Refrigerant conversion

Since no more service is allowed to be carried out on existing R 12 plants, the latter must be placed out of service or converted to a different refrigerant in the case of repairs or leaks. It is not allowed to add R 134a to R 12 because the refrigerant properties of the mixture significantly differ from those of the pure substances. Two options are available for the continued operation of the plant.

5.5.1 Retrofit

Within the context of a **retrofit measure** not only a **conversion of the refrigerant** is necessary, but also a **conversion of the oil**. In accordance with the state of the art, only refrigerant **R 134a** can be considered for use in automotive refrigeration/air-conditioning applications. R 134a requires a "polar" lubricant (PAG or ester) to ensure reliable oil return with a view to the miscibility refrigerant/oil.

Any retrofit measure is only "justified" if the refrigerant hoses are replaced with hoses that comply with the latest state of the art technology. With regard to the compressor a check must be performed as to whether the sealing materials used are appropriate for the new refrigerant/lubricant combination. The filter drier needs also to be replaced in any case. The expansion valve must be adapted to ensure proper superheat regulation.

Residual oil content containing chlorine have the following negative effects:

- circulates in the refrigeration cycle in the form of insoluble oil
- clogging caused by accumulation in the receiver-drier
- flow rate variations in the area of the expansion device
- high water contents have a catalytic effect in con-

junction with this residual oil content and result in a chemical breakdown of the lubricant or refrigerant. The use of copper results in a copper plating effect, the use of other materials results in corrosion.

- shift of the miscibility gap - depending on the residual oil content - even in temperature ranges as they occur in automotive refrigeration/air-conditioning applications. This may result in oil return problems.

Due to the fact that in automotive refrigeration systems an increased amount of water in the system must always be expected, demands for a residual oil content as low as possible are inevitable. This can only be achieved by flushing. Suitable flushing devices are commercially available.

5.5.2 Drop-in

In the case of "**drop in**" refrigerants the conversion is carried out **without oil change**. In this case, **refrigerant blends** (service refrigerants, transitional refrigerants) are required containing an oil-soluble component. These blends have, as a rule, no constant evaporating or condensing temperatures ("temperature glide").

Effects of the "temperature glide":

- Increase or decrease in concentration of a given component of the blend in individual components of the refrigeration cycle
- Concentration shift of the blend components in the case of leaks
- Possible problems with the superheat regulation
- Refrigerant blends may only be charged into the system in the liquid state

A problem with many service refrigerants is a residual ODP caused by the R 22 content required for the oil return.

An advantage of "drop in" refrigerants is that hardly any changes need to be performed on the system and that the oil is allowed to remain in the refrigeration cycle.

5.6 Refrigeration oils

5.6.1 Function of refrigeration oils

The main function of refrigeration oils is the lubrication of the compressor and heat rejection. Beyond that, the oil seals the compression chamber and the valves. The oil is only required in the compressor. However, it also reaches the other components within the system through the oil throw of the compressor. It is assumed that in typical bus refrigeration systems 5 - 10 % oil is in circulation. In motor vehicles up to 80 % is circulating.

A small percentage of oil in the system may even be beneficial. The system becomes more leak-proof and the heat

5 Refrigerants, Lubricants and Heat Transfer Media

exchanger performance somewhat higher. Also, leaks can be easier localized owing to the oil that emerges in small amounts. An oil film is formed at the location of the leak.

5.6.2 Problems caused by oil

Frequent oil-related problems are:

- Moisture in the system through the use of ester and PAG
- Oil return
- Refrigerant migration to the crankcase during shutdown periods
- Viscosity differences at different temperatures in the refrigeration cycle

In addition to moisture, the oil return is a crucial factor for the operational reliability of the plant. Where the refrigerant is in a gaseous state the oil must be swept along by

the velocity of the refrigerant. Especially in the part-load range it is imperative that minimum velocities be maintained.

During shutdown periods of the compressor the pressure in the crankcase is higher than during operation. This causes more refrigerant to be absorbed by the oil. Upon start-up the refrigerant/oil mixture starts foaming preventing proper oil lubrication. This effect can be eliminated by installing a crankcase heater or a pump-down circuit.

The oils used must still be sufficiently viscous at the high temperatures prevailing in the compressor and at the same time still liquid at the temperatures in the evaporator. Here, problems can be encountered particularly at low temperatures.

The viscosity is dependent on the temperature and the amount of refrigerant dissolved in the oil. The higher the temperature the lower the viscosity of the oil. At high pres-






Oil-hydraulic power control	Static seals	Working valves	Shaft seal	Piston/cylinder	Drive bearing	
						Hydrodynamic power transmission
			●	●	●	Heat rejection
		●	●	●	●	Reduced friction (drive energy)
		●	●	●	●	Reduced wear
	●	●	●	●		Increased tightness
		●	●	●	●	Noise reduction
			●	●	●	Transport of particles causing wear
●						Hydrostatic power transmission

Fig. 5-3. Functions of the lubricant

5 Refrigerants, Lubricants and Heat Transfer Media

sure a large amount of refrigerant is dissolved in the oil resulting in a decrease in viscosity.

5.6.3 Types of typical refrigeration oils

5.6.3.1 Mineral Oils

Mineral oils can be used with refrigerants containing chlorine, and with hydrocarbons.

5.6.3.2 Alkylbenzene

Alkylbenzene oils are synthetic base oils. They are thermally more stable than mineral oils and their miscibility with refrigerants is better. Moreover, foaming is less pronounced when the plant is started up. When converting a plant from R 12 to some transitional refrigerants, alkylbenzene should be added.

5.6.3.3 Polyalkylene glycols (PAG)

PAGs are synthetic lubricants used in automotive air-conditioning applications with R 134a. These oils have the following properties:

- highly hygroscopic (water-absorbing)
- polyalkylene glycols are supplied with a residual moisture of max. 300 ppm. This value should not exceed approx. 700 ppm during operation
- stable to reactions with water
- good viscosity temperature behaviour
- high thermal stability
- high wear protection
- sensitive to chlorine (R 12)
- low mineral oil compatibility
- good refrigerant miscibility, possible miscibility gap at high temperatures

PAGs are not suitable for systems with copper components (copper plating effect) due to the high water absorption capacity.

5.6.3.4 Ester oils

Polyol ester oils (POE) are synthetic lubricants that are used, among others, in bus air-conditioning applications. These oils have similar properties as PAGs; they are, however, by far not as resistant to decomposition at high water contents. Ester oils are supplied with a residual moisture of 50 ppm and less.

Recent experience has shown that the water content in systems with ester oil should not exceed 100 ppm. Otherwise the danger exists that metallic materials corrode, an accumulation of mud or the formation of acids and alcohols occurs. Once moisture has been adsorbed by the oil it cannot be removed by evacuation. Also drying with nitrogen or by means of the refrigerant filter-drier is not

satisfactory. Oil containers cannot be used multiple times. Once a container has been opened, the entire oil should be used immediately or disposed of.

5.7 Heat transfer medium

If refrigeration systems are operated with indirect evaporation another medium for heat transport is required in addition to the refrigerant. Apart from water, refrigerants or special heat transfer liquids mixtures of water and anti-freeze agents (brine) are frequently used.

The brine must protect the cycle against freezing, corrosion and deposits. In most cases the antifreeze agent glycol is used resulting in a low melting point and the formation of slush upon congealing (burst protection).

6 Plant Design

6.1 Safety components

6.1.1 General

Art. 6 of VBG 20 (regulations of the trade association) demands under the keyword "Components under compressive stress" that components under compression stress in a refrigeration system must be able to withstand the pressures occurring during system operation or when the plant is at rest. Art. 7 deals with safety devices providing protection against excessive pressures. Said Art. 7 contains 5 sections which broadly express the following:

Safety devices designed to control excessive pressures must meet the following requirements:

- They must be adjusted in such a way that the maximum allowable operating pressure is not exceeded by more than 10%. The level of the maximum allowable operating pressure is dependent on the refrigerant used, the operating mode and the operating principle of the condenser, or the high-pressure side. The maximum allowable operating pressures (minimum requirements) are detailed in DIN 8975 T1 or are to be determined based on the 'weakest' component of the high-pressure side.
- They must be protected against unauthorized tampering of settings.
- They are not required in plants containing an R 134a refrigerant charge of less than 2.5 kg. In this case, however, the refrigeration plant must be intrinsically safe (designed so that no pressure in excess of the maximum allowable operating pressure can occur). The definition of the term "intrinsically safe" is contained in DIN 8975 T7. A refrigeration system is deemed to be intrinsically safe if, with the plant at rest, the maximum allowable operating pressure is higher than the saturation pressure at a temperature of 63 °C.

Under operating conditions a refrigeration system is deemed to be intrinsically safe if the maximum allowable operating pressure is higher than the pressure generated at an ambient temperature of 32 °C. This upper operating limit can be higher for given device types; for more details refer to the relevant test specifications (DIN standards). A refrigeration plant is deemed to be intrinsically safe if, for example, despite the fact that the compressor is operated continuously the maximum allowable operating pressure is not exceeded thanks to a large dead space or an internal overflow valve.

- The devices installed must not be able to be shut off.

6.1.2 Types of safety devices

Safety devices providing protection against excessively high or low pressures may be:

- Safety valves in the form of drain or overflow valves
- Rupture discs
- Pressure relief joints
- Thermostats
- Pressure cut-outs (pressostats)

Pressure cut-outs (pressostats) are very frequently used as safety devices as protection against over- or under-pressure. They act directly on the magnetic clutch of the compressor.

If the system pressure rises excessively, for example, on account of a highly contaminated condenser, a failed fan or a defective expansion valve, the **high-pressure switch** will interrupt the power supply to the magnetic clutch when the set pressure is reached (manufacturer-specific, approx. 26 bar). The compressor is restarted as soon as the system pressure has dropped by the preset differential pressure (approx. 5 - 6 bar) below the cut-out pressure (20 - 21 bar).

The **low-pressure switch** will interrupt the power supply to the magnetic clutch when the preset pressure is fallen short of (possible causes: refrigerant shortage, defective expansion valve, insufficient heat load, defective evaporator fan, ...). Since the refrigerant R 134a has an evaporating temperature of 0 °C at a gauge pressure of 2 bar, the low pressure switch is frequently set to a value of approx. 2 bar as protection against icing. In most cases the cut-in pressure lies 0.6 bar above the cut-out pressure. Therefore, the compressor will start up again at a pressure of 2.6 bar if the cut-out pressure is 2 bar. In uncontrolled compressors, this principle is also used for capacity control purposes.

In many refrigeration plants another **pressure switch** is installed on the high-pressure side for the connection of an additional condenser fan. The additional fan lowers the condensing pressure under difficult operating conditions (slow driving speed, vehicle at rest) thus increasing the refrigeration capacity of the plant.

A **freeze protection thermostat** is frequently installed to protect the evaporator against the formation of ice. Just like the pressure controllers, it is designed to activate and deactivate the magnetic clutch of the compressor. Depending on the design, the sensor of the temperature controller is placed between the evaporator fins or fitted to the evaporator outlet (suction side). In the case of temperature controllers with fixed settings, the controller switches off the compressor at approx. -1 °C to 0 °C and back on again at approx. +3 °C. In the case of adjustable temperature controllers, the switching point can be modified so that the compressor will already switch off at higher temperatures enabling regulation of the air temperature.

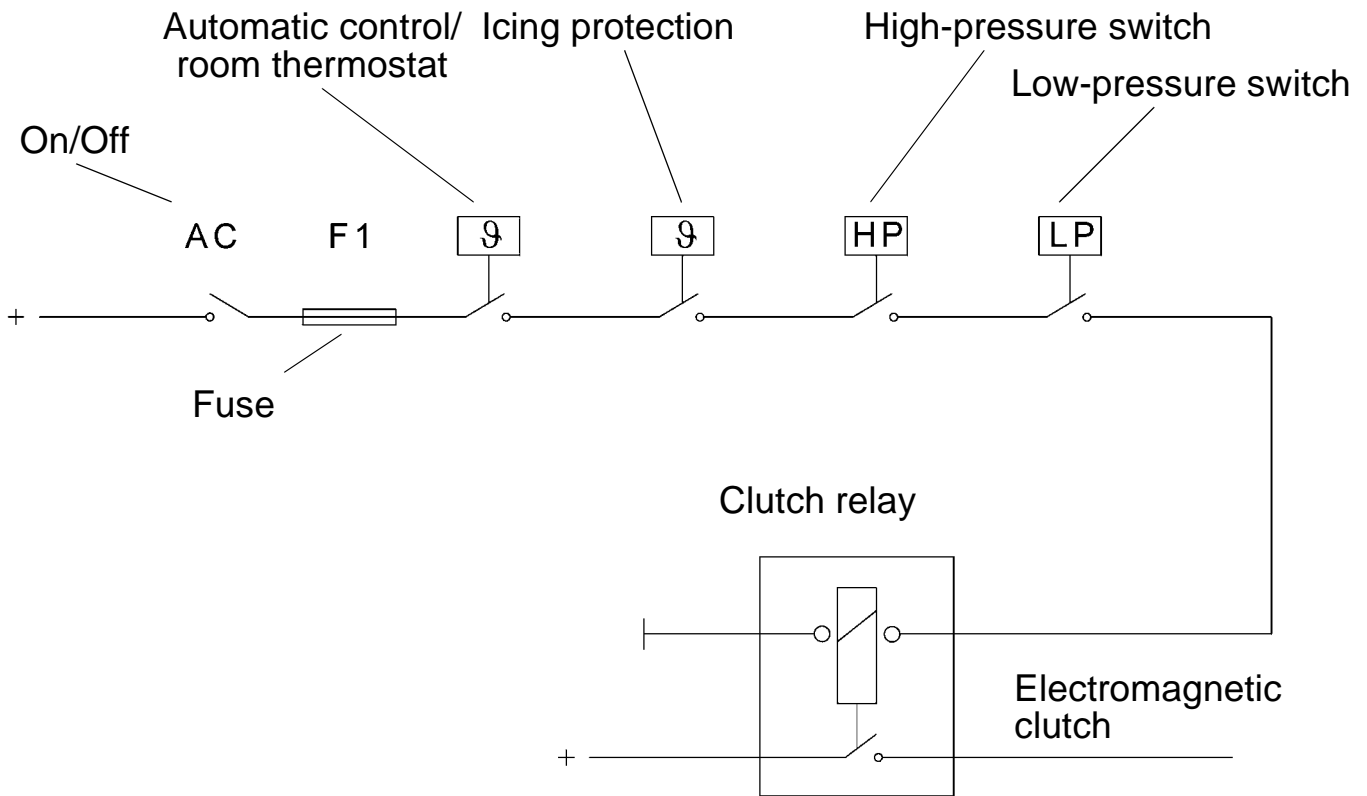


Fig. 6-1. Safety chain made up of a number safety components

Fig. 6-1 shows a possible electric diagram for the connection of these safety components.

In compressors installed in buses an internal **overflow valve** is frequently fitted to protect the compressor against excessively high pressures. At high pressures this valve causes refrigerant to flow from the high-pressure side to the low-pressure side of the compressor thus preventing further pressure build-up.

In the case of extreme overcharging or extremely high ambient temperatures very high pressures within the system occur when the plant is at rest. To prevent components from bursting, a **drain valve** is frequently installed on the high-pressure side that opens from a pressure of, for example, 40 bar allowing refrigerant to escape to the atmosphere.

6.2 Capacity control

6.2.1 Uncontrolled compressors

6.2.1.1 Control by means of magnetic clutch

Up to the mid-eighties uncontrolled compressors were installed in cars worldwide. Excess refrigeration capacity - e.g. at higher engine speeds - was "down-regulated" in that the compressor was deactivated by the magnetic

clutch. The latter, in turn, received the signal from a temperature sensor at the evaporator to prevent ice formation on the evaporator fins.

The most significant disadvantage for the driver is the jolt experienced upon switch-on, which is especially noticeable with less powerful engines. Moreover, temperature fluctuations of the air introduced into the passenger compartment occur.

Fig. 6-2 shows the air outlet temperatures at the air discharge vents on the dashboard as well as the compressor's power consumption in a passenger car air-conditioning system with uncontrolled compressor in comparison with a system with controlled compressor.

In the former case fluctuations of the discharge temperature ranging between approx. 6 and 8 °C are experienced as well as a compressor power consumption between (initially) 4,2 kW and (just before switch-off) approx. 2.7 kW. The curve of both parameters is nearly constant when using an infinitely variable compressor.

As shown by comparison measurements, the energy requirement of uncontrolled compressors is higher than that of regulated ones. This is also the case with stationary refrigeration systems as also there the unsteady switch-on/off actions are accompanied by energy losses.

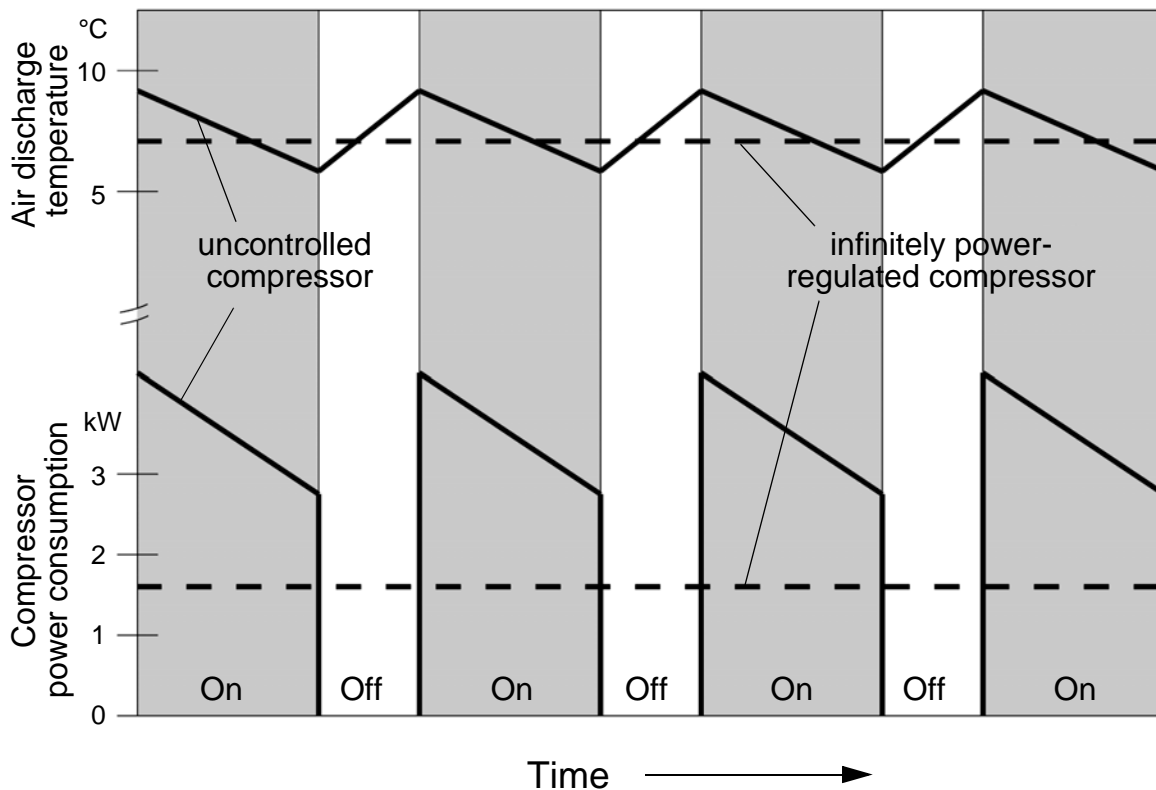


Fig. 6-2. Curves of air discharge temperature and compressor power consumption in the case of uncontrolled and infinitely variable compressors

The full refrigeration capacity of the compressor is required only infrequently, e.g. in the following cases

- immediately after starting once the interior has been heated up, to achieve a temperature reduction as quickly as possible
- in city traffic or during "stop and go operation"

The available refrigerating capacity is too high during most of the compressor operating period - the compressor is switched off. Measurements on passenger car refrigeration compressors in the mostly hot and humid city of Tokyo have shown that - over a one-year period - the compressors operate in 90 % of the time in the part-load mode and only 10 % of the time at full-load.

6.2.1.2 Capacity adjustment through "counter-heating"

As a rule, the possibility exists to adjust the air temperature as desired by the simultaneous operation of the air-conditioning system and the vehicle heating system.

This principle is used for dehumidification. In so doing, the air is initially cooled in the evaporator. As cold air cannot absorb as much water as warm air, part of the moisture contained in the air will condense on the evaporator tubes or fins. The air is subsequently heated in the heater's heat exchanger to the desired inlet temperature.

As the air-conditioning system operates continuously thus

requiring driving energy, this operating mode should only be used for dehumidification, not, however, for the capacity control of the refrigeration system.

6.2.1.3 Hot gas bypass

The hot gas bypass offers a stepless type of capacity control. Here, part of the compressed refrigerant from the compressor outlet is throttled on the suction side. Since no longer the entire refrigerant mass flow reaches the compressor, the refrigeration capacity decreases.

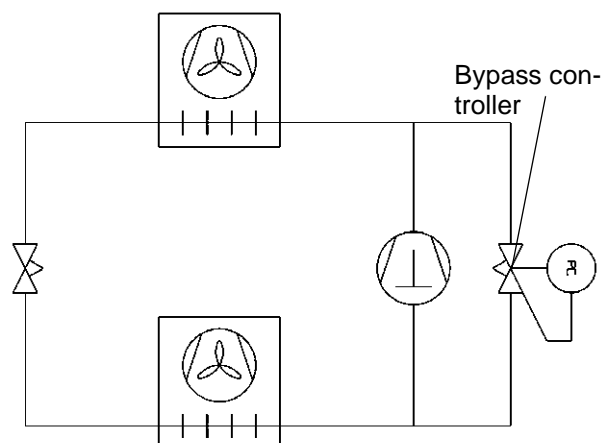


Fig. 6-3. Circuit with hot gas bypass

6 Plant Design

When the evaporating pressure drops to a preset value, the bypass controller starts to open thus causing the evaporating pressure to be limited in downward direction.

A problem with this type of capacity control is the greatly increased hot gas temperature (the throttled hot gas results in higher superheating) and the barely reduced compressor driving performance (the compressor must deliver the full refrigerant volume).

6.2.1.4 Suction throttling

As the refrigeration capacity greatly depends on the suction pressure, the refrigeration capacity can be regulated through influencing this pressure. The installation of a control valve between the evaporator and the compressor can reduce the suction pressure down to the operating limit of the compressor. Consequences are a lower volumetric efficiency and a lower density of the suction vapor. Both effects reduce the refrigerant mass flow delivered and thus the refrigeration capacity. An increase in hot gas temperature and a lower system efficiency must be taken into account.

6.2.1.5 Speed control

While in stationary refrigeration applications using electrically driven compressors, speed control via pole-changing electric motors (stepwise) or frequency converters (stepless) is effected, the speed of belt-driven vehicle compressors can only be regulated with the aid of a gearbox.

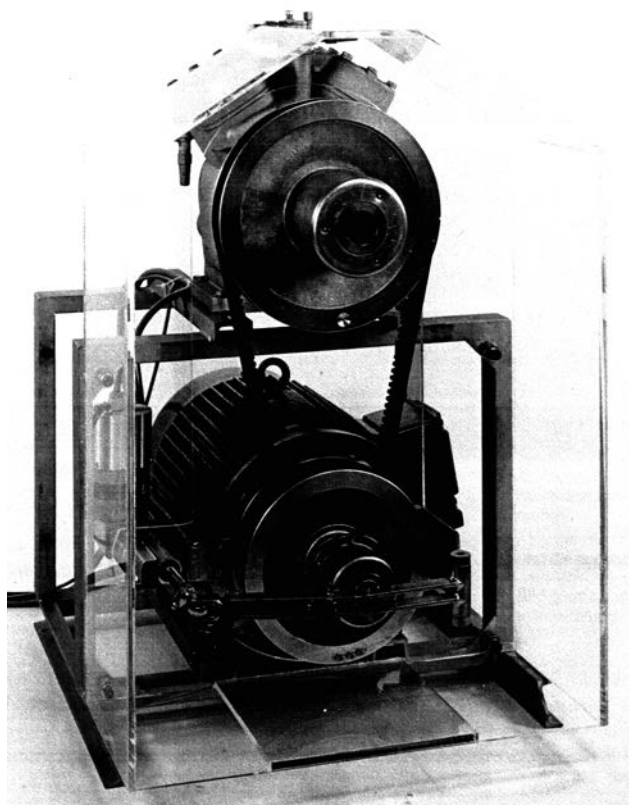


Fig. 6-4. Speed control by means of a variable gearbox

The gearbox shown consists of a mechanically adjustable control pulley with a central adjustment feature and a second, spring-loaded control pulley. Adjustment of the V-belt effective diameter is effected by an electrically driven lifting cylinder which acts on the central adjustment feature via a lever mechanism. The signals for a speed change which may originate from a temperature or pressure sensor act on the variable-speed motor. The advantage of the mechanical variable-speed gearing is its simple construction. A drawback, on the one hand, is the lower part-load efficiency of the controller system caused by the drive motor, and on the other hand the increased space requirement and the additional weight which prevent an increased application of this system.

6.2.2 Internally controlled compressor

6.2.2.1 Cylinder deactivation

Open compressors with a large number of cylinders offer the possibility to deactivate cylinders for a stepwise capacity control. The attainable refrigeration capacity in the part-load mode is dependent on the ratio of the number of operating cylinders to the number of deactivated cylinders. Often it is possible to select between cylinder pair and single cylinder deactivation. A combined deactivation of single cylinders and cylinder pairs enables a finely tuned capacity control. In the example of a cylinder pair deactivation shown below a performance reduction of 50 % is possible.

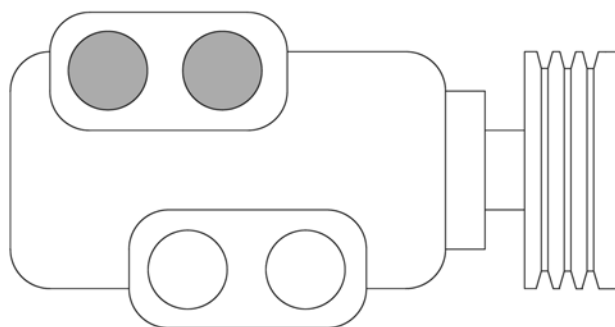


Fig. 6-5. Cylinder pair deactivation

The operating limits of an open compressor change only to a minor extent through cylinder deactivation.

Figures 6-6 and 6-7 show the design of a device for cylinder deactivation. To this end, the compressor is provided with a special cylinder head with a solenoid valve and a valve unit.

When the **solenoid valve is closed** no gas from the high-pressure side can flow into the space above the valve piston. The pressure spring pushes the piston into the upper seat, the suction channel is open and the compressor operates with all cylinders.

When the **solenoid valve is open** gas from the high-pressure side can flow into the space above the valve piston

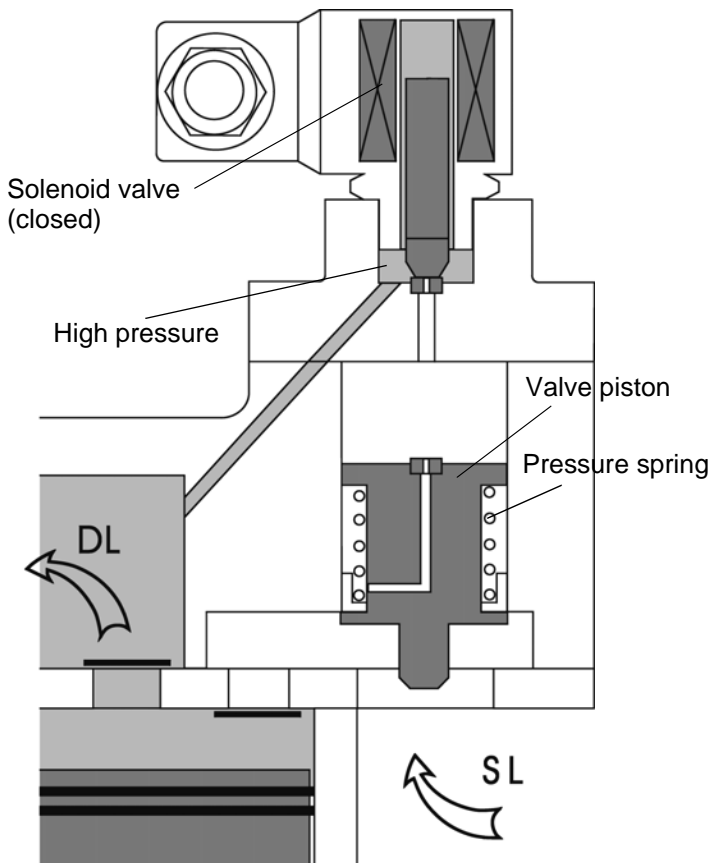


Fig. 6-6 Cylinder deactivation, full-load operation

pushing the same against the force of the spring into the lower seat. The suction channel is blocked preventing any new refrigerant from being drawn in.

If more refrigeration capacity is required, the solenoid valve closes, the refrigerant above the valve piston is sucked through the compensation line and the spring can push the valve piston back into the upper seat. The cylinder or the cylinder pair concerned are operating again.

Cylinder deactivation is very economical since apart from the friction losses of the piston no further losses occur. In addition, the solenoid valve can be activated and deactivated any number of times without any negative effect on the life of the compressor.

6.2.2.2 Swash plate adjustment (internally controlled)

Another method of capacity control is changing the piston capacity. This will be explained by taking an axial piston compressor with internal control valve as an example.

By changing the tilting angle of the swash plate different piston capacities can be steplessly selected (from approx. 2% to 100%). Fig. 6-8 shows a schematic view of a swash plate compressor with the swash plate in maximum delivery volume position (large tilting angle of swash plate). This position is required for full-load operation.

If the swash plate has a smaller tilting angle, the compres-

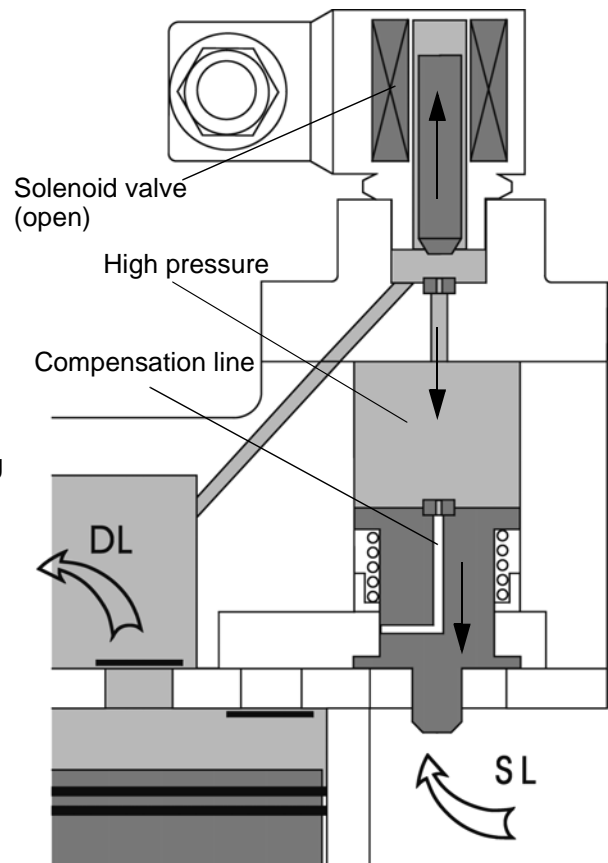


Fig. 6-7. Cylinder deactivation, controlled operation

sor operates at minimum piston capacity. This position is assumed by the swash plate when the compressor has been shut down or in the case of very low heat loads.

Upon the start-up of the system and at full-load operation, there is initially a high evaporating pressure. The lower the interior temperature becomes the lower the evaporating pressure gets. When the evaporating pressure has reached the pressure preset at the control valve (mostly 3 bar corresponding to 0 °C evaporating temperature), the tilting angle of the swash plate is reduced in such a way that the preset evaporating pressure remains constant. The tilting angle of the swash plate results from the equilibrium of the mass moments of inertia (piston acceleration) and the pressure difference between piston capacity and crankcase. An increasing pressure in the crankcase, for example, moves the swash plate in upward direction reducing the piston capacity.

The required position of the swash plate is controlled by a **control valve** (Fig. 6-9) installed in the compressor.

In the deactivated state the forces in the cylinder chambers and the compressor housing are the same. In this case, the spring pushed the swash plate into the upper position. The capacity is minimal.

The pressure in the cylinder chamber rises upon start-up. A resultant force comes into existence that is stronger than the forces acting from below pushing the swash plate

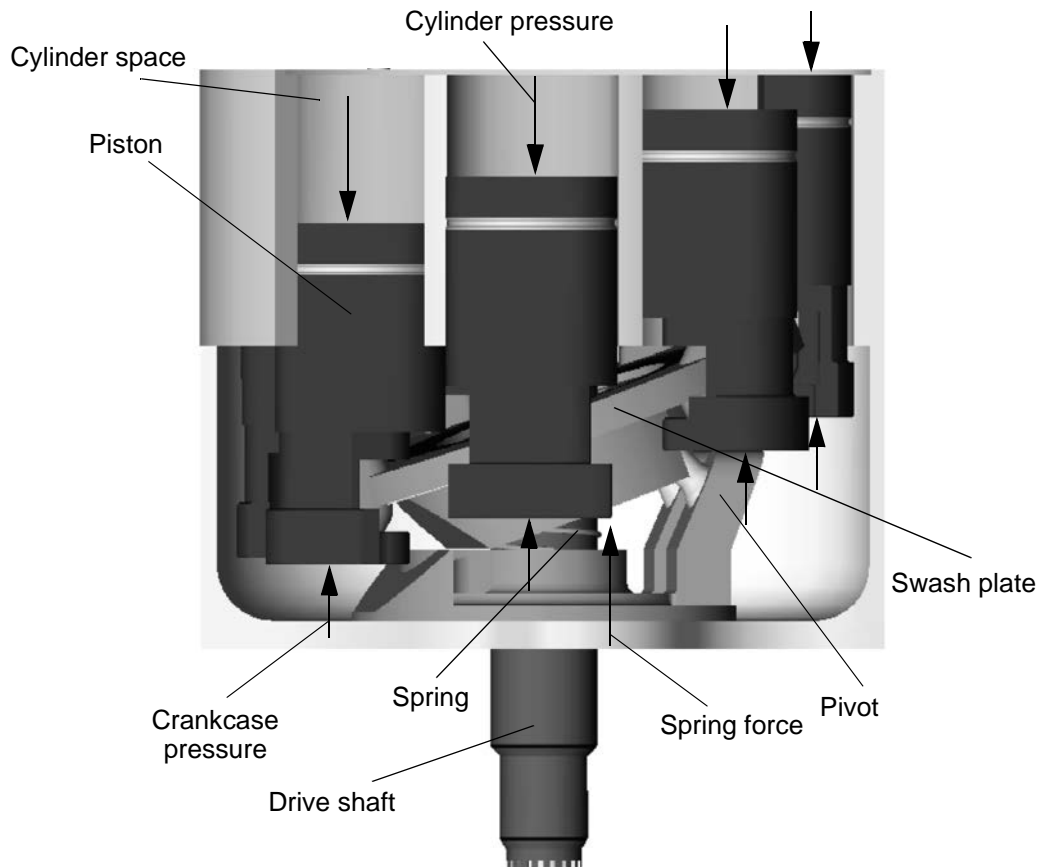


Fig. 6-8 Capacity control of an internally controlled swash plate compressor

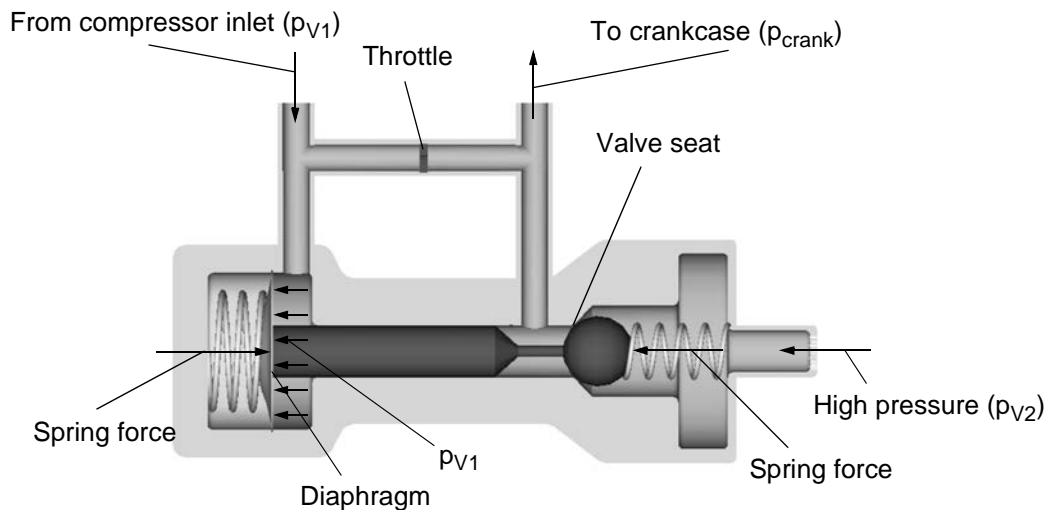


Fig. 6-9 Control valve of an internally controlled swash plate compressor

in downward direction (max. capacity). The compressor is working at full-load.

Since the suction pressure is initially higher than the spring force of the control spring, the control valve remains closed. The crankcase pressure drops as the refrigerant is drawn off via the throttle (required for damping).

Due to the increasing cooling of the interior, the evapora-

ting temperature drops and hence also the suction pressure at the compressor. To prevent the formation of ice, the control valve is generally set to a suction pressure of 3 bar. If the suction pressure drops to 3 bar, the control valve opens allowing the refrigerant to flow from the high-pressure side into the crankcase. The crankcase pressure rises and causes the resultant force acting on the bottom of side to increase; as a result, the swash plate is moved upward reducing the piston-swept volume of the compressor. The compressor capacity is thus reduced. The preset

suction pressure cannot be fallen short of as the valve sets the chamber pressure so that the desired suction pressure is maintained.

If a higher cooling capacity is required, the suction pressure rises above 3 bar closing the control valve. The crankcase pressure falls because refrigerant is sucked from the crankcase through the bypass via the throttle. The tilt angle of the swash plate is increased again.

6.2.2.3 Swash plate adjustment (externally controlled)

Modern passenger car air-conditioning systems feature an externally controlled compressor. The swash plate angle is controlled by an electronic valve controlled by the on-board computer.

6.2.2.4 Vane Compressor

The infinitely variable performance control of a vane compressor can be effected by changing the displacement. The rotation of a control disk causes the intake opening to be extended decreasing the compression area. A two-stage capacity control can be achieved by an internal bypass. If the compressor is to work at part-load, this bypass is opened so that part of the drawn in refrigerant is returned to the suction side.

6.2.2.5 Scroll Compressor

For the capacity control of a scroll compressor, it is possible to lift off the movable spiral. This results in an internal floodback condition thus reducing the flow rate (cooling capacity).

6.3 Refrigerant migration

When the refrigeration system is at rest, the refrigerant contained in the system always condenses at the point with the lowest temperature. If the compressor is the coldest point, the refrigerant contained in the evaporator condenses in the crankcase of the compressor. Upon start-up of the system liquid slugging (destruction of the valves or valve plate) and foaming of the oil (extensive oil slinging in the system, low lubricity of the oil) may occur. If a leak exists at the expansion valve, refrigerant can condense from the high-pressure side to the low-pressure side. The condenser is frequently exposed to solar radiation on the vehicle roof whereas the compressor is located in the cooler engine compartment, which promotes refrigerant migration. Measures against refrigerant migration into the suction line include the installation of a liquid separator in the suction line as well as correct line routing to the compressor (P-trap). Refrigerant migration into the compressor can only be eliminated by installing a crankcase heater in the compressor or by means of a **pump-down circuit** and/or a **check valve**.

In the case of a pump-down circuit a solenoid valve is

installed in the liquid line (Fig. 6-10).

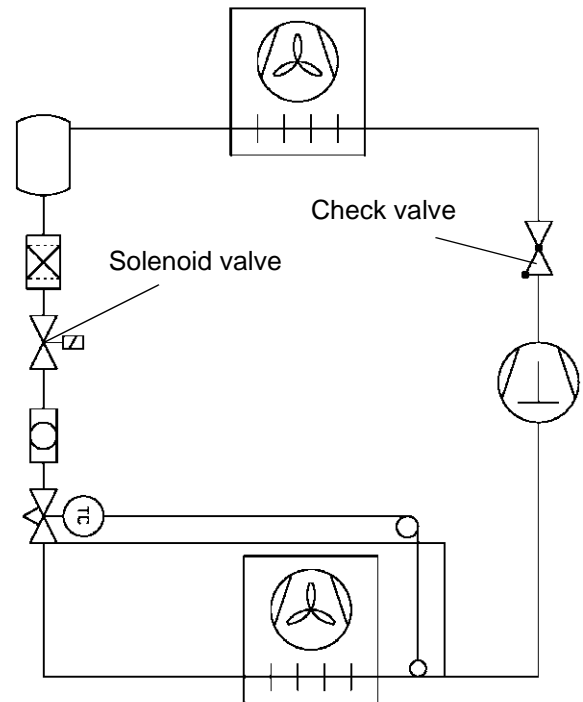


Fig. 6-10 Pump-down circuit

If the plant is to be switched off, the solenoid valve is closed before the compressor is switched off and the refrigerant is transported from the low-pressure to the high-pressure side (into the condenser and the receiver). Deactivation of the compressor is then effected either by a signal from the low-pressure cut-out or timer-controlled. No refrigerant migration to the compressor or into the suction line can occur since no more liquid refrigerant is present on the low-pressure side.

To prevent refrigerant migration on the high-pressure side into the cylinder head, a check valve may be installed downstream of the compressor.

7 Commissioning

The operational reliability and service life of a refrigeration system primarily depend on the degree of contamination, the moisture and non-condensables contained in the refrigeration cycle. Already at the start of the production of components for refrigeration systems, state-of-the-art quality requirements must be complied with. Only in this way can it be ensured that technically competent personnel in industrial manufacturing assemble components complying with specifications into a complete refrigeration plant and achieve reliable operating conditions of the plant over a long period of time through conscientious evacuation, charging and maintenance of the plant. The increasing hermetization of the refrigeration cycles enables a reduction of substances entering and escaping from the system during operation.

7.1 Moisture in the refrigeration cycle

7.1.1 General

Many years of practical experience have shown that reliable operation of refrigeration systems is ensured if the water content of the refrigerant contained in the system does not exceed 100 ppm (parts per million = mg/kg) (ester oil). As early as during the production phase of refrigerants can the presence of a residual water content in the refrigerant not be prevented. Water contents > 100 ppm in conjunction with ester oils and CFC or HCFC will definitely lead to chemical reactions.

High water contents may have the following effects:

- Corrosion damage
- Copper plating
- Ice crystal formation
- Acid and sludge formation

7.1.2 Corrosion/acid formation

Halogenated refrigerants in combination with refrigeration oils will form under certain circumstances (water content, pressure, high temperature, abrasion) acids in the refrigeration cycle which in turn accelerate corrosion in the materials contained in the cycle. The at times extreme operating conditions cause these reactions to be intensified.

7.1.3 Copper plating

The high water content in the refrigeration cycle leads to the formation of acids removing copper from the surface of the heat exchangers and transporting it to parts with high thermal loads (bearings, valves, ...). The deposit of these copper particles that were released by metallic precipitation can result in the seizing of sliding and bearing surfaces.

7.1.4 Formation of ice crystals

Basically, water is present in every refrigeration cycle. Care must simply be taken that the water content that could lead to a failure of the system does not exceed certain limit values. Unfortunately, most refrigerants have the property to absorb more water than is permissible for the safe continuous operation of the refrigeration system. Refrigerants can be considered extremely hygroscopic. In Fig. 7-1 the water absorption of various refrigerants is represented as a function of the temperature. As the water absorption diminishes when the refrigerants cool down, this effect must be especially observed during cooling in the expansion device since under certain circumstances the water is precipitated from the refrigerant. Since in the case of R 134a the liquid can absorb more water than the vapor, hardly no formation of ice crystals can occur.

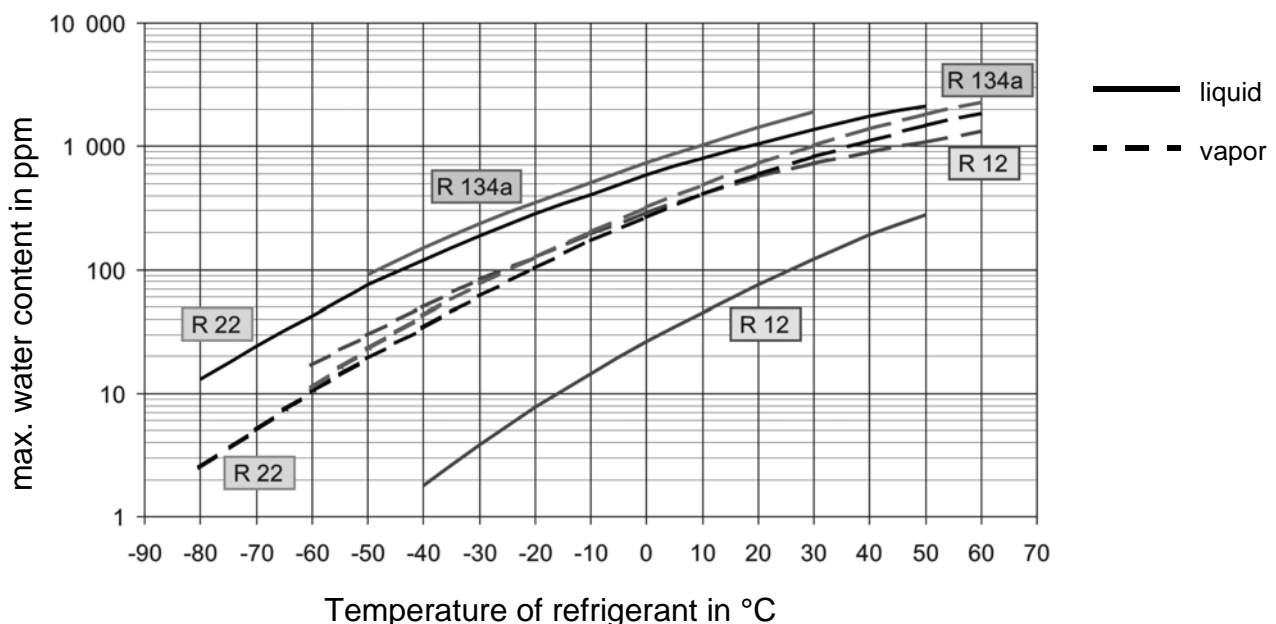


Fig. 7-1. Maximum water content of various refrigerants

7 Commissioning

7.1.5 Reduction of moisture

Water is introduced into the system with every component installed in the refrigeration cycle. For this reason, the following points must be observed already before the installation of the plant:

- Components in stock must, on principle, be protected against ingress of moisture and contamination. The components are delivered and stored with a charge of protective gas and additional protective caps.
- Protective caps of the components should not be removed until immediately before installation. This particularly applies to the drier and the compressor.
- Before the components are used further, sufficient time should be available at the installation site allowing the component parts to adjust to ambient temperature. If cold components are opened, the danger of condensate forming in the components exist.
- Compressors containing an oil charge must not remain open for a prolonged period of time to prevent water from diffusing into the oil. The same applies to oil containers. See also Fig. 7-2. The contents of an oil container holding ester or PAG oil cannot be used any more if the container is left open for a day.
- Used refrigerant should be filtered and dried before it is reintroduced into the refrigeration cycle.

Malfunctions can only be prevented if the water content of the refrigerant is as low as possible and is kept within reasonable limits through use of an adequately dimensioned drier.

To assess the total water content in a refrigeration system, the following points must be taken into account:

- Water in components
- Water ingress during installation
- Water ingress during operation
- Water in lubricant

Ester oil (for bus refrigeration systems) is supplied with a water content of max. 100 ppm, PAG (for cars) with max. 300 ppm.

7.2 Solid and soluble substances in the refrigeration cycle

The required purity of components used in refrigeration plants is defined in DIN 8964. The soluble (grease, oils) and solid (swarf, tinder) residues that can be detected by flushing with solvents must not exceed certain - very low - limit values. The amount and particle size have been defined.

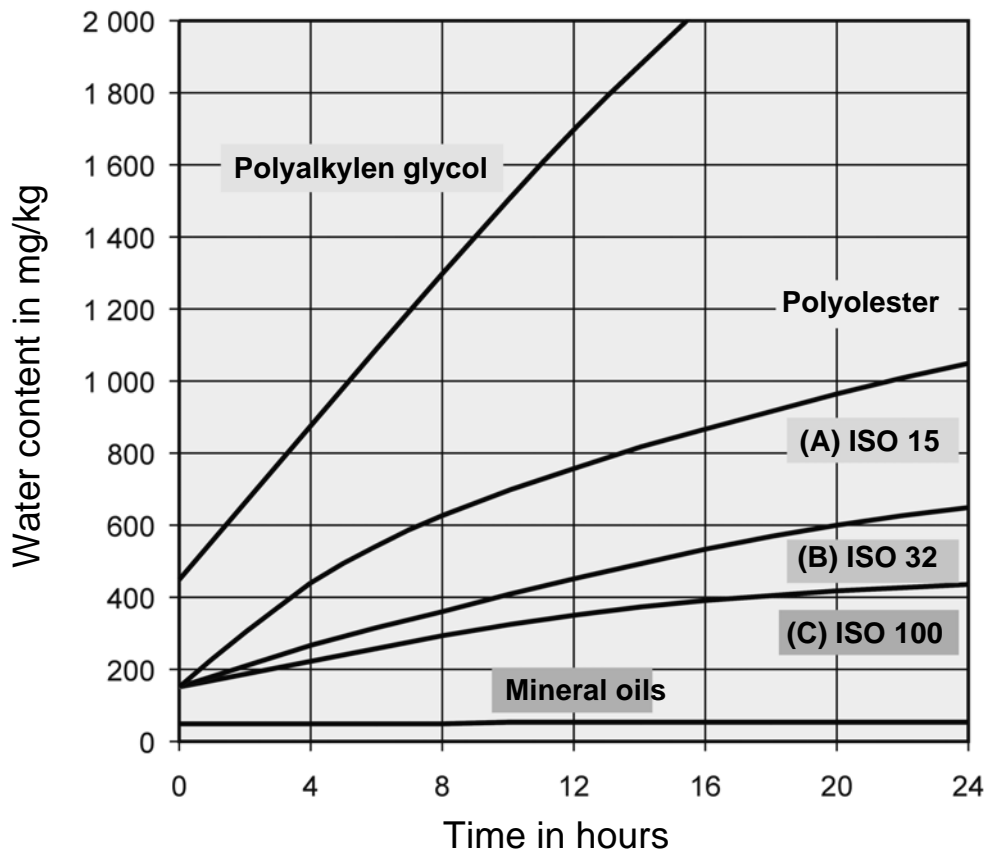


Fig. 7-2. Water absorption of various oil types

7.3 Non-condensables in the refrigeration cycle

Non-condensables in the refrigeration cycle include gases that will not condense at operating pressure of the system, e.g. air, nitrogen, hydrogen, helium, etc. According to DIN 8960 (Nov. 1998), the content of non-condensable gases in refrigerants should not exceed 1.5 % in the vapor phase. These values actually apply to refrigerants, but they can also be extended to a refrigeration system. Only by a thorough evacuation of the entire system can the problem involving non-condensables be avoided. For compliance with the limit value, every refrigeration cycle would have to be evacuated up to a final pressure of 15 mbar, at an atmospheric pressure of 1 000 mbar (to be measured in the refrigeration cycle, not at the vacuum pump).

The problems related to non-condensables are considerable and may under certain circumstances result in a failure of the entire system. In any case, however, non-condensables will on the one hand result in a higher power consumption of the compressor and, on the other hand, to a lower refrigeration capacity. In extreme cases a shutdown due to high pressure may occur.

As a rule, the non-condensables contained in a refrigeration system should be kept to a minimum.

7.4 Evacuation of a refrigeration cycle

7.4.1 General

The evacuation of a refrigeration cycle primarily serves to reduce non-condensables. Only in exceptional cases should the system be dried by using a vacuum pump since the removal of water by suction using the vacuum pump is a very lengthy process. As described above, the pressure measured in the system during evacuation should be < 15 mbar. To accomplish this, the devices for evacuation and pressure measurement must meet certain minimum requirements.

Absolute pressure measurements of 0.1 to 1,000 mbar are only possible using special measuring equipment appropriate for this range. In most of the cases mechanical spring vacuum gauges or capsule element vacuum gauges are used.

Mechanical Spring Vacuum Gauges

These vacuum gauges are designed to measure the vacuum generated relative to the ambient pressure. The vacuum indicated therefore depends on the altitude of the site as well as on the weather situation.

To compensate for the environmental conditions, the commonly used vacuum gauges are provided with a reference pointer. Prior to the evacuation of the refrigeration system, the vacuum pump will pump down the system to the lowest possible point, and the reference pointer is set

at this value. This value defines the maximum attainable vacuum on that day at this specific location and which must be reached again when the refrigeration system is evacuated. When this method is used, the vacuum pump must be regularly checked for adequate quality.

Mechanical Capsule Element Vacuum Gauge

The operating principle of a capsule element vacuum gauge is based on the measuring system of an aneroid-type meter. The aneroid meter consists of a hermetically sealed and evacuated system. The expansion of the meter at decreasing ambient pressure influences the measured-value readout. This procedure enables - independent of atmospheric pressure - pressure measurements up to 1 mbar to be carried out.

7.4.2 Vacuum pump

Vacuum pump, similar to compressors in refrigeration systems, are gas pumping devices that have to overcome a given pressure ratio. The final compression pressure of a vacuum pump is approx. 1,000 mbar. If the suction pressure is, for example, 1 mbar, a pressure ratio of 1,000 needs to be overcome.

Pressure ratios of up to 1,000 and higher are only possible if the moving parts are provided with a sufficient lubricating oil film. One and two-stage vacuum pumps are available. With two-stage vacuum pumps lower final pressures can be obtained.

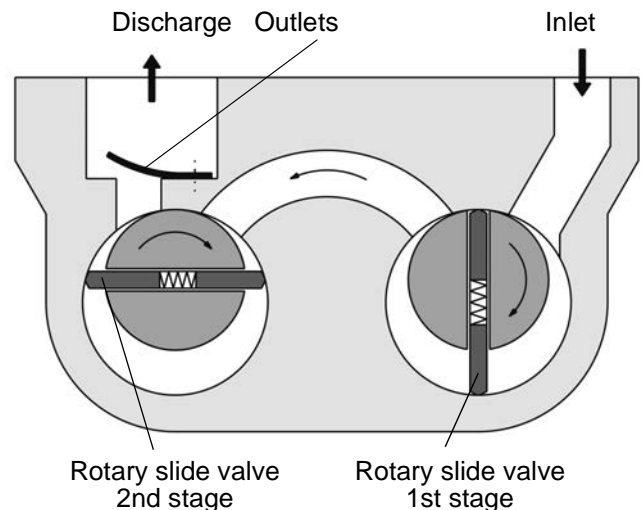


Fig. 7-3. Schematic illustration of a two-stage rotary slide valve vacuum pump

Any operator of vacuum pumps should ensure that any water that may have been removed by suction will not result in a failure of the vacuum pump. Due to the external conditions in the pump housing, the sucked-off water vapor will condense. This results in corrosion and possibly in a decomposition of the vacuum pump oil. Gas ballast operation may reduce or prevent condensate formation. During the evacuation operation, the pump should first be operated with gas ballast to prevent any damage due to moisture.

7 Commissioning

Correct maintenance of a vacuum pump includes regular operation with gas ballast and monitoring of the oil level to ensure proper lubrication of moving parts and hence also the attainment of the final pressure.

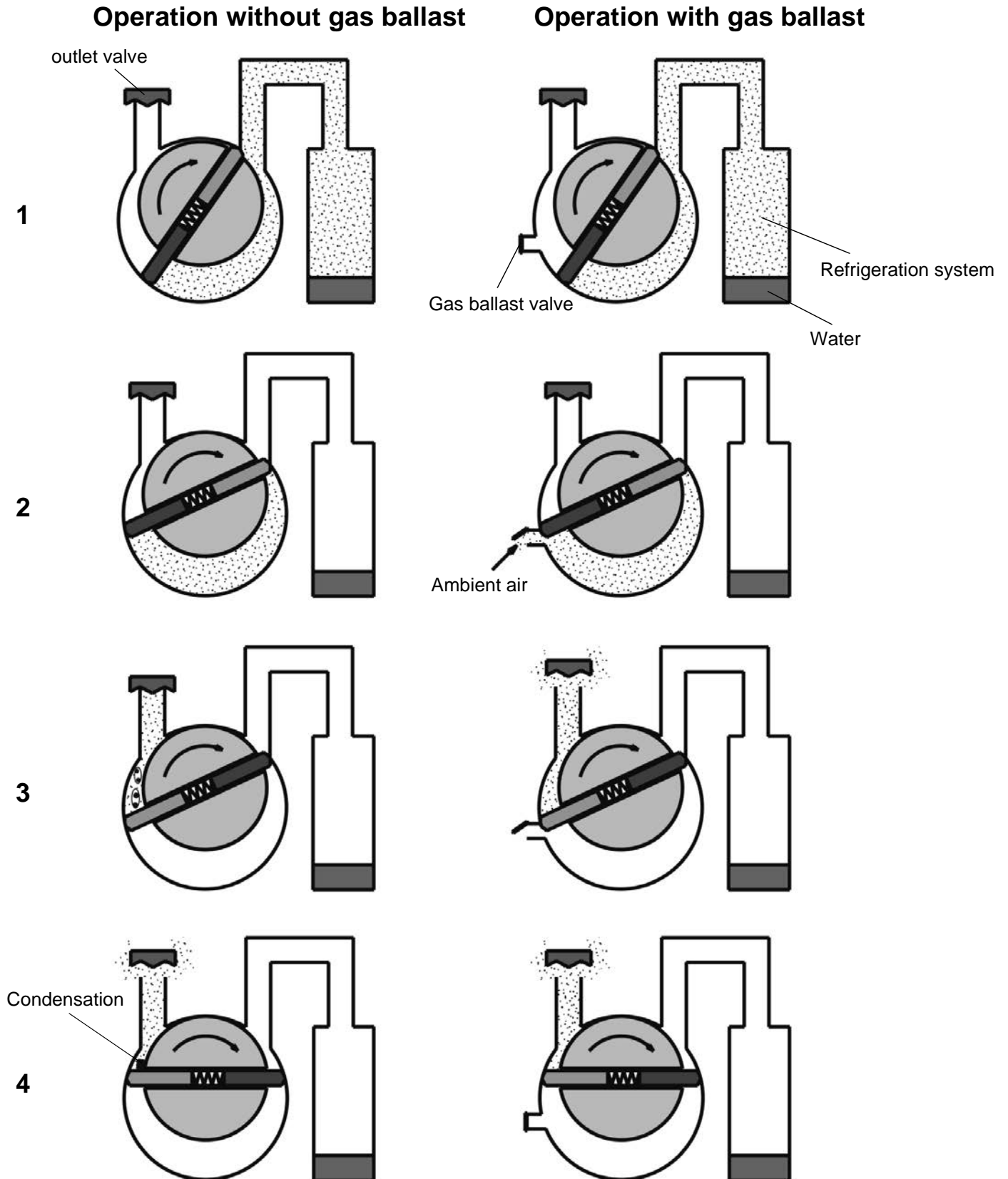


Fig. 7-4 Illustration of the evacuation process with and without gas ballast

Description of the individual steps during the evacuation procedure without gas ballast

Fig. 1

The pump is connected to the vessel that is already nearly void of air (approx. 70 mbar). It must therefore deliver almost only vapor particles.

Fig. 2

The suction chamber is separate from the vessel. Compression starts.

Fig. 3

The suction chamber content has already been compressed to such an extent that the vapor condenses into droplets. Overpressure has not been reached yet.

Fig. 4

Only now is the required overpressure generated by the residual air and the exhaust valve opens. The condensed vapor remains in the pump and may in the long term result in corrosion.

Description of the individual steps during the evacuation process with gas ballast

Fig. 1

The pump is connected to the vessel that is already nearly void of air (approx. 70 mbar). It must therefore deliver almost only vapor particles.

Fig. 2

The suction chamber is separate from the vessel. Compression starts. The gas ballast valve opens and the suction chamber is additionally filled with air (the gas ballast).

Fig. 3

The exhaust valve is pushed open by the additional gas ballast air. Vapor and air can escape and no more undesirable condensate can form.

Fig. 4

The residual vapor/air mixture is expelled. In the case of evacuation without gas ballast somewhat lower final pressures are attained.

7.4.3 Correct evacuation

Evacuation for reducing non-condensables

To comply with the requirement of a permissible residual air content of 1.5 % by volume, the entire system must have a residual pressure of < 15 mbar. If possible, a cycle should always be evacuated on both sides, i.e. the high- and low-pressure sides. It must always be checked whether existing components such as the expansion valve or spring-loaded check valves possibly impede the evacuation of entire system sections. Since the evaporating pressure in the thermostatic expansion valve along

with the spring constitute the closing valve component, this valve can be evacuated one-sided, i.e. from the low-pressure side. A very low pressure exists during evacuation, the valve senses very high overheat and opens completely.

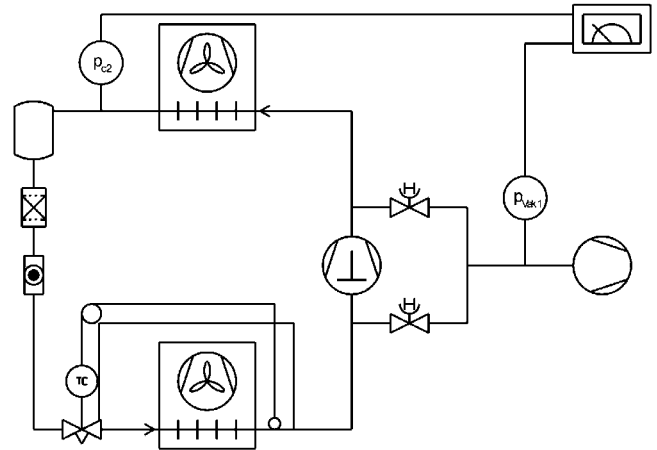


Fig. 7-5 Schematic illustration of two-sided evacuation

To be able to evaluate the final pressure attained it is necessary to measure the pressure at a point in the system as far from the vacuum pump as possible. Where this is not possible, interval evacuation should be carried out. Here, the pressure gauge is located in the direction of flow ahead of a shut-off valve at the vacuum pump. The shut-off valve is to be closed from time to time and the prevailing pressure to be read off the pressure gauge.

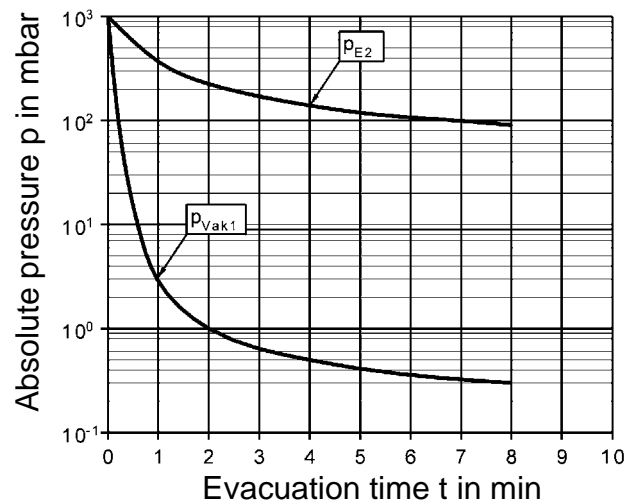


Fig. 7-6. Pressure curves at the vacuum pump and in the system

The pressure curves show that directly at the vacuum pump the pressure is lower than in the system. To be able to ensure the specified 15 mbar, it is therefore necessary that the pressure be measured in the system and not directly at the vacuum pump. Failing this, large amounts of non-condensables may accumulate in the system after commissioning resulting in malfunctions or shutdowns due to high pressure.

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Evacuation for drying the system

Any water that may be present in a refrigeration cycle may in exceptional cases be removed using a vacuum pump. Since water can only be removed in the vaporous state, the liquid must first be converted into vapor. To this end, either heat must be supplied or the pressure decreased.

Table 7-1. Vapor pressure of water as a function of the temperature

Temperature in °C	40	20	0	-10	-20
Vapor pressure of water in mbar	74	23	6,1	2,6	1,0

The table shows the dependency of the vapor pressure on the temperature. If, for example, warm water with a temperature of 20 °C is to vaporize, a pressure level of 23 mbar must be attained. Only then can the water vapor be pumped down. During evaporation, however, the temperature of the water drops as well. If large amounts of water are to vaporize, the required evaporation heat must be supplied externally, as otherwise extremely low pressures would be necessary and the time required for evacuation would be too long.

The specific volume of water vapor is another factor that must be taken into account. Due to the high specific volume at low pressures, pump-down should be carried out with a pressure drop as little as possible, via the suction lines of the vacuum pump. This is necessary in order to keep the suction pressure as high as possible hence achieving short pump-down times. An acceleration of the drying process can be accomplished by "breaking the vacuum". In this case, evacuation is performed first and then dried nitrogen is charged into the system. The nitrogen adsorbs a large part of the remaining water. Upon a second evacuation this water is pumped down along with the nitrogen thus quickly reducing the water content existing in the system.

7.5 Leak tests in refrigeration cycles

7.5.1 General

Demands on the leaktightness of refrigeration cycles have considerably increased over the years. Absolute leaktight components and systems, however, do not exist (hydrogen will even diffuse through metal screens). DIN 8964 specifies limit values concerning leak tightness. For components with an inner volume up to 5 dm³ the annual refrigerant loss at a test pressure of 10 bar must not exceed 2.4 g of R 134a.

7.5.2 Pressure leak test

This method is a leak test that only provides information on the total leak tightness of a system. The unit under test

is charged with nitrogen, for example, until the subsequent operating pressure is reached. Then the charging valve is closed. The pressure drop is now recorded as a function of time. For R 134a a detection sensitivity of approx. 250 kg/year is determined. This method is therefore a gross leak test. The same is true for the identical vacuum method where the system is tested in the vacuum.

7.5.3 Bubble test

The pressurized unit under test is completely immersed into a tank filled with water. The gas bubbles that can be seen escaping enable the specific leak to be detected. With a refrigerant loss of 280g/year (R134a) it takes 13 sec. for a bubble to form. With a leak rate of 15 g/year 290 sec. are required. The subsequent drying of the submerged components is often found to be a disadvantage.

7.5.4 Soap bubble test

A soap solution (leak detection spray) is sprayed on the joints or components suspected of leaking. Bubbles will indicate any leaks. The detection limit of R 134a is 250 g/year.

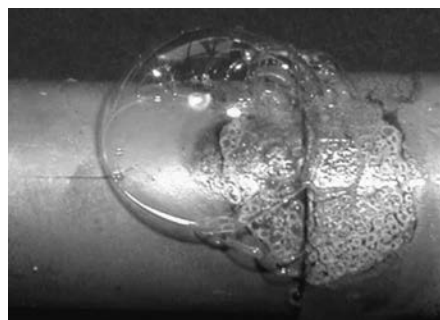


Fig. 7-7 Soap bubble leak test

7.5.5 Leak test by means of electronic leak detector

Here, the detection sensitivity is approx. 0.2 to 20 g/year for R 134a. Very satisfactory results can be obtained with these devices. Chlorine-free refrigerants and partly also other test gas leaks can be reliably pin-pointed. The specified leakage rates can be verified using these devices.

When checking for leaks with the electronic leak detector, the sniffer instrument must be passed over all joints starting at the top. R 134a is heavier than air and therefore sinks to the bottom. If leak detection is started at the bottom, incorrect readings may be obtained as a result of refrigerant that may have emerged further up.



Fig. 7-8 Electronic leak detector

7.5.6 Leak test using test gases

For the leak detection by means of leak test gases mainly hydrogen or helium are used as test gases. With this method the complete unit under test is placed into an evacuated chamber and is charged, for example, with helium. A helium gas detector measures the helium leakage by recording the helium atoms escaping. Such an instrument has a measuring accuracy that detects annual leak rates for R 134a of 0.07 g/year. In combination with a gas sniffer this instrument can also be used for local leak detection.

7.6 Service units

For the commissioning or service of a refrigeration system frequently so-called service units are used. A service unit contains, depending on the type, the following components:

- Vacuum pump
- High- and low-pressure gauge
- Vacuum gauge
- Suction unit
- Oil vessel
- Filter-drier



Fig. 7-9. Service unit

The service unit with its hoses can be connected to the system hence considerably facilitating service and commissioning activities.

7.7 Service gauge manifold

For the measurement of operating pressures and for service purposes, frequently a service gauge manifold in addition to the service unit is used. The high- and low-pressure

side of the system can be connected to this manifold as well as the vacuum pump, the evacuation station or refrigerant cylinders. The service gauge manifold is equipped with a high-pressure and a low-pressure gauge. Some versions are also available with an additional vacuum gauge.

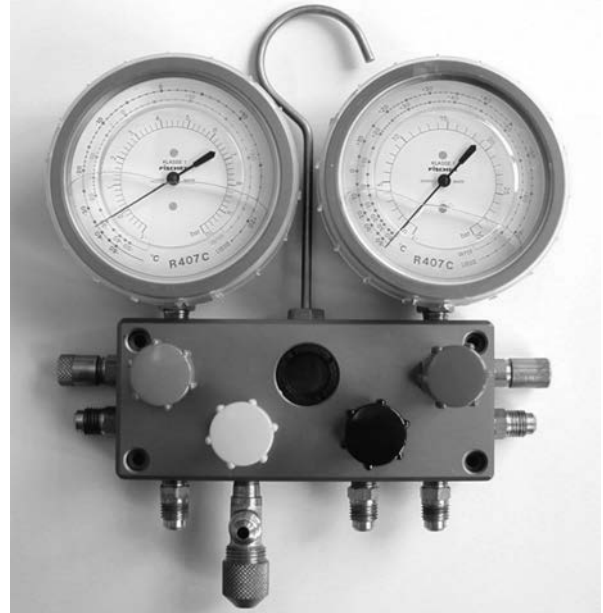


Fig. 7-10 Service gauge manifold

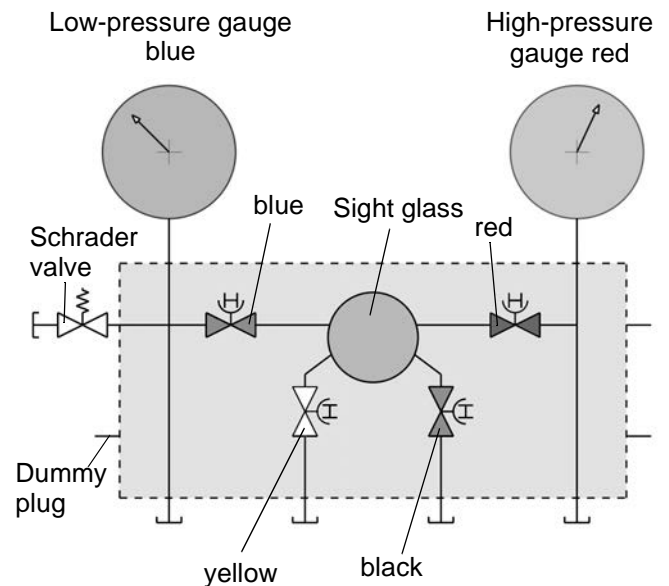


Fig. 7-11. Schematic view of service gauge manifold

7 Commissioning

Steps to be performed for fitting and removing the service gauge manifold to and from a refrigeration system

- Turn stems of service valves to the back seat position.
- Carefully remove protective caps from connections. Pressure may have built up underneath caps owing to a leaky stuffing box.
- Connect hoses between low-pressure and high-pressure sides of service gauge manifold and the connections of the service valves.
- Ensure that the needles of the gauges are positioned at 0 bar.
- Connect vacuum pump to the yellow or black connection.
- Briefly evacuate service gauge manifold and hoses to prevent air from entering the system. When so doing, open the yellow or black valves as well as the red and blue valves. When switching off the vacuum pump first close the shut-off valve to ensure that no air can flow back in the case of a defective check valve of the vacuum pump.
- Close all valves on the manifold.
- To open the service valve in the low-pressure side turn the valve stem approx. one turn clockwise.
- To open the service valve in the high-pressure side turn the valve stem approx. one turn clockwise.
- Record pressures
- When removing gauges, first back seat the service valve in the high-pressure side.
- Open the red and blue valves enabling the compressor to evacuate the hose and manifold against the closed service valve.
- Where fitted, activate the “pump-down feature“ in order to draw as much refrigerant as possible from the service gauge manifold.
- Back seat the service valve in the low-pressure side.
- Close all valves on the service gauge manifold, remove all hoses from the service valves and connect them to the fittings of service gauge manifold as otherwise water vapor will condense into the hoses that are cold from the evacuation process.
- Replace the service valve caps and tighten.
- Remove hoses once they have reached room temperature and replace caps on the fittings of the service gauge manifold.

It is mandatory that all relevant safety regulations be adhered to when handling refrigerants.

7.8 Evacuation equipment

An evacuation unit is required if the refrigerant is to be removed from a system. Depending on the type, this may be used for various refrigerants, liquid and vapor evacuation as well as refrigerant cleaning/recycling, and possibly for lubricant removal purposes.

According to EN 378-4 (Sept. 2000) the transfer or evacuation of refrigerants should be performed as follows:

If the compressor of the refrigeration system cannot be used for refrigerant transfer, a recovery device must be connected to the refrigeration system to either transfer the refrigerant into another part of the refrigeration plant or into a separate container. The latter must not be overfilled. With a refrigerant-oil mixture, the useful capacity must be reduced to approx. 80 % by volume of liquid (must be weighed!). The permissible filling weight of the cylinder must be observed (nameplate). Special recycling cylinders are required. Most normal refrigerant cylinders are protected against excessive charging by a check valve.

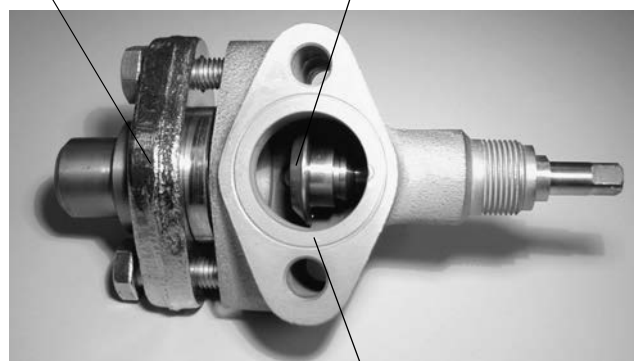
Prior to any maintenance, repair, etc., for which the refrigeration system must be opened, the pressure of the refrigeration system must be reduced to 0.05 bar absolute. The refrigerant concentration in the plant is thus reduced to a minimum to prevent any unnecessary outflow of refrigerant or to keep refrigerant breakdown during soldering to a minimum.

During the evacuation process above a liquid seal in containers, lower segments, pockets, pipe bends, etc., the liquid becomes increasingly colder and hence the pressure increasingly lower. It is possible that with a final pressure of 0.3 bar large amounts of very cold liquid are present in the system. From the outside, this area can be recognized by frost formation or condensation. Opening the system could result in the sudden release of this liquid and hence to personal injuries. During the evacuation process heat should be specifically applied to these places. Moreover, the time required for evacuation is considerably reduced. To prevent condensation of water in the plant, the refrigeration cycle should not be opened until all components have been warmed up to ambient temperature.

7.9 Service valve

Compressors are frequently fitted with shut-off valves on the discharge and suction sides.

Connecting flange
suction- or discharge line conical nipple



Compressor connecting flange

Fig. 7-12 Compressor shut-off valve

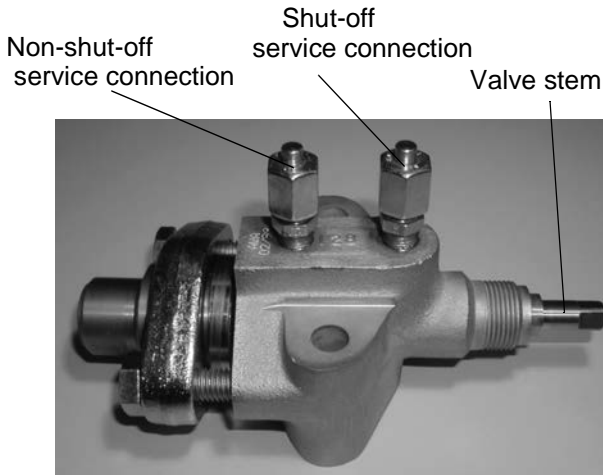


Fig. 7-13 Compressor shut-off valve, top view

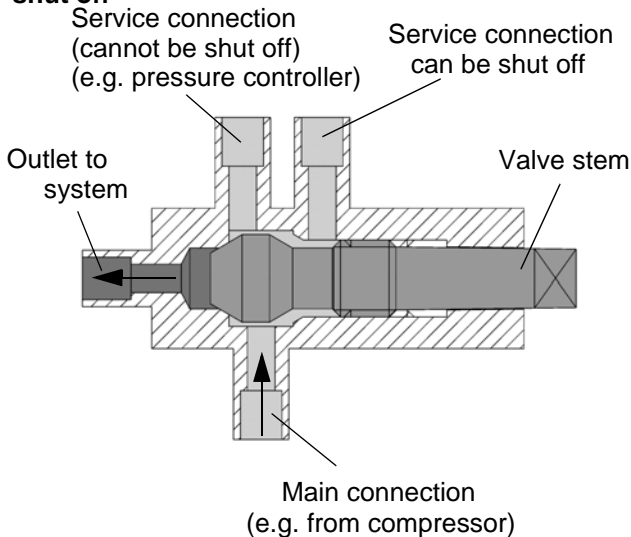
The valve shown above features two service connections in addition to the connections on the compressor and the flange for connection to the suction and discharge lines.

The connecting fitting that cannot be shut off toward the compressor is designed for the connection of a safety device. A service gauge manifold may, for example, be connected to the shut-off service connection. If the valve stem is back seated (completely open), the main line is open, whereas the shut-off fitting is closed. To open the service fitting, the valve stem must be turned approx. one turn in clockwise direction ("closed").

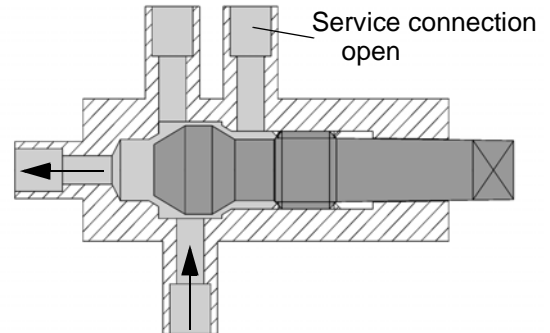
The valve stem is protected with a seal cap (not illustrated) due to possible leaks around the stuffing box.

The illustration below shows the function of a service valve in three positions.

Valve stem turned completely to the left, main line shut off



Valve stem in midway position, all ports connected



Valve stem turned completely to the right (back seated)

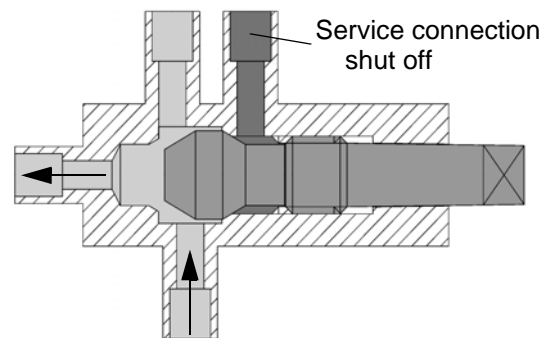


Fig. 7-14 Service valve in three positions (schematic)

- Position 1: compressor shut off
- Position 2: pressure check
- Position 3: normal operation

A refrigeration system typically has two service valves that are fitted as closely to the compressor as possible, both on the high-pressure and the low-pressure side.

The Schrader valve illustrated below is frequently used as a charging or service valve.

A needle valve insert is screwed into the valve body. By screwing a suitable adapter, such as a charging hose with a "depressor" onto the threaded pin, the valve insert is pushed downward and the valve is opened against the action of a spring.

When the adapter is removed, the valve will close again automatically. Upon connecting and removing a small amount of refrigerant will escape.

To prevent refrigerant losses due to leaky valve inserts and to protect the valve from damage, a seal cap must be fitted.

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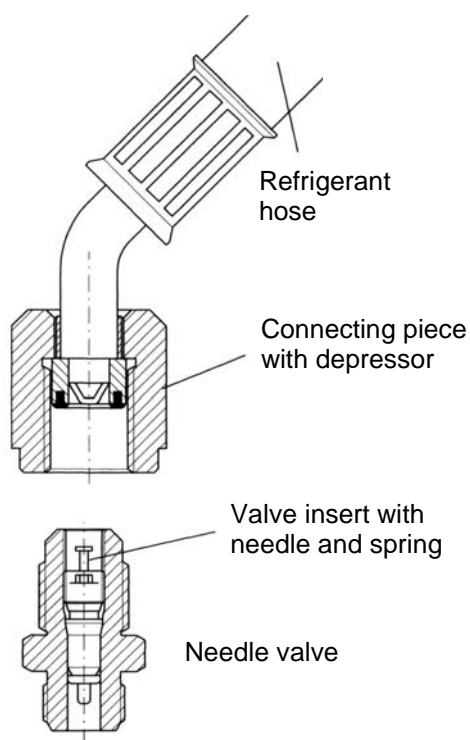


Fig. 7-15 Automatic needle valve (Schrader valve)

Another variant is the combination needle valve/quick release adapter. This variant allows refrigerant loss during connection and disconnection to be kept to a minimum.

The valve is opened or closed by the hand wheel of the quick release adapter.

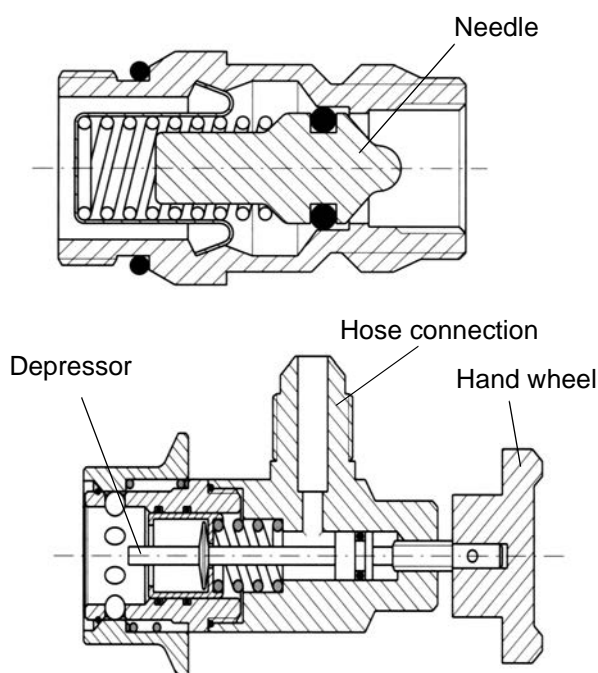


Fig. 7-16 Quick release adapter for service valves

7.10 Commissioning procedure

This chapter contains a description of the commissioning procedure of a small refrigeration plant in accordance with DIN 8975-5. All activities refer to a completely assembled system.

7.10.1 Visual Inspection

At the completion of all required work a check must be carried out to verify that all lines have been soldered or screwed together and all components properly installed. In the case of components with a defined installation position or location check that the mounting instructions have been complied with. In addition, the components must be inspected for any damage.

7.10.2 Pressure test

The purpose of the pressure test is to ensure that the system is capable of withstanding all pressure loads encountered during operation. Refrigerant loss and personal injuries must be avoided.

The pressure test is carried out with dry nitrogen. The test pressure is the permissible operating pressure of the individual pressure stage. The permissible operating pressure can be determined as defined in DIN 8975-1 (minimum requirements for refrigeration system components, primarily applicable to stationary refrigeration plants, however), or on the basis of the nameplate ratings of the components (weakest link in the chain). It must be ensured that no system parts are shut off during the pressure test and that the components are not subjected to over-loading.

Charging with nitrogen must be carried out slowly and steadily via a pressure reducing valve. It is not permissible to connect the nitrogen cylinder without a pressure reducer.

7.10.3 Leak test

A general air tightness check is first performed by brushing a water/soap solution around all joints at a gauge pressure between 1 bar and the maximum allowable operating pressure. It would seem practical to use the same charge that was used for the pressure test.

For the detection of leaks using an electronic leak detector it is required that the system is charged with a refrigerant. The leak rates defined by currently valid regulations can only be checked by means of the electronic leak detector.

7.10.4 Drying, evacuation

Evacuation is mostly not sufficient for drying purposes. It is advantageous, to first evacuate the system at system temperature (e.g. 23 mbar at 20 °C) to the water vapor

pressure. By “breaking” the vacuum (with a small charge of dry nitrogen) and evacuating the system again, the water content existing in the plant can be reduced more quickly.

During the evacuation procedure it must be ensured that all plant parts can be evacuated and that the vacuum pump has the required capacity.

To keep the content of non-condensables in the plant to a minimum, the plant must be evacuated to a vacuum as deep as possible before charging is carried out, but to below 15 mbar at the minimum.

The vacuum attained must be checked with a vacuum meter.

7.10.5 Charging, determination of the required refrigerant charge

On no account must the compressor be operated without refrigerant.

If the **filling capacity of the plant is known**, liquid charging of the refrigerant into the high side of the system can be carried out with the plant at rest, against vacuum, using a charging cylinder or scales.

Be extremely careful when charging liquid refrigerant to the low side of the system. On no account must large amounts of liquid enter the compressor. R 134a is a single substance refrigerant and may therefore be charged into the system from the refrigerant cylinder in vapor or liquid form.

If the fill quantity is yet to be determined, first fill in refrigerant until the low-pressure cutouts respond and the compressor can be started. In general, it suffices to charge half the rated quantity to be able to operate the compressor during the charging operation without compressor damage.

1. System with capillary or expansion tube

One of the charging criteria is superheat. At full load (high evaporator supply air temperature) so much refrigerant must be charged that slight superheating at the evaporator outlet takes place, or the phase change at the evaporator outlet has just been completed (0 K superheat). At part load the surplus refrigerant collects in the separator. A crucial factor for the evaluation of a system is the air out-

let temperature at the evaporator.

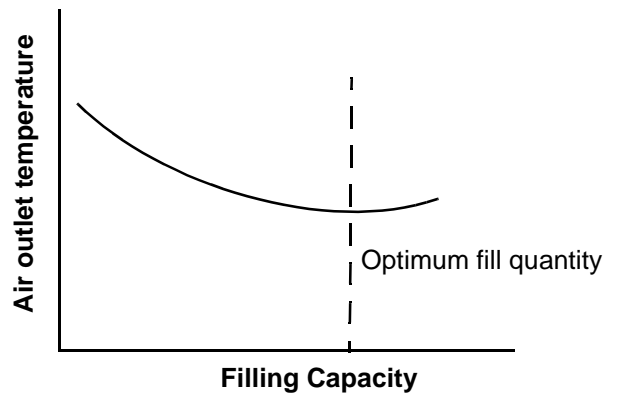


Fig. 7-17 Optimum filling capacity

The dependency of the air outlet temperature on the refrigerant capacity is shown in Fig. 7-17. The point with the lowest air outlet temperature determines the optimum filling capacity of the plant.

2. System with thermostatic expansion valve

Depending on the sensor charge, the superheat criterion at the evaporator outlet at part-load or full-load mode is of foremost importance. As long as the system is not fully charged, liquid refrigerant containing vapor bubbles flows to the expansion valve. Even with the valve fully open can the evaporator not be sufficiently supplied with refrigerant. The superheat of the suction vapor at the evaporator outlet is high. During the charging operation superheating decreases continuously. At the point at which the superheat enters the steady state condition, i.e. the valve receives pure liquid and exhibits correct control characteristics, the system receives the minimum charge. Normally some additional charge of refrigerant is added to the system. The charge must be checked under different operating conditions.

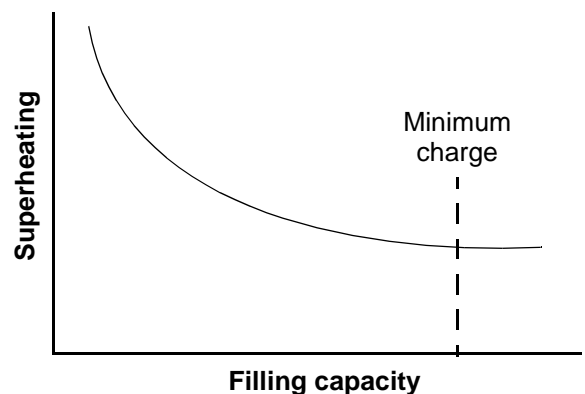


Fig. 7-18 Superheat as a function of filling capacity

If a **sight glass** is installed in the liquid line, this can be used for determining the minimum charge. No more vapor bubbles must be visible.

7 Commissioning

Without receiver the second criterion, i.e. subcooling, can be considered. Typically, the optimum filling quantity corresponds to 2 - 3 K subcooling.

With receiver the steady state value of superheating is the criterion for the minimum filling quantity. Additional refrigerant is collected in the receiver. Only when the receiver is filled to overflowing can subcooling be measured. The plant is then overcharged. The system with a receiver fitted is less susceptible to overcharging.

Monitoring the condensing pressure is another option for assessing the filling quantity. Fig. 7-19 shows the relevant interrelationships. In the case of undercharging a relatively low condensing pressure exists. If refrigerant is charged the condensing pressure rises until a given liquid level is obtained in the receiver. While the receiver is being filled the pressure remains constant. Once the receiver is full, refrigerant backs up into the condenser, the useful condenser area is reduced and the condensing pressure rises further.

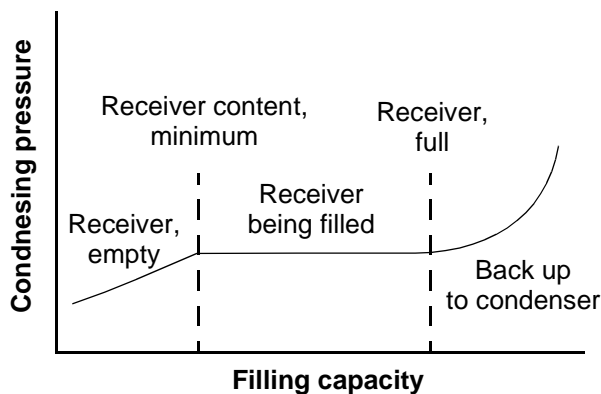


Fig. 7-19 Condensing pressure as a function of filling capacity

In most of the cases, the lowest refrigerant filling level in the receiver is attained at:

- part load
- 5 - 8 K below the MOP point
- maximum condensing pressure

Charging at an evaporating pressure near the MOP point is not possible since the operating superheat is too high.

The required filling capacity must be documented on the nameplate.

7.10.6 Check and adjustment of safety devices

After the completion of the charging process the adjustment and functioning of the safety devices must be checked.

8 Troubleshooting

8.1 Requirements

For the performance of troubleshooting activities two requirements must be met:

- competence
- technical equipment

These two terms are mentioned in the prohibition on the production of HCFC-Halon. Also VBG 20 (regulation of the trade association) emphasizes the competence required for carrying out service activities.

Technical Equipment

The most important tools for troubleshooting are pressure gauge and thermometer. The refrigerant states such as superheating and subcooling provide essential clues in locating a fault. Also our own senses are valuable tools in the location of faults. Gas bubbles in a sight glass, contamination and frost formation can be perceived with your eyes. If the compressor draws in wet vapor this can be observed on the suction line and under certain circumstances also be heard. And an overloaded compressor can even be smelled.

For troubleshooting activities the following tools and aids should be available:

- service station
- pressure gauges
- thermometers
- dry nitrogen
- refrigerant cylinders for new refrigerant
- refrigerant cylinders for used refrigerant
- collection vessel for used oil
- vacuum pump
- hoses
- scales
- purge station
- leak detector

The testing instruments must be inspected at a regular basis. Calibration, however, may only be carried out by an officially approved test facility.

Pressure gauges

Pressure gauges used for servicing are in most of the cases used for measuring gauge pressure. Any temperature scales on the pressure gauges always refer to absolute pressures. Keep in mind that it is not possible to measure the temperature directly with a pressure gauge. Temperature values are pure reference values. Merely the saturation temperature may also be indicated on the pressure gauge dial. A lower temperature applies if the refrigerant is in the liquid state. If it is gaseous the temper-

ature will be above the saturation temperature. Pressure gauges that are not connected to the system must indicate a pressure of 0 bar. Pressure gauges should feature a setscrew enabling an adjustment of the settings.

Thermometers

As a rule, digital thermometer with surface or clip-on probes are used. In particular in the case of great temperature differences an appropriate thermal insulation of the measuring point must be ensured. It is recommended that a heat conducting paste be applied sparingly. If the measuring point is heavily contaminated, it must be cleaned and treated with fine emery paper, if necessary. Only the temperature of the probe will be measured. Lacking thermal insulation and the presence of insulating oxide deposits on the line may lead to temperature differences of a few Kelvin between the measured and actual values.

8.2 Troubleshooting procedures

8.2.1 Refrigerant

As a first step, the refrigerant contained in the system must be determined. In accordance with the regulations, the refrigerant and the filling capacity must be stated in the operating manual of the plant and on its nameplate.

8.2.2 Plant layout

It is not possible to carry out any troubleshooting activities without any knowledge of the design of the plant, the parts and components installed as well as their principle of operation.

8.2.3 Visual inspection

Some faults can be detected visually or recognized intuitively if you have the necessary experience. Compressor contamination or formation of vapor bubbles in the sight glass are a frequent cause of trouble and can be quickly detected.

In the more unusual case of frost formation on the evaporator, the frost pattern provides valuable hints. Partial frosting only on the inlet side of the evaporator is a clear sign of insufficient refrigerant supply, which in turn suggests that the expansion device is not operating properly or that a refrigerant shortage exists. Incomplete frost formation points to load problems, i.e. insufficient air flow rate or none at all.

Even superheating can sometimes be perceived by observation. At the end of the evaporator there must be an area that is dry - or at least drier - at evaporating temperatures above -2 °C.

The liquid line of the refrigeration plant is warm. If local cool spots can be felt or if condensation water is formed,

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this is an indication of a severe pressure drop across the line. Also in the case of clogged filters similar occurrences can be observed.

The hot gas line is so hot that you can touch it only briefly. Exceptionally cold pressure lines are an indication of the compressor drawing in liquid refrigerant.

The oil level in the compressor sight glass provides information on the amount of oil or the oil return in the system. However, the oil level may be significantly influenced by condensed refrigerant. Discolorations permit conclusions on the condition of the oil to be drawn.

It is easy to detect water in the system via a sight glass with moisture indicator.

Universal rules have the disadvantage that they can be applied most of the time, but not all the time. The refrigerant conditions prevailing in the individual line sections or components must therefore be determined in an unambiguous way by pressure and temperature measurements.

8.2.4 Measurements

The flow diagram (Fig. 8-1) contains “minimum informa-

tion” required for the evaluation of the system or troubleshooting activities. The more complex and widely branched a system is, the more measuring points are required.

Details on evaporator superheating can be inferred from the temperature and the pressure at the evaporator outlet. **Superheat** is a good indicator for the evaporator charge. If superheat is excessive, too little refrigerant is supplied to the evaporator, if superheat is too low, the evaporator receives too much refrigerant. It must then be checked on a case by case basis whether the expansion valve or undercharging has led to this situation. An appropriate differentiation is only possible if an unambiguous indicator for the amount of refrigerant charge exists.

Details on **subcooling** can be inferred from the pressure and the temperature at the evaporator outlet. This allows the refrigerant charge to be assessed. In systems equipped with a receiver, the sight glass in the liquid line is best suited to provide information on the filling quantity. Subcooling in this case is more an indication of overcharging.

The hot gas temperature is an indication of whether the compressor is operating within the permissible operating limits.

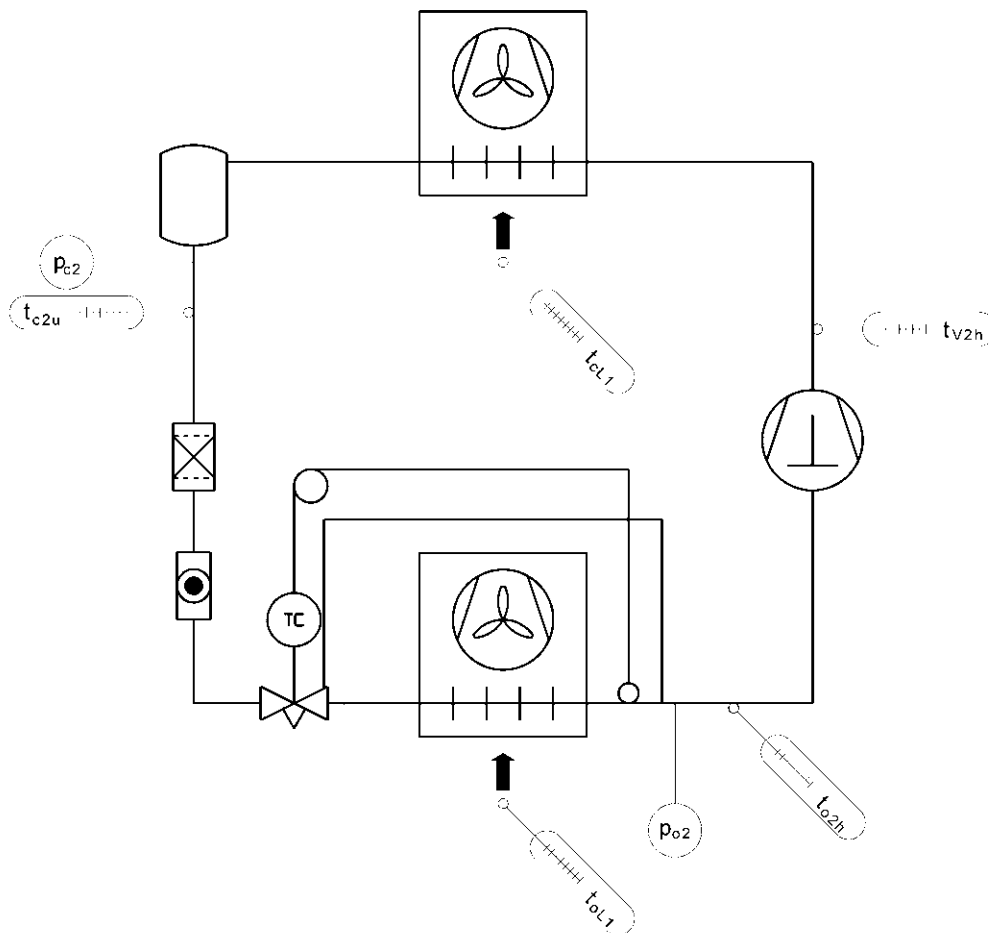


Fig. 8-1 Measuring points for evaluation during troubleshooting

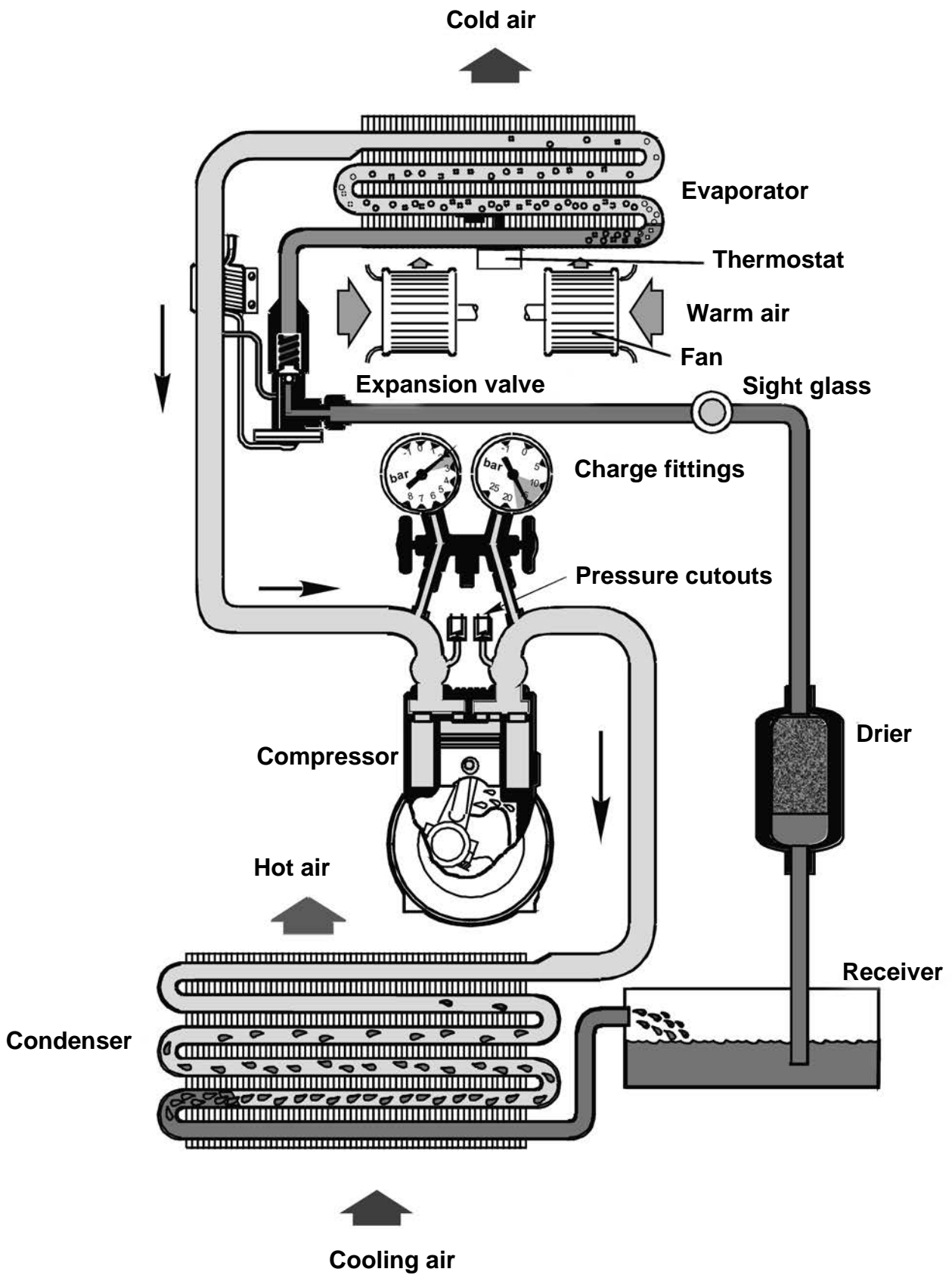


Fig. 8-2 Schematic illustration of the refrigeration cycle with gauge manifold as charge fittings

8 Troubleshooting

8.3 Typical malfunctions and possible causes

Most refrigerant-related malfunctions can be allocated with the aid of a checklist (e.g. Tables below). Malfunctions frequently have similar symptoms but different causes. Evaporator frosting may, for example, be completely normal. However, it may also be the result of a defective expansion valve, oil return problems or the evaporator being insufficiently dimensioned, etc. A systematic consideration of these malfunctions by measurements (superheating, subcooling, hot gas temperature, ...) and observations (sight glass, frost pattern, ...) enables them to be identified.

As is shown clearly in Fig. 8-3, the most frequently encountered problems resulting in compressor failure are lack of oil (oil return problems), refrigerant migration and electrical faults.

High- and low pressures as a function of ambient conditions (compressor speed, inside temperature, outside temperature, relative humidity) are also frequently used for troubleshooting purposes. To be able to use this method it is indispensable that the operating values of the particular system under normal operating conditions are known. Any universal application to any number of systems is hardly possible. The installation of an internally-controlled swash plate compressor limits, for example, the low pressure in the case of compressor contamination, or a failed fan, to a preset value. If one were only to concentrate on the low pressure, these faults would therefore not

be detectable. Additional measurements, e.g. of air inlet and air outlet temperatures, are necessary.

The charts below state the pressure values to be expected in a system at differing outside temperatures (measured at medium speed).

Suction pressure (low-pressure gauge)

Table 8-1. Expected suction pressure

Outside temperatures in °C	Gauge pressure in bar
25	approx. 2.0
30	approx. 2.5
35	approx. 3.0

Discharge pressure (high-pressure gauge)

Table 8-2. Expected high pressure

Outside temperatures in °C	Gauge pressure in bar
25	approx. 8
35	approx. 13
40	approx. 16
45	approx. 18

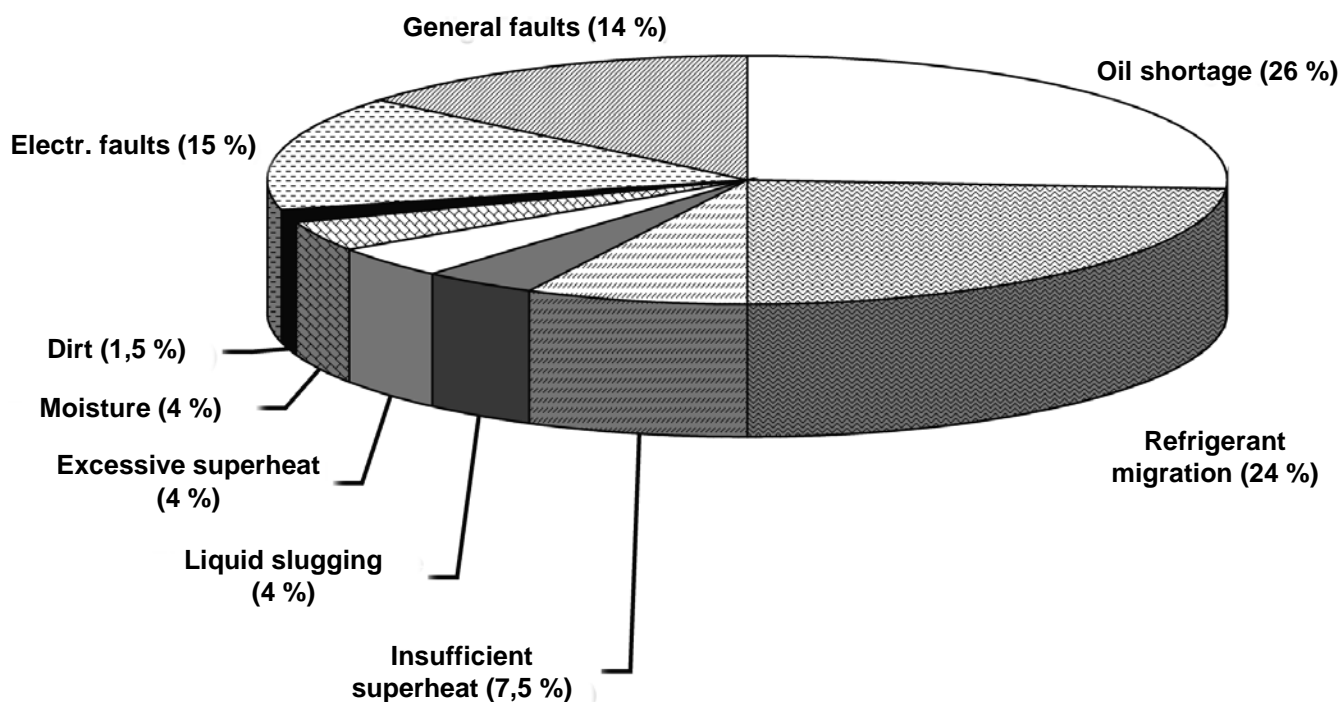


Fig. 8-3 Causes of malfunctions of approx. 40 000 failed compressors

Since the pressures occurring in a refrigeration plant greatly depend on ambient conditions it is of crucial importance to be familiar with these dependencies. Table 8-3 contains some of these dependencies.

Table 8-3. Parameters influencing operating pressures

Measured variable		Suction pressure		Discharge pressure	
		rises	falls	rises	falls
Compressor speed	rises		X	X	
	falls	X			X
Vehicle inside temperature	rises	X		X	
	falls		X		X
Outside temperature	rises	X		X	
	falls		X		X
Relative humidity	rises	X		X	
	falls		X		X

For troubleshooting purposes, so-called fault diagnosis slides are used. Fig. 8-4 shows an example of such a slide available from Bock. As a rule, these slides offer several possible causes of error for a given problem. For a more detailed fault allocation measurements for the evaluation of the cycle and experience of the person performing the troubleshooting activity are required.

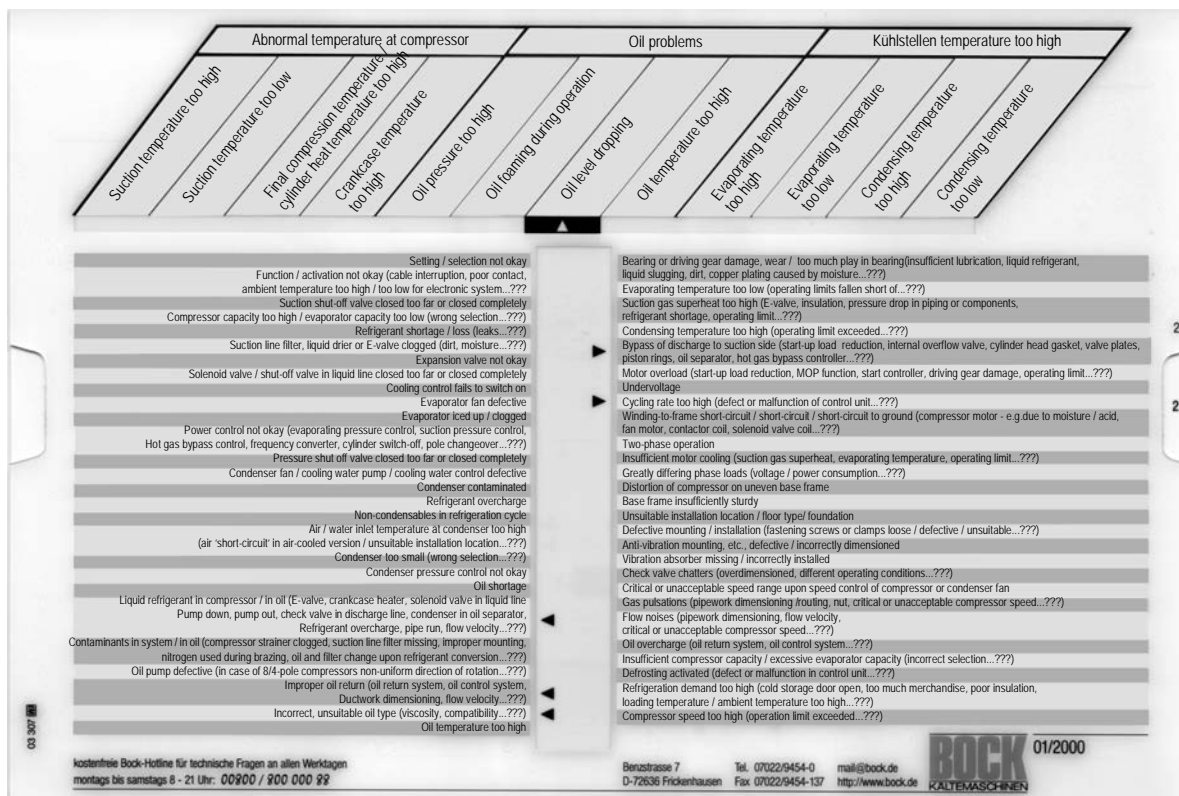


Fig. 8-4 Fault diagnosis slide available from Bock

In the following malfunctions are listed based on measured pressure values concerning suction pressure and discharge pressure of the refrigeration cycle.

8 Troubleshooting

8.3.1 Suction pressure too low, discharge pressure too low to normal

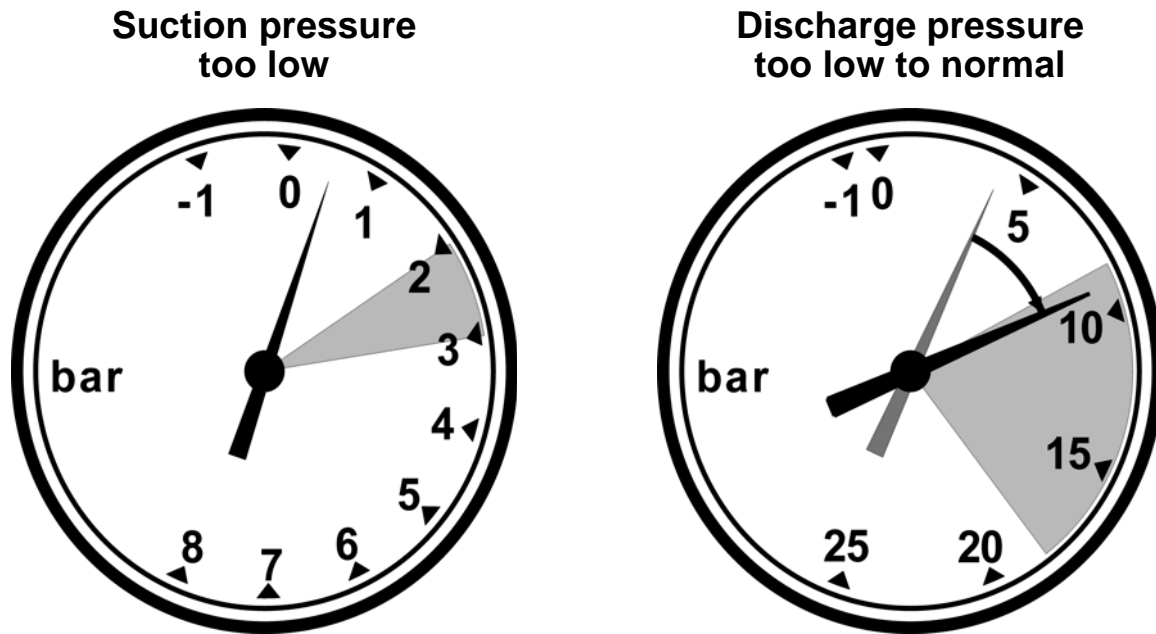


Fig. 8-5 Suction pressure too low, discharge pressure too low to normal

Table 8-4. Possible faults at low suction pressure and low to normal discharge pressure

Cause	Possible effect	Remedy
Shortage of refrigerant	no subcooling, bubbles in sight glass, extreme superheat, evaporator frosting	leak detection, top up
Suction shut-off valve on compressor throttled or closed	suction pressure gauge indicates a vacuum	check valve position and adjust
Evaporator fins or air filter contaminated	insufficient cooling capacity	clean
Evaporator fan failure	low pressure cut-out	repair fan
Brine frozen (in indirect systems)	evaporator damage	check freeze protection, increase brine concentration
Defective expansion valve (leak in sensor system)	suction pressure gauge indicates a vacuum because valve is closed	replace valve
Clogged strainer or nozzle of expansion valve	extreme superheat	clean
Clogged filter-drier	bubbles in sight glass, extreme superheat, filter-drier cold	replace filter-drier
Shut-off valve on receiver throttles, pressure drop in liquid line	bubbles in sight glass, extreme superheat, cold line section	open valve, check line
Too little heat load	frequent low pressure cut-out, freeze protection thermostat cycles frequently	check capacity control

8.3.2 Suction pressure normal, discharge pressure too high

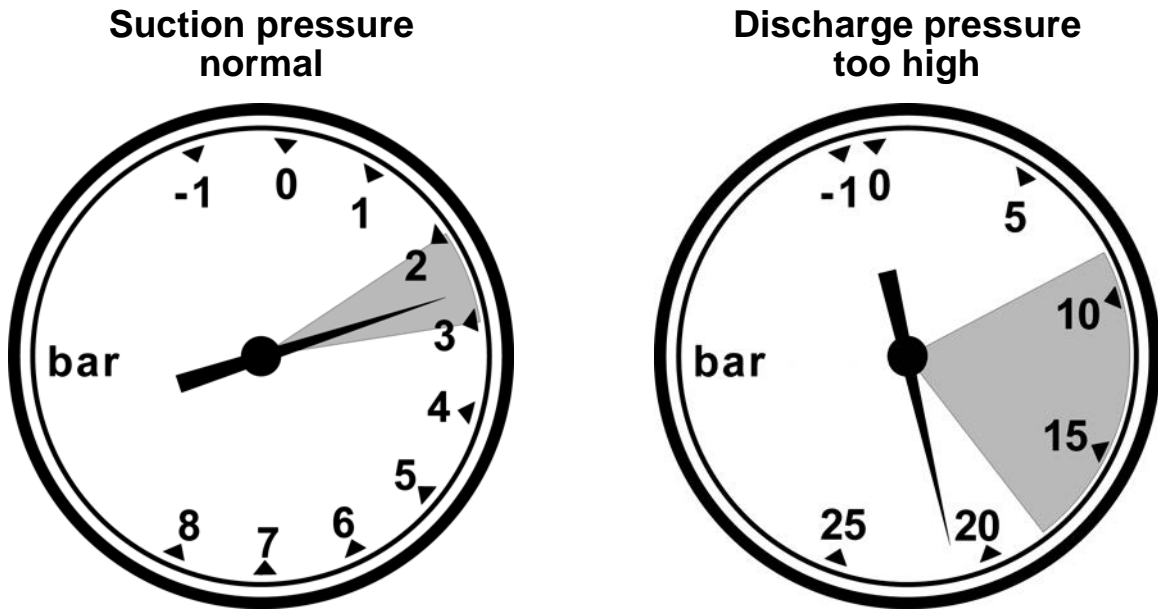


Fig. 8-6 Suction pressure normal, discharge pressure too high

Table 8-5. Possible faults at normal suction pressure and extremely high discharge pressure

Cause	Possible effect	Remedy
Condenser contaminated	high hot gas temperature, low cooling capacity	clean
Condenser fan failure	high hot gas temperature, high pressure cut-out	repair
Overcharge	high hot gas temperature, high subcooling, low cooling capacity	optimize refrigerant charge
Non-condensables (air)	high hot gas temperature, high measured subcooling, low cooling capacity	recharge
Throttling between compressor and condenser	high hot gas temperature, low cooling capacity	check lines and valves

8 Troubleshooting

8.3.3 Suction pressure too high, discharge pressure too low to normal

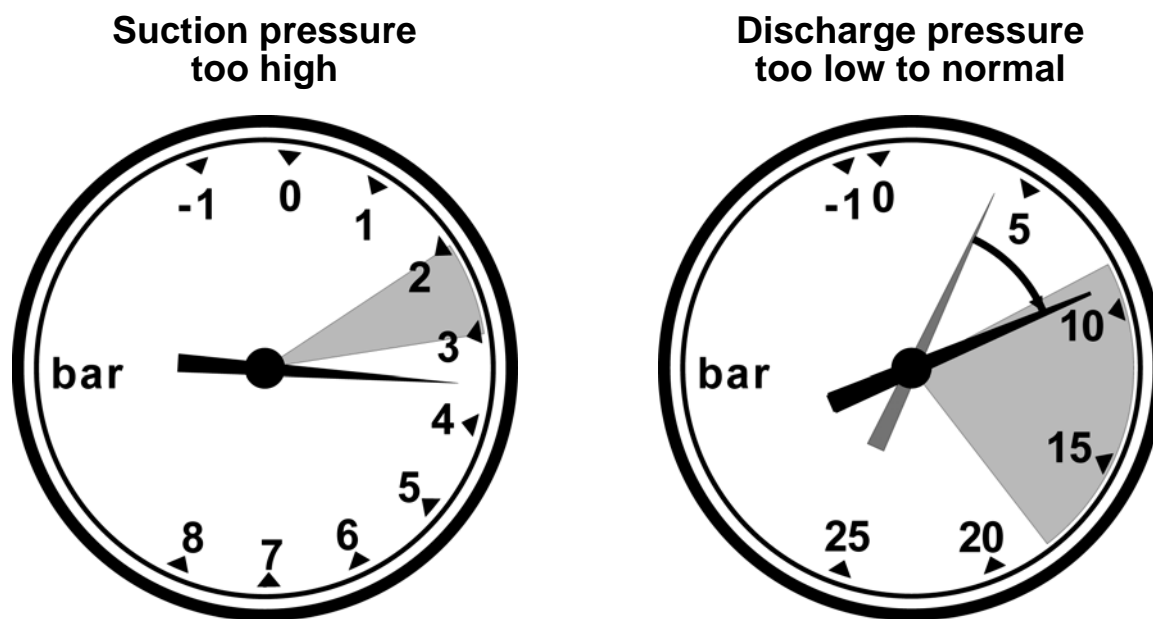


Fig. 8-7 Suction pressure too high, discharge pressure too low to normal

Table 8-6. Possible faults at extremely high suction pressure and extremely low to normal discharge pressure

Cause	Possible effect	Remedy
Defective compressor (valve plate, piston rings)	insufficient cooling capacity	replace compressor or defective parts
Defective capacity control	insufficient cooling capacity extremely high hot gas temperature with bypass open	repair capacity control

8.3.4 Suction pressure too high, discharge pressure too high

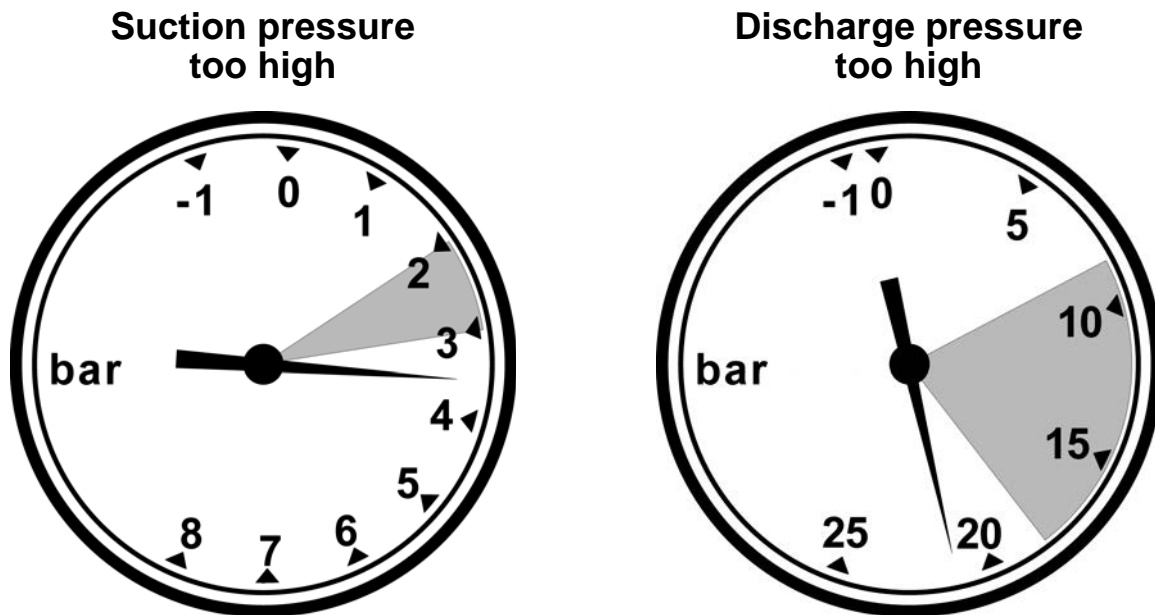


Fig. 8-8 Suction pressure too high, discharge pressure too high

Table 8-7. Possible faults at high suction pressure and high discharge pressure

Cause	Possible effect	Remedy
Sensor of thermostatic expansion valve loose, has poor contact or is not heat insulated	too little superheat, compressor draws in liquid	adjust sensor attachment

8 Troubleshooting

8.3.5 Other malfunctions

Table 8-8. Other Malfunctions

Symptom	Cause	Possible effect	Remedy
Hot gas temperature too high	oil shortage	undue compressor wear	top up oil
Compressor not running	pressure switch or other safety cut-out has tripped, electrical fault, cylinder filled with liquid refrigerant	plant stopped	check switching gear locate and eliminate cause of tripping
Compressor cycles continuously	switching difference too little, tripping of a cut-out device (overpressure switch, low pressure switch) shortage of refrigerant, defective fan, overcharge	compressor cycling, increased wear, insufficient cooling capacity	check switching devices, eliminate cause of cycling on/off
Extreme superheat	valve maladjusted, valve nozzle too small, valve sieve clogged, shortage of refrigerant, MOP reached (no fault)	low cooling capacity high hot gas temperatures	adjust valve, replace nozzle, install filter, clean, top up, leak detection
Insufficient superheat	valve maladjusted, valve nozzle too large, overcharge (capillaries), incorrect sensor attachment	compressor draws in liquid, compressor damage	adjust valve, replace nozzle, optimize refrigerant charge, correct sensor attachment

9 Safety Regulations

9.1 Handling of refrigerants

All work on refrigeration systems must be carried out in conformance with EN 378. Safety data sheets or substance data sheets (can be obtained from the manufacturer) and general notes of the trade association for the chemical industry are available for every refrigerant.

For the safe and proper use of refrigerants certain requirements apply that must be complied with:

- Always wear **safety glasses** when handling refrigerants. Refrigerant contact with your eyes can cause severe freeze burns. Immediately flush eyes with plenty of water and get medical attention.
 - Always wear **protective gloves** when handling refrigerants. Liquid refrigerant must not get into contact with your skin. Hands must be protected against frostbites (emerging R 134a vaporizes at - 26,5 °C) and removal of the protective layer of the skin (refrigerants dissolve fats)! In the event of skin contact immediately flush the areas affected with plenty of water and get medical attention.
- Leather and textile gloves are not suited (fluorelastomers are preferable).
- Danger of **asphyxiation** exists when refrigerants escape into the atmosphere. Refrigerants are heavier than air. In fact, starting with a concentration of approx. 12 % by volume in the air there is a lack of oxygen required for breathing. Unconsciousness and increased cardiovascular problems caused by stress and a lack of oxygen are the result. This is a deadly danger!
 - **Smoking is prohibited** when handling refrigerants. The burning ash of cigarettes can result in a decomposition of the refrigerant causing toxic substances to be produced.
 - Prior to the performance of **welding and soldering work on refrigeration systems** the refrigerant must be evacuated and any residues removed by purging with nitrogen.

Under the influence of heat decomposition products of the refrigerant are produced which are not only detrimental to your health but are also liable to cause corrosion.

A pungent smell that can be noticed during soldering is an indication that toxic decomposition products have formed. If these are inhaled damage to the respiratory passages, the lung and other organs may be caused.

- **A fire hazard** also exists in the case of non-flammable refrigerants through ignition of carried-over oil residues and insulation material as well as oil mist due to heavy leaks.

9.2 Handling of pressure containers

- **Cylinders must always be secured to prevent tipping or rolling**
- **Do not throw refrigerant cylinders.** When containers are dropped they may become distorted to such an extent that they rip open. Enormous forces will be released upon any sudden evaporation and escape of the refrigerant. The same applies when cylinder valves are broken off. Cylinders may therefore only be transported with the cover cap screwed on.
- **Refrigerant cylinders must not be stored in the vicinity of radiators.** Higher temperatures also result in higher pressures with the danger that the pressure permissible for the individual container may be exceeded. The pressure equipment directive therefore specifies that containers must not be subjected to temperatures in excess of 50 °C.
- **Never heat refrigerant cylinders with an open flame.** The material can be damaged and refrigerant decomposition may occur if cylinders are exposed to extremely high temperatures.
- **Keep valves closed on empty cylinders** to prevent any ingress of moisture.
- **Never overcharge refrigerant cylinders** as enormous pressures may build up when the temperature is increased.

9.3 Technical regulations for compressed gases (TRG)

Only those excerpts from regulations are listed below that are relevant to automobile manufacturers and workshops.

TRG 400 (Technical Regulations for Filling Stations in General)

2. Definitions and explanations
 - 2.1 Filling stations
 - 2.1.1 Filling stations are facilities for charging mobile compressed gas containers. The term filling station includes all workshops and facilities.
 - 2.4 Filling stations subject to approval

Filling stations subject to approval are filling stations where compressed gases are charged into mobile compressed gas containers subject to

9 Safety Regulations

being sold to third persons.

- 2.4 Filling stations not subject to approval
Filling stations not subject to approval are filling stations where compressed gases are charged into mobile compressed gas containers for exclusive use in one's own company.

TRG 402 (Operation of filling stations)

- 2 Employees and their training
- 2.1 Filling stations may only be operated and maintained by persons
1. having reached the age of 18
 2. having the required expertise
 3. from whom it can be expected that they fulfill their tasks reliably.
- 2.2 Dependent work may also be performed by persons not complying with the provisions of sect. 2.1 para. 1. and 2. above.
- 2.3 Prior to the start of their work and after that at regular intervals, but at least once a year, the employees shall be instructed with respect to the following:
1. special dangers involved in the handling of compressed gases,
 2. safety regulations, in particular this present technical regulation for compressed gases,
 3. measures to be taken in the event of malfunctions, damage and accidents,
 4. handling of fire extinguishing equipment and personal protective equipment,
 5. operation and maintenance of the filling station based on the operating manual.
- 5 Filling (excerpts therefrom)
- 5.1 A compressed gas container may only be charged with the compressed gas identified on it and only in the quantity as results from the data on the container with respect to pressure, weight and volume (see g15 para. 2 Pressure Equipment Directive).
- 5.2 With containers that may optionally be used for multiple compressed gases, the compressed gas to be charged and - in as far as the critical temperature of the compressed gas is above -10 °C - the maximum allowable charge weight in accordance with TRG 104 no. 3.3 must be identified on the container prior to filling.
- 5.3 Compressed gas containers on which the maximum allowable gauge pressure of the charge at a temperature of 15 °C is indicated in bar, must be charged based on the pressure (manometric). In

the event that upon charging a temperature other than 15 °C prevails, the pressure corresponding to that deviating temperature must be determined by the filling company; it must be ensured that the maximum allowable gauge pressure of the charge in the compressed gas container at 15 °C is not exceeded. To determine any overcharge, the charged containers should be subjected to a random pressure check.

- 5.4 In the case of compressed gas containers on which the maximum allowable filling quantity is stated as the net weight (charging weight, permissible weight of the charge) in kg, the filling charge must be checked by weighing. The scales for the weighing check must be calibrated.
- 5.7 Charging and verification measurements must not be conducted by the same person. Verification measurements must be carried out immediately after completion of the charging process.
- 5.8 Overcharged containers must immediately be safely discharged down to the permissible filling quantity. Thereafter, the charged compressed gas amount is to be newly determined.
- 5.9 Items 5.4 to 5.7 do not apply to containers for liquid deep-cold compressed gases that are neither flammable nor poisonous; regulations concerning traffic laws remain unaffected thereof.

9.4 Waste Act, Ordinance on the Determination of Residual Materials, Residual Material Supervision Ordinance

These guidelines are the legal basis for the official directive to keep records of the recycling of residual substances subject to supervision.

Waste Act (AbfG) § 2 para. 3

Delegated powers of the Federal Government to issue legal regulations for the determination of residual substances which are suspected to be impair the general public to a great extent.

Ordinance on the determination of residual materials (RestBestV) § 1

Residual materials within the context of this ordinance are residual substances within the meaning of art. 2 para. 3 AbfG. Prerequisite: The annual quantity of one or more of these substances is > 500 kg/a.

(AbfG) § 11 para. 2

Authorization of the authorities concerned with waste law to issue record keeping regulations. Details are subject to regulation by administrative order of the Federal Ministry.

Residual Material Supervision Ordinance

Directive governing the accountability for residual materials in accordance with art. 11 para. 2 AbfG.

Obligation to maintain recycling records, delivery notes and waste record books.

Prerequisite: The annual amount of one or more of these residual materials is > 500 kg/a.

Disposal of refrigerants and refrigeration oils

Refrigerants intended for disposal must be filled into properly labeled recycling containers taking into account the maximum allowable filling quantity.

Used **refrigeration oils** from installations using halogenated hydrocarbons must be disposed of as hazardous waste. A mixture with other oils or substances is not permissible. Proper storage and disposal must be in accordance with the directives applicable in the individual states.

9.5 Other standards and directives

EN 378 Parts 1-4

This previously 4-part standard (effective since September 2000) contains safety and environmentally relevant requirements superseding the formerly applicable standard DIN 8975. The additional parts 5 to 10 are available as drafts (Jan. 1994).

VBG 20

These regulations of the trade association (VBG) for foodstuffs and restaurants: "Refrigerating systems, heat pumps and cooling devices" contain just as DIN 8975 principles regarding the design, equipment and installation of refrigerating plants. It will be replaced in future by EN 378.

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10 Appendix

10.1 Symbols and Indices used

Table 10-1. Symbols used

No.	Symbol	Term	SI-Unit	Comment
1	A	area, surface	m ²	
2	c	specific heat capacity	J/(kg·K)	
3	F	force	N	
4	h	specific enthalpy	J/kg	
5	Δh	difference of specific enthalpy	J/kg	
6	k	heat transmission coefficient	W/(m ² ·K)	
7	l	length	m	
8	m	mass	kg	
9	\dot{m}	mass flow rate	kg/s	
10	P	power	W	
11	P _V	compressor driving power	W	
12	p	pressure	Pa	1 Pa = 10 ⁻⁵ bar
13	p _{abs}	absolute pressure	Pa	
14	p _{amb}	ambient pressure	Pa	
15	p _c	condensing pressure (absolute)	Pa	
16	p _{E1}	pressure at expansion valve inlet (absolute)	Pa	
17	p _{E2}	pressure at expansion valve outlet (absolute)	Pa	
18	p _e	difference between absolute pressure and ambient atmospheric pressure	Pa	p _e = p _{abs} - p _{amb}
19	p _{ec}	difference between condensing pressure and ambient atmospheric pressure	Pa	p _{ec} = p _c - p _{amb}
20	p _{eo}	difference between evaporating pressure and ambient atmospheric pressure	Pa	p _{eo} = p _o - p _{amb}
21	p _o	evaporating pressure (absolute)	Pa	
22	p _{o1}	pressure at evaporator inlet (absolute)	Pa	
23	p _{o2}	pressure at evaporator outlet (absolute)	Pa	
24	p _{V1}	pressure at compressor inlet (absolute)	Pa	
25	p _{V2}	pressure at compressor outlet (absolute)	Pa	
26	p _{zul}	permissible operating pressure	Pa	
27	q	heat of fusion	J/kg	
28	Q	heat	J	
29	\dot{Q}	heat flow, heat output	W	W = J/s

10 Appendix

Table 10-1. Symbols used

30	\dot{Q}_c	condenser performance, heat output	W	
31	\dot{Q}_o	refrigeration capacity, evaporator capacity	W	
32	r	evaporating heat	J/kg	
33	T	thermodynamic temperature	K	
34	t	temperature	°C	$t = T - 273,15 \text{ K}$
35	t_{amb}	ambient temperature	°C	
36	t_c	condensing temperature	°C	saturation temp. at p_c
37	t_{c1h}	temperature of superheated refrigerant at condenser inlet	°C	
38	t_{c2u}	temperature of subcooled refrigerant liquid at condenser outlet	°C	
39	t_{cL1}	air temperature at condenser inlet	°C	
40	t_{cL2}	air temperature at condenser outlet	°C	
41	t_{E1u}	temperature of subcooled refrigerant liquid at expansion valve inlet	°C	
42	t_o	evaporating temperature	°C	saturation temp. at p_o
43	t_{o2h}	temperature of superheated refrigerant at evaporator outlet	°C	
44	t_{oL1}	air temperature at evaporator inlet	°C	
45	t_{oL2}	air temperature at evaporator outlet	°C	
46	t_{V1h}	temperature of superheated refrigerant at compressor inlet	°C	
47	t_{V2h}	temperature of superheated refrigerant at compressor outlet	°C	
48	Δt	temperature difference	K	$\Delta t = \Delta T$
49	Δt_1	driving inlet temperature gradient	K	$= t_{c2} - t_{cL1}$
50	Δt_{c2u}	subcooling at condenser outlet	K	$= t_{c2} - t_{c2u}$
51	Δt_{E1u}	subcooling at expansion valve inlet	K	$= t_{E1} - t_{E1u}$
52	Δt_{o2h}	superheat at evaporator outlet	K	$= t_{o2h} - t_{o2}$
53	Δt_{V1h}	superheat at compressor inlet	K	$= t_{V1h} - t_{V1}$
54	V	volume	m^3	
55	V_g	geometric displacement of compressor	m^3	
56	\dot{V}	volume flow	m^3/s	
57	\dot{V}_g	geometric volume flow rate of compressor	m^3/s	
58	\dot{V}_{V1}	suction volume flow rate of compressor	m^3/s	
59	v	specific volume	m^3/kg	$v = 1/\rho$
60	x	vapor content	%	
61	Δ	difference (delta)	-	
62	ε	coefficient of performance (epsilon)	-	also COP

Table 10-1. Symbols used

63	λ	volumetric efficiency of compressor (lambda)	m^3/m^3	
64	π	pressure ratio of compressor (pi)	-	
65	ρ	density (rho)	kg/m^3	$\rho = 1/v$

Table 10-2. Indices used

No.	Index	Term	Comment
1	B	container	e.g. receiver
2	E	throttling device	e.g. expansion valve
3	K	cooling medium	
4	L	air	
5	M	motor, prime mover	
6	P	pump	
7	R	refrigerant	
8	V	compressor	
9	c	condenser	
10	h	superheated vapor	
11	o	evaporator	spoken: zero
12	u	subcooled liquid	
13	1	inlet	
14	2	outlet	
15	'	applicable to boiling liquid	state on boiling line
16	''	applicable to saturated vapor	state on dew point line

10 Appendix

10.2 Measuring point designations

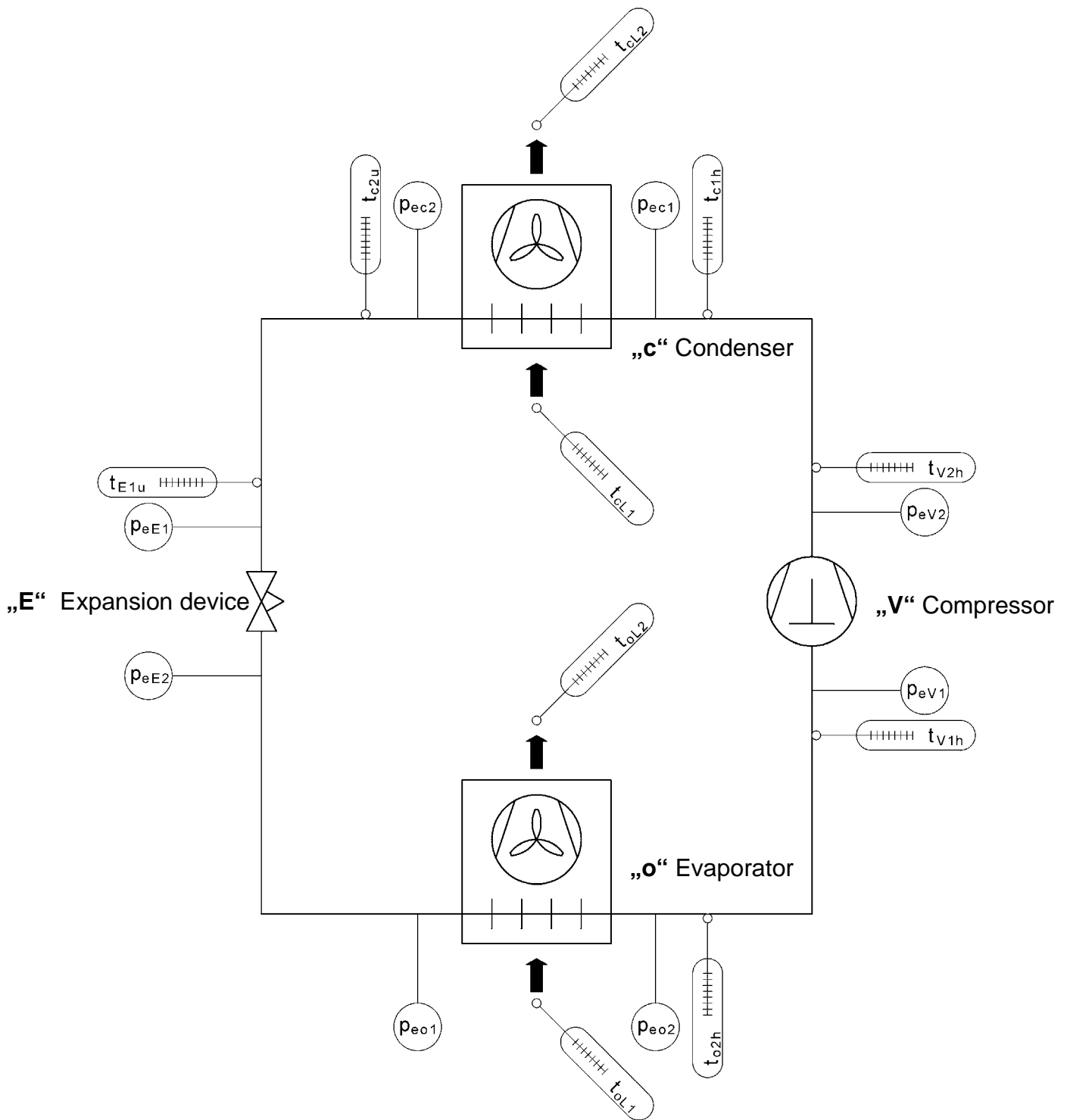
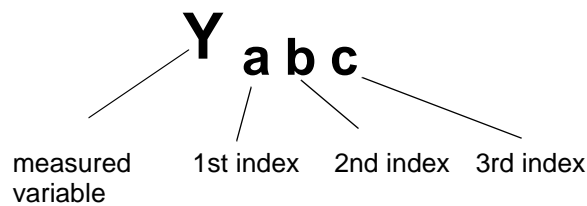


Fig. 10-1 Measuring points for pressure and temperature

The measuring point designations are represented in encoded form on the page below.

Codes of measuring points**Measured variable Y:**

Reference to the measured variable or state variable

Measured variable: p = pressure t = temperature

State variable: h = specific enthalpy

1. Index Y_a..: Indicates the component where the measured variable is measured.

Examples: V = compressor E = throttling device
o = evaporator c = condenser

Note: If a **gauge pressure measuring point** (gauge pressure reading) is indicated, the first index is an **e**. The designation of the component is shifted to the 2nd index. The **absolute pressure** is not identified. In the case of pressure measuring points **without e** as the 1st index an absolute pressure is to be assumed.

2. Index Y_.b.: Indicates whether the measured variable is measured at the inlet or outlet of the designated component.

Examples: 1 = inlet 2 = outlet

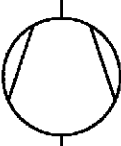

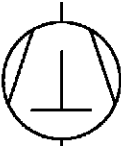

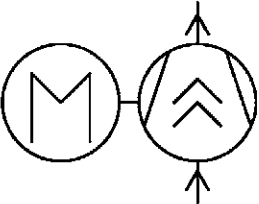
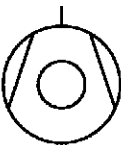
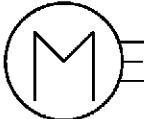
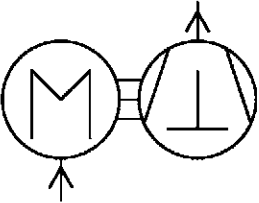

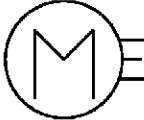
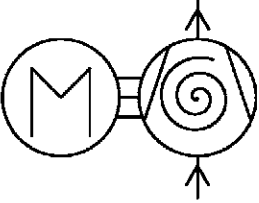

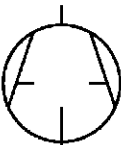
Note: In the case of gauge pressure measuring points it is the component that is identified here. Inlet or outlet is shifted to the 3rd index.

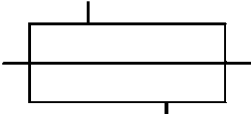

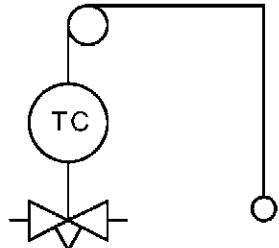
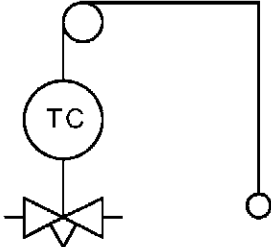
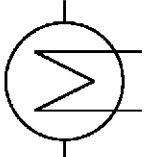
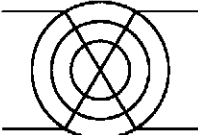
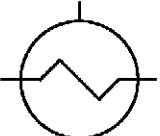
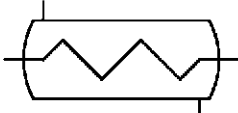
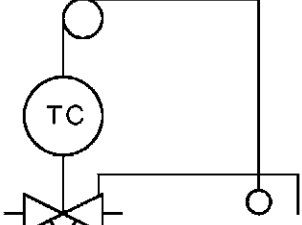
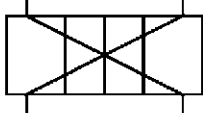
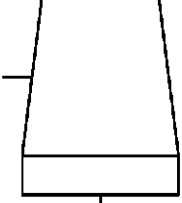
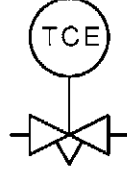
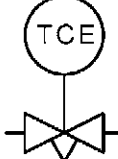
3. Index Y_..c: This index exists only at thermometric temperature measuring points - with the exceptions referred to in the notes under Index 1 and 2. Manometric temperatures (temperature derived from a saturation pressure) do not have this index. At the same time, this index clarifies the state of the refrigerant.



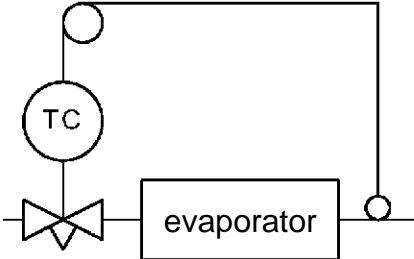


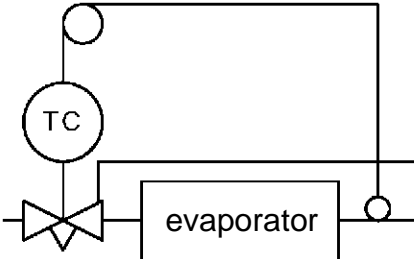

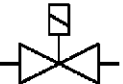

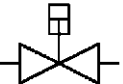




Examples: h = superheated u = subcooled or liquid


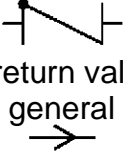

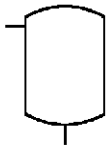
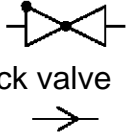

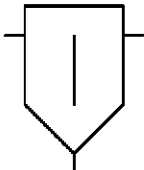


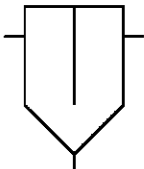

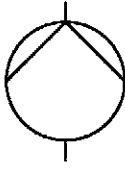
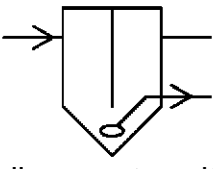
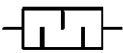
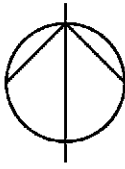
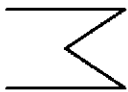
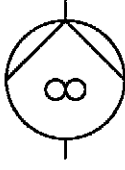
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10.3 Graphical symbols (EN 1861, April 1998)

Compressor	Motors	Example
 <p data-bbox="197 501 456 591">compressor compressor pump in general</p>	 <p data-bbox="536 501 732 557">electric motor in general</p>	
 <p data-bbox="233 808 419 864">reciprocating compressor</p>	 <p data-bbox="536 804 735 875">electric motor open</p>	 <p data-bbox="940 842 1203 904">open screw compressor</p>
 <p data-bbox="240 1122 411 1178">turbo compressor</p>	 <p data-bbox="536 1155 735 1234">electric motor suction-vapor cooled</p>	 <p data-bbox="911 1155 1278 1234">hermetic/semi-hermetic suction-vapor cooled reciprocating compressor</p>
 <p data-bbox="236 1435 411 1491">screw compressor</p>	 <p data-bbox="517 1435 732 1514">electric motor hermetic/ semi-hermetic</p>	 <p data-bbox="919 1469 1262 1525">hermetic/semi-hermetic scroll compressor</p>
 <p data-bbox="236 1749 411 1805">scroll compressor</p>		
 <p data-bbox="213 2051 419 2107">rotating piston compressor</p>		

Heat exchanger		Expansion valves
 <p>double tube heat exchanger</p>		
 <p>tubular heat exchanger</p>	 <p>air-cooled finned tube heat exchanger</p>	 <p>thermostatic expansion valve with internal pressure equalization</p>
 <p>heat exchanger without crossing of flow lines</p>	 <p>spiral heat exchanger</p>	
 <p>heat exchanger with crossing of flow lines</p>	 <p>heat exchanger with tube coil</p>	 <p>thermostatic expansion valve with external pressure equalization</p>
 <p>plate heat exchanger</p>	 <p>cooling tower in general</p>	 <p>pressure-controlled expansion valve</p>
		 <p>electronic expansion valve</p>

Shut-off valves	Actuators	Example: Expansion device
 valve, in general	 drive, in general with auxiliary power or automatic	 thermostatic expansion valve with internal pressure equalization
 valve, corner type, in general	 drive, hand-operated	 thermostatic expansion valve with external pressure equalization
 three-way valve, in general	 electromagnetic drive	
 slide valve	 piston actuator	
 ball valve	 diaphragm actuator	
 safety valve, corner type, spring-loaded	 electric motor actuator	

Other Components		Fans/Pumps
 filter-drier	 non-return valves, in general	 fan, in general
 container with dished head	 check valve	 axial fan
 separator, in general	 sight glass	 radial fan
 baffle separator	 sight glass with moisture indicator	 pump, in general
 oil separator with float	 silencer	 centrifugal pump
	 device for heating or cooling, in general	 gear pump

10 Appendix

10.4 Vapor chart of R 134a

Table 10-3. Physical characteristics of R 134a [calculation program Solkane Refrigerants Version 2.0]

Temperature t °C	Pressure p bar	Density		Spec. Volume		Spec. Enthalpy		Evaporating heat r kJ/kg
		of liquid ρ' kg/dm ³	of vapor ρ'' kg/m ³	of liquid v' dm ³ /kg	of vapor v'' dm ³ /kg	of liquid h' kJ/kg	of vapor h'' kJ/kg	
-60	0.16	1.469	0.92	0.681	1082.60	123.17	359.85	236.68
-55	0.22	1.456	1.24	0.687	803.83	129.49	362.99	233.50
-50	0.29	1.443	1.65	0.693	606.45	135.75	366.14	230.39
-45	0.39	1.429	2.15	0.700	464.29	141.98	369.28	227.30
-40	0.51	1.415	2.78	0.707	360.30	148.21	372.41	224.19
-35	0.66	1.401	3.53	0.714	283.08	154.47	375.52	221.05
-30	0.84	1.387	4.45	0.721	224.97	160.78	378.61	217.84
-25	1.06	1.373	5.53	0.728	180.67	167.14	381.68	214.55
-24	1.11	1.370	5.78	0.730	173.12	168.42	382.29	213.88
-23	1.16	1.367	6.03	0.732	165.95	169.70	382.90	213.20
-22	1.22	1.364	6.28	0.733	159.14	170.98	383.51	212.53
-21	1.27	1.361	6.55	0.735	152.66	172.27	384.12	211.85
-20	1.33	1.358	6.83	0.736	146.50	173.56	384.72	211.16
-19	1.39	1.355	7.11	0.738	140.64	174.86	385.33	210.47
-18	1.45	1.352	7.40	0.740	135.06	176.15	385.93	209.78
-17	1.51	1.349	7.71	0.741	129.75	177.45	386.53	209.08
-16	1.57	1.346	8.02	0.743	124.68	178.75	387.13	208.38
-15	1.64	1.343	8.34	0.745	119.86	180.06	387.73	207.67
-14	1.71	1.340	8.68	0.746	115.26	181.37	388.33	206.96
-13	1.78	1.337	9.02	0.748	110.87	182.68	388.93	206.25
-12	1.85	1.334	9.37	0.750	106.68	183.99	389.52	205.53
-11	1.93	1.331	9.74	0.752	102.68	185.31	390.11	204.80
-10	2.01	1.328	10.12	0.753	98.86	186.63	390.71	204.07
-9	2.09	1.324	10.50	0.755	95.21	187.96	391.30	203.34
-8	2.17	1.321	10.90	0.757	91.73	189.28	391.88	202.60
-7	2.26	1.318	11.31	0.759	88.39	190.61	392.47	201.86
-6	2.34	1.315	11.74	0.760	85.21	191.94	393.06	201.11
-5	2.43	1.312	12.17	0.762	82.16	193.28	393.64	200.36
-4	2.53	1.309	12.62	0.764	79.24	194.62	394.22	199.60

Table 10-3. Physical characteristics of R 134a [calculation program Solkane Refrigerants Version 2.0]

Temperature t °C	Pressure p bar	Density		Spec. Volume		Spec. Enthalpy		Evaporating heat r kJ/kg
		of liquid ρ' kg/dm ³	of vapor ρ'' kg/m ³	of liquid v' dm ³ /kg	of vapor v'' dm ³ /kg	of liquid h' kJ/kg	of vapor h'' kJ/kg	
-3	2.62	1.305	13.08	0.766	76.44	195.96	394.80	198.84
-2	2.72	1.302	13.56	0.768	73.77	197.30	395.38	198.08
-1	2.82	1.299	14.04	0.770	71.20	198.65	395.96	197.31
0	2.93	1.296	14.55	0.772	68.75	200.00	396.53	196.53
1	3.04	1.292	15.06	0.774	66.39	201.35	397.10	195.75
2	3.15	1.289	15.59	0.776	64.13	202.71	397.68	194.97
3	3.26	1.286	16.14	0.778	61.96	204.07	398.24	194.18
4	3.38	1.282	16.70	0.780	59.88	205.43	398.81	193.38
5	3.50	1.279	17.27	0.782	57.89	206.79	399.38	192.58
6	3.62	1.276	17.87	0.784	55.97	208.16	399.94	191.78
7	3.75	1.272	18.47	0.786	54.13	209.53	400.50	190.97
8	3.88	1.269	19.10	0.788	52.36	210.90	401.06	190.16
9	4.01	1.265	19.74	0.790	50.66	212.28	401.62	189.34
10	4.15	1.262	20.40	0.792	49.03	213.66	402.17	188.51
11	4.29	1.259	21.07	0.795	47.45	215.04	402.72	187.68
12	4.43	1.255	21.77	0.797	45.94	216.42	403.27	186.85
13	4.58	1.252	22.48	0.799	44.49	217.81	403.82	186.01
14	4.73	1.248	23.21	0.801	43.08	219.20	404.36	185.17
15	4.89	1.245	23.96	0.804	41.74	220.59	404.91	184.32
16	5.04	1.241	24.73	0.806	40.44	221.99	405.45	183.46
17	5.21	1.237	25.52	0.808	39.19	223.38	405.98	182.60
18	5.37	1.234	26.33	0.811	37.98	224.78	406.52	181.73
19	5.54	1.230	27.16	0.813	36.82	226.19	407.05	180.86
20	5.72	1.226	28.01	0.815	35.70	227.60	407.58	179.98
21	5.90	1.223	28.89	0.818	34.62	229.01	408.11	179.10
22	6.08	1.219	29.78	0.820	33.58	230.42	408.63	178.21
23	6.27	1.215	30.70	0.823	32.57	231.83	409.15	177.32
24	6.46	1.212	31.65	0.825	31.60	233.25	409.67	176.41
25	6.66	1.208	32.61	0.828	30.66	234.68	410.18	175.51
26	6.86	1.204	33.60	0.831	29.76	236.10	410.69	174.59
27	7.06	1.200	34.62	0.833	28.88	237.53	411.20	173.67

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Table 10-3. Physical characteristics of R 134a [calculation program Solkane Refrigerants Version 2.0]

Temperature t °C	Pressure p bar	Density		Spec. Volume		Spec. Enthalpy		Evaporating heat r kJ/kg
		of liquid ρ' kg/dm ³	of vapor ρ'' kg/m ³	of liquid v' dm ³ /kg	of vapor v'' dm ³ /kg	of liquid h' kJ/kg	of vapor h'' kJ/kg	
28	7.27	1.196	35.66	0.836	28.04	238.96	411.71	172.75
29	7.48	1.192	36.73	0.839	27.22	240.40	412.21	171.81
30	7.70	1.189	37.83	0.841	26.44	241.83	412.71	170.87
31	7.93	1.185	38.95	0.844	25.67	243.28	413.20	169.92
32	8.16	1.181	40.10	0.847	24.94	244.72	413.69	168.97
33	8.39	1.177	41.28	0.850	24.22	246.17	414.18	168.00
34	8.63	1.173	42.49	0.853	23.53	247.63	414.66	167.03
35	8.87	1.169	43.74	0.856	22.86	249.08	415.14	166.05
36	9.12	1.164	45.01	0.859	22.22	250.55	415.61	165.07
37	9.37	1.160	46.31	0.862	21.59	252.01	416.08	164.07
38	9.63	1.156	47.65	0.865	20.99	253.48	416.55	163.07
39	9.90	1.152	49.02	0.868	20.40	254.96	417.01	162.05
40	10.17	1.148	50.43	0.871	19.83	256.44	417.47	161.03
41	10.44	1.143	51.87	0.875	19.28	257.92	417.92	160.00
42	10.72	1.139	53.35	0.878	18.74	259.41	418.37	158.96
43	11.01	1.135	54.87	0.881	18.23	260.90	418.81	157.90
44	11.30	1.130	56.42	0.885	17.72	262.40	419.24	156.84
45	11.60	1.126	58.02	0.888	17.24	263.91	419.68	155.77
46	11.90	1.121	59.65	0.892	16.76	265.42	420.10	154.68
47	12.21	1.117	61.33	0.895	16.30	266.94	420.52	153.58
48	12.53	1.112	63.06	0.899	15.86	268.46	420.93	152.47
49	12.85	1.108	64.82	0.903	15.43	269.99	421.34	151.35
50	13.18	1.103	66.64	0.907	15.01	271.53	421.74	150.21
51	13.51	1.098	68.50	0.910	14.60	273.07	422.14	149.06
52	13.85	1.094	70.41	0.914	14.20	274.62	422.52	147.90
53	14.20	1.089	72.38	0.919	13.82	276.18	422.90	146.72
54	14.55	1.084	74.39	0.923	13.44	277.75	423.28	145.53
55	14.91	1.079	76.47	0.927	13.08	279.32	423.64	144.32
56	15.28	1.074	78.59	0.931	12.72	280.91	424.00	143.09
57	15.66	1.069	80.78	0.936	12.38	282.50	424.35	141.85
58	16.04	1.064	83.03	0.940	12.04	284.10	424.69	140.58

Table 10-3. Physical characteristics of R 134a [calculation program Solkane Refrigerants Version 2.0]

Temperature t °C	Pressure p bar	Density		Spec. Volume		Spec. Enthalpy		Evaporating heat r kJ/kg
		of liquid ρ' kg/dm ³	of vapor ρ'' kg/m ³	of liquid v' dm ³ /kg	of vapor v'' dm ³ /kg	of liquid h' kJ/kg	of vapor h'' kJ/kg	
59	16.42	1.059	85.34	0.945	11.72	285.71	425.02	139.30
60	16.82	1.053	87.72	0.949	11.40	287.34	425.34	138.00
61	17.22	1.048	90.16	0.954	11.09	288.97	425.65	136.68
62	17.63	1.043	92.68	0.959	10.79	290.61	425.95	135.34
63	18.04	1.037	95.27	0.964	10.50	292.26	426.24	133.97
64	18.47	1.032	97.94	0.969	10.21	293.93	426.51	132.58
65	18.90	1.026	100.69	0.975	9.93	295.61	426.78	131.17
66	19.34	1.020	103.53	0.980	9.66	297.30	427.03	129.74
67	19.78	1.014	106.45	0.986	9.39	299.00	427.27	128.27
68	20.24	1.009	109.46	0.992	9.14	300.72	427.50	126.78
69	20.70	1.003	112.58	0.997	8.88	302.45	427.71	125.26
70	21.17	0.996	115.79	1.004	8.64	304.19	427.90	123.71
71	21.65	0.990	119.11	1.010	8.40	305.95	428.08	122.13
72	22.14	0.984	122.55	1.016	8.16	307.73	428.25	120.52
73	22.63	0.977	126.11	1.023	7.93	309.52	428.39	118.87
74	23.13	0.971	129.79	1.030	7.70	311.33	428.51	117.19
75	23.65	0.964	133.60	1.037	7.48	313.15	428.62	115.46
76	24.17	0.957	137.56	1.045	7.27	315.00	428.70	113.70
77	24.70	0.950	141.67	1.052	7.06	316.86	428.76	111.90
78	25.23	0.943	145.95	1.060	6.85	318.74	428.79	110.05
79	25.78	0.936	150.40	1.069	6.65	320.64	428.80	108.15
80	26.34	0.928	155.03	1.078	6.45	322.57	428.77	106.21
81	26.90	0.920	159.87	1.087	6.26	324.51	428.72	104.21
82	27.48	0.912	164.92	1.096	6.06	326.48	428.63	102.15
83	28.06	0.904	170.22	1.106	5.87	328.47	428.50	100.04
84	28.66	0.895	175.77	1.117	5.69	330.48	428.34	97.86
85	29.26	0.887	181.60	1.128	5.51	332.52	428.12	95.61
86	29.88	0.878	187.75	1.140	5.33	334.58	427.86	93.29
87	30.51	0.868	194.25	1.152	5.15	336.67	427.55	90.88
88	31.14	0.858	201.14	1.165	4.97	338.78	427.17	88.39
89	31.79	0.848	208.48	1.179	4.80	340.92	426.72	85.80

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Table 10-3. Physical characteristics of R 134a [calculation program Solkane Refrigerants Version 2.0]

Temperature t °C	Pressure p bar	Density		Spec. Volume		Spec. Enthalpy		Evaporating heat r kJ/kg
		of liquid ρ' kg/dm ³	of vapor ρ'' kg/m ³	of liquid v' dm ³ /kg	of vapor v'' dm ³ /kg	of liquid h' kJ/kg	of vapor h'' kJ/kg	
90	32.45	0.837	216.31	1.195	4.62	343.09	426.20	83.10
91	33.12	0.826	224.73	1.211	4.45	345.29	425.58	80.29
92	33.80	0.813	233.84	1.229	4.28	347.53	424.86	77.33
93	34.49	0.801	243.77	1.249	4.10	349.79	424.00	74.21
94	35.19	0.787	254.71	1.271	3.93	352.08	422.99	70.91
95	35.91	0.771	266.92	1.296	3.75	354.41	421.78	67.37
96	36.64	0.755	280.84	1.325	3.56	356.77	420.30	63.53
97	37.39	0.736	297.17	1.359	3.37	359.17	418.44	59.27
98	38.14	0.714	317.43	1.400	3.15	361.60	415.97	54.37
99	38.92	0.687	346.24	1.455	2.89	364.07	412.16	48.09
100	39.71	0.650	636.32	1.537	1.57	366.58	375.04	8.46
101.06	40.56	0.515	515.30	1.941	1.94	390.05	390.05	0.00

