Technical handbook for Refrigerant Applications

A DOVER COMPANY

TECHNICAL HANDBOOK FOR
REFRIGERANT APPLICATIONS

Contents

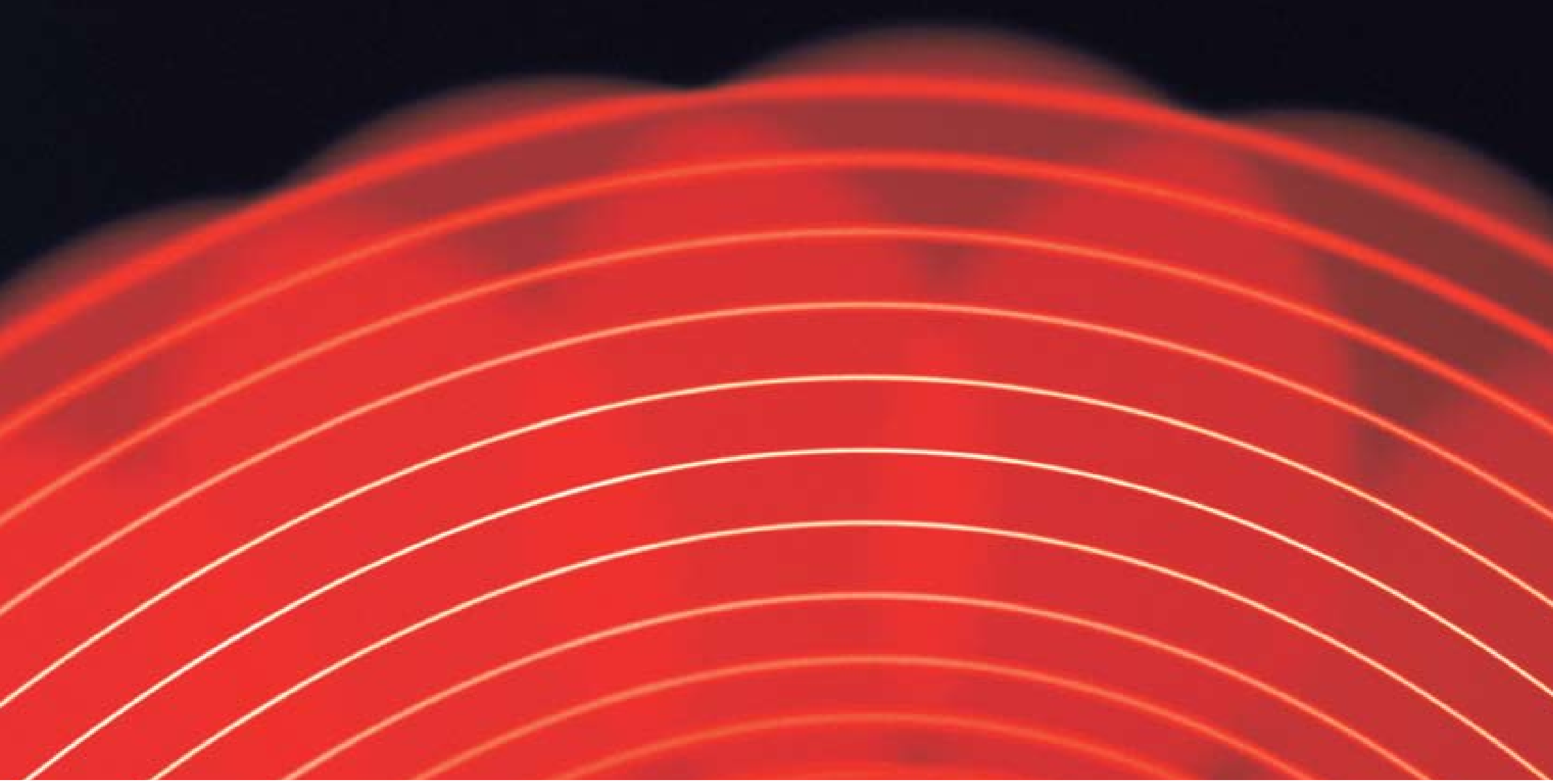
Introduction

Cooling food in supermarkets or cooling buildings with air-conditioners, reusing waste heat in desuperheaters and heating houses with heat pumps – all examples of the largest application field for SWEP CBEs, i.e. Refrigerant applications.

The theory behind refrigeration phenomena may seem deep and complicated. The purpose of the refrigerant applications handbook is to make this theory accessible and easier to understand. The handbook is a powerful tool for learning more about the background to heat transfer. You can read the handbook from start to finish, or go directly to chapters of interest.

This refrigerant applications handbook is also available in a complementary CD-ROM version. The basic contents are the same, but the CD-ROM version also includes useful online solutions such as versatile Search and Print functions.

Finally, we look forward to receiving your feedback on this material. All your thoughts, comments and ideas are highly appreciated. Please forward them to the Application Management Department at SWEP International, Sweden.



1 Basic Heat Transfer

The most basic rule of heat transfer is that heat always flows from a warmer medium to a colder medium. Heat exchangers are devices to facilitate this heat transfer with the highest possible efficiency. A good heat exchanger is able to transfer energy (heat) from the hot side to the cold side with small thermal losses and high efficiency. Good heat exchangers are typically small vessels with a small pressure drop, and are able to handle small temperature differences between two media.

1.1 What is Heat Transfer?

The heat transfer from one medium to another is controlled by a few simple but fundamental rules:

- There must be a temperature difference in order for energy transfer to take place.
- Energy (heat) will always flow from the warmer medium towards the colder medium.
- The energy (heat) rejected from the warm medium is equal to the heat absorbed by the cold medium plus losses to the surroundings.

1.2 One-Phase Heat Exchange

In a one-phase heat transfer process, there is no phase change in the media. An example of such a process is a water-to-water application where the water flow on side 1 changes temperature from 30 to 50°C and the flow on side 2 changes temperature from 65 to 45°C, i.e. without a phase change. The most commonly occurring one-phase applications for CBEs are water-to-water and oil-to-water. The purpose of the CBE in oil applications is to cool engine oils, hydraulic oils or similar with water.

1.3 Two-Phase Heat Exchange

In a two-phase heat exchange process there is a phase change on the cold side, the warm side or both. What happens when a liquid or a gas changes phase is described below.

If heat is added to a liquid, the temperature of the liquid will rise until it reaches its **boiling point**. Adding further heat will not raise the temperature. Instead, it increases the gas content of the liquid, resulting in a two-phase mixture of liquid and gas. The gas generated forms bubbles during the boiling process. The temperature will not rise until all liquid has been vaporized. When the temperature of the gas becomes higher than the boiling point, the gas is described as superheated. This process is typical of what happens inside an evaporator in a cooling system. The refrigerant enters the evaporator as liquid and leaves as **superheated** gas.

The opposite occurs in a condenser. First, superheated gas is cooled until it reaches its **saturation point**, where liquid droplets are formed. When all the gas has been transformed to liquid, the **bubble point** is reached. Maintaining the same pressure in the vessel while further cooling the liquid leads to a lower temperature, the result being described as a **sub-cooled liquid**.

The heat added or lost when the temperature changes within a phase is called **sensible heat,** while the heat added or lost in a phase-change is called **latent heat.** The latent heat of the phase transition between liquid and gas is many times higher than that of the liquid phase. The latent heat that must be added to transform water (100°C, 1 atm) to steam (100°C, 1 atm) is 2257 kJ/kg, while the sensible heat added in transforming water (0°C, 1 atm) to water (100°C, 1 atm) is only 419 kJ/kg. Figure 1.2 shows what happens when water in different states is mixed.

The water example illustrates that the reason for the commercial utiliza-

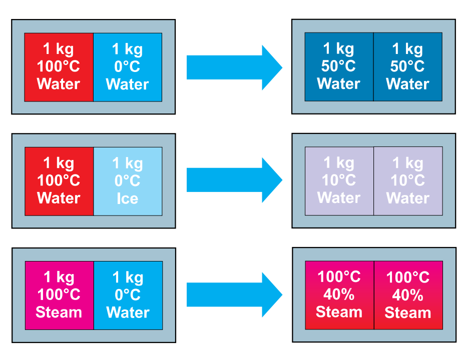


Figure 1.2 Impact of sensible and latent heat.

tion of evaporation and condensation is to gain or lose, respectively, the large amount of latent heat involved in the phase-transition between liquid and gas.

1.4 Flow Regimes

The ability to transfer heat in liquid and gaseous media depends on the turbulence of the medium. High turbulence is desirable for efficient transfer. Near a plane wall, there is always a film of laminar flow of gas or liquid, depending on the type of medium present (cf. chapter 1.5). The heat transfer in a laminar flow film is poor, because convectional heat transfer is more or less non-existent. However, if higher turbulence can be achieved, the insulating film becomes thinner, convection increases and, consequently, heat can be transported more efficiently, as shown in Figure 1.3.

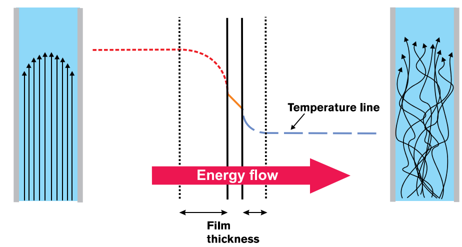


Figure 1.3 Temperature profile with laminar and turbulent flow on each side of the wall, respectively.

The Reynolds (Re) number is a dimensionless number used to describe the "state" of a fluid (turbulent or laminar flow). The Reynolds number is defined as in equation 1:

(1) $$Re = \frac{v \cdot X}{u}$$

Where:

$u = \eta / p$

η = Dynamic viscosity [$C_p = kg/m \cdot s$]

u = Kinematic viscosity [$St = cm^2/s$]

p = Fluid density [kg/m^3]

v = Fluid velocity [m/s]

X = Hydraulic diameter [m]

(For SWEP's heat exchangers, the hydraulic diameter is approximated as twice the pressing depth for the CBE.)

High turbulence is achieved by increasing the disturbances in a flow. A rough surface thus results in a more turbulent flow than a plane surface.

The flow inside a CBE is much more turbulent than the flow inside a Shell & Tube (S&T) heat exchanger, for example. This is because the plates of the CBE are rough and folded due to the herringbone pattern (see Figure 1.13). By contrast, in an S&T the fluids flow through flat pipes. Full turbulence is reached at approximately Re = 2300 in a tube but at an Re as low as approximately 150 in a CBE, which indicates that a smaller flow velocity is needed in a CBE than in an S&T.

For a proper comparison between the Reynolds numbers of different passages, their geometries should be exactly the same. This is of course not the case for a comparison between a CBE channel and an S&T channel, which have different hydraulic diameters. However, the practical fact remains: inside a CBE, a lower flow velocity is required to achieve fully turbulent flow.

In addition to the herringbone pattern on the plates, other parameters that lead to a high turbulence include:

- Small cross-sectional channel area
- Low fluid viscosity

With a constant pump power, a smaller cross-sectional channel area leads to a higher velocity and better heat transfer. For CBEs, a smaller cross-sectional area is obtained if a narrower plate is chosen, while lower velocity is obtained when more plates are added. The "cost" of high velocity and small cross-sectional area is an increased pressure drop through the heat exchanger. When designing a heat exchanger, e.g. in SWEP Software Package (SSP), the heat exchanger with the calculated pressure drop closest to the maximum allowed should be selected, in order to achieve maximum efficiency.

Viscosity is also an important factor when discussing flow regimes. For example, oil has a higher viscosity than water, and it is therefore more difficult to achieve turbulence in oil flows. A medium with a low viscosity might therefore be more useful as a heat conductor.

1.5 Energy Balance

The energy flow ($\dot{Q}$) goes from the warm medium to the cold medium through the heat transfer area (A) of the CBE. In addition to the size of the heat transfer area, the amount of energy transported also depends on the heat transfer coefficient (k) and the temperature difference between the two sides (dT). This relation is described in the heat transfer equation (eq. 2):

$$\dot{Q} = k \cdot A \cdot dT \qquad (2)$$

Where:
$\dot{Q}$ = Energy flow [W]
k = Overall heat transfer coefficient [W/(m²·°C)]
A = Heat transfer area [m²]
dT = Temperature difference [°C]

Area (A)

Increasing the area of a heat exchanger implies that more energy can be transferred. For CBEs, a larger area can be achieved by increasing the size and/or number of plates, which means more stainless steel and copper brazing. Hence, increasing the surface area implies higher costs.

Heat Transfer Coefficient (k)

Energy may be transported from a hot fluid to a colder fluid in three ways:

1. **Conduction** – The heat is conducted through solid material or a stationary liquid. In the stainless steel walls of a heat exchanger and in laminar flow (slow moving) regions, heat is transported only by conduction. The conductivity varies with the physical properties of the medium.

2. **Convection** – Movements of the fluid itself also transport energy. Turbulently flowing media and boiling/condensing fluids are very agitated, and will therefore transport energy mostly by convection.

3. **Radiation** – For very hot surfaces (T > 1000°C), electromagnetic radiation will become the most important means of heat transport. Radiation does not contribute significantly to the heat transfer in CBEs, due to their considerably lower working temperature.

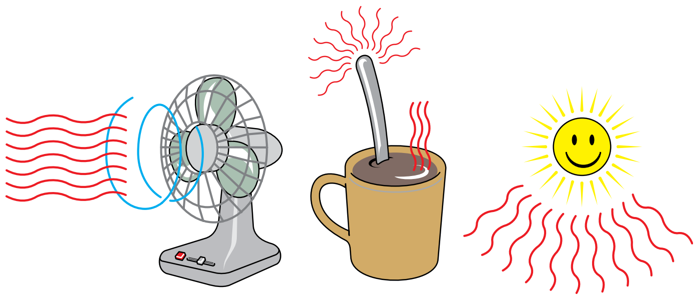


Figure 1.4 Convection, conduction and radiation.

In CBEs, the energy is therefore transferred through conduction and convection, and examples of these types of energy transport are shown in Figure 1.4.

The space between the dotted lines and the wall in Figure 1.5 is often called the film thickness. The heat transfer rate within the film is significantly lower than in the bulk liquid, because the temperature gradient decreases dramatically in this area (see Figure 1.5). The reason for the poorer heat transfer is the laminar flow that is always obtained near a plane wall. Laminar flow does not transfer energy as well as turbulent flow.

The overall heat transfer coefficient (k) describes the total effect of conduction and convection on the energy transfer:

(3) $$\frac{1}{k} = \frac{1}{\alpha_{HOT}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{COLD}}$$

Where:
k = Overall heat transfer coefficient [W/(m²/°C)]
δ = Plate thickness [m]
λ = Thermal conductivity [W/(m·°C)]
$\alpha_{HOT/COLD}$ = Film coefficient for hot/coldside [W/(m²/°C)]

The essence of equation 3 is that a high film coefficient and thermal conductivity and a thin plate lead to a high k-value. Thermal conductivity is a material-specific constant, and the film coefficient is a measure of how well heat is transferred by a specific fluid. For turbulent flows, α is always higher than for laminar flows.

With a higher overall heat transfer coefficient (k), more energy can be transferred per heat transfer area. Because this leads to a more cost-effective heat exchanger, it is very important to improve the k-value by all means possible.

Temperature Difference (dT)

The temperature difference between the hot and cold media is the driving force in energy transfer. A large temperature difference means that a smaller heat transfer area and/or a smaller heat transfer coefficient may be used to achieve the same energy transfer. It is therefore important to try to maximize the temperature difference between the hot and cold sides.

Figure 1.6 shows a single-phase temperature profile through a CBE.

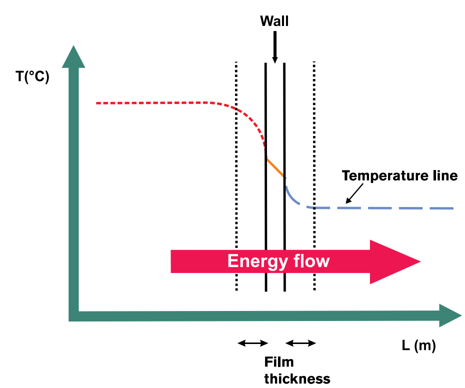


Figure 1.5 Energy transfer to a plane wall.

Because the temperature difference between the hot and cold sides varies through the heat exchanger, the logarithmic mean temperature difference (**LMTD**) is used. The definition of LMTD is shown in equation 4:

$$(4) \qquad LMTD = \frac{\Delta_A - \Delta_B}{\ln \frac{\Delta_A}{\Delta_B}}$$

Where:
Δ_A = Temp. diff. between the warm inlet and the cold outlet
Δ_B = Temp. diff. between the warm outlet and the cold inlet

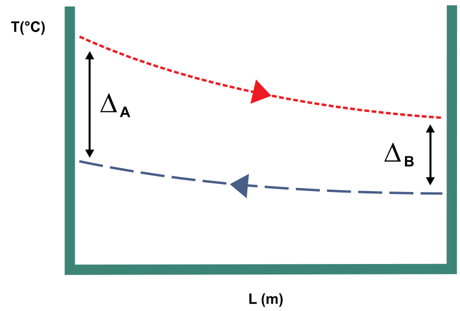


Figure 1.6 The temperature program for calculation of LMTD.

Please note that the logarithmic mean temperature difference (LMTD) may be used only for single-phase calculations (see chapter 6.10).

Preservation of Energy

The energy in a liquid flow can be described with the following formula:

$$(5) \qquad \dot{Q} = \dot{m} \cdot C_p \cdot \Delta T$$

Where:
$\dot{Q}$ = Energy flow [W]
$\dot{m}$ = Mass flow [kg/s]
C_p = Specific heat capacity [J/(kg·°C)]
ΔT = Temperature difference between outlet and inlet [°C]

Note that equation 5 is valid only for one-phase heat exchange.

The specific heat capacity can be interpreted as the amount of energy required to increase the temperature of 1 kg liquid by 1°C at constant pressure. The specific heat capacity varies for different liquids and different temperatures.

Equations 2 and 5 together describe the preservation of energy inside a CBE, which is shown in equation 6. This equation, as well as Figure 1.7, indicates that there are no theoretical heat losses to the surroundings in a CBE.

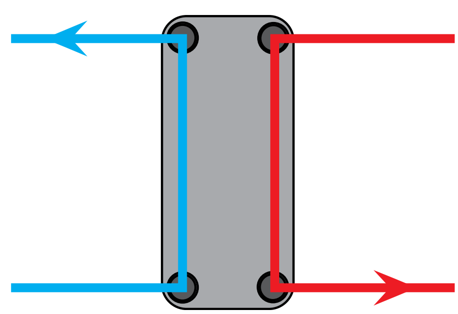


Figure 1.7 The preservation of energy in a CBE.

$$(6) \qquad \dot{Q} = \dot{m}_1 \cdot C_{p1} \cdot \Delta T_1 = k \cdot A \cdot LMTD = \dot{m}_2 \cdot C_{p2} \cdot \Delta T_2$$

1.6 Co- versus Counter-Current Flow

The flows inside a heat exchanger can be arranged in various ways to fulfil different purposes. The possibilities are counter-current flow, co-current flow and cross-flow, the first two of which are shown in Figure 1.8. There are different forms of so-called cross-flow heat exchangers in which the flows are more or less mutually perpendicular.

In CBEs, counter-current flow is by far the most common arrangement. In this case, it is possible for the cooling liquid to leave at a higher

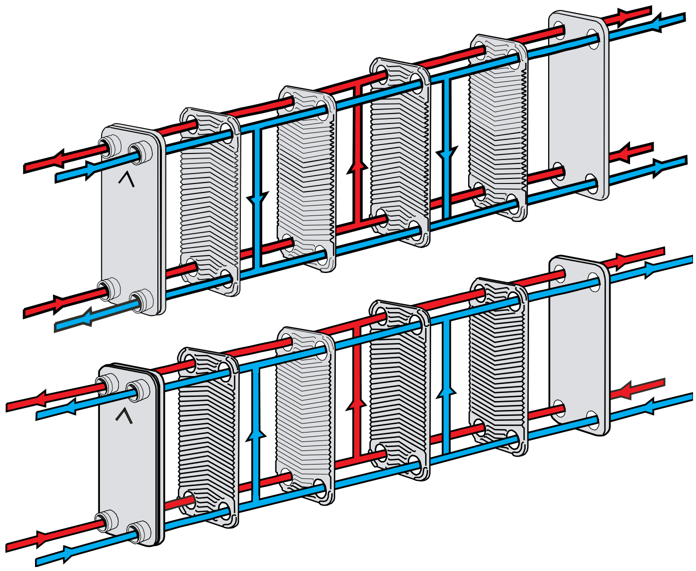


Figure 1.8 Counter-current flow (top) and co-current flow (bottom) through a CBE.

temperature than the heating liquid outlet temperature. One of the great advantages of counter-current flow is the possibility of extracting a higher proportion of the heat content of the heating fluid. It is important to note that the LMTD value for counter-current flow is much larger than for co-current flow at the same terminal temperature (see Figure 1.9).

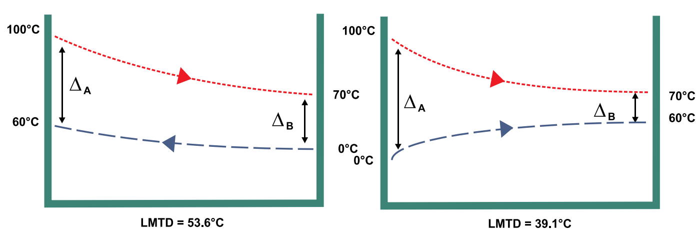


Figure 1.9 LMTD comparison between counter-current and co-current arrangement.

The motive for using counter-current flow becomes obvious by referring to equation 6. A high LMTD implies that a smaller heat transfer area is needed, i.e. the CBE can be manufactured with fewer plates. However, co-current flow also occurs in CBEs when the application so demands, e.g. in flooded evaporators. A consequence of a co-current arrangement is that the outlet temperature of the cooling medium can never exceed the outlet temperature of the warming medium. It is also worth noting that in a co-current arrangement there is a large temperature gradient at the beginning of the heat exchanger, which makes boiling start earlier.

1.7 Thermal Length

The thermal length demand is a measure of how "difficult" a certain operational case is to solve for the heat exchanger. The thermal length can be expressed as the Number of Heat Transfer Units (**NTU** or θ). As shown in Figure 1.10, it is possible to calculate the NTU for each side of the heat exchanger.

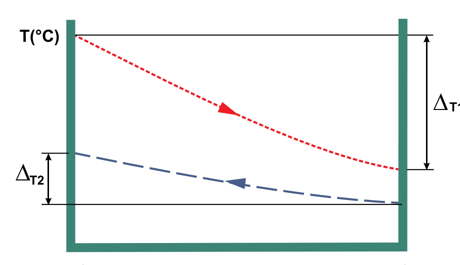


Figure 1.10 The definition of NTU.

A CBE with a long thermal length can solve cases that are thermodynamically more problematic than a CBE with short thermal length. Different cases are shown in Figure 1.11, and the possible solutions for these cases are discussed below.

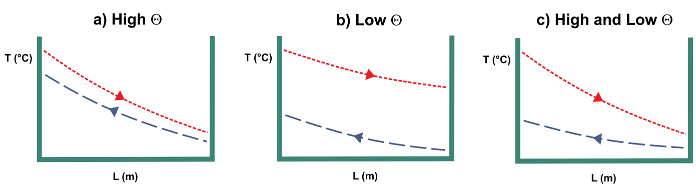


Figure 1.11 Different temperature programs require different solutions.

Operations with **close temperature programs** (Figure 1.11 a) demand long CBE plates or a **multi-pass** CBE. The purpose of those solutions is to enhance heat transfer through the CBE. Another solution is to design the CBE with a **high-theta pattern** (cf. Figure 1.13), which will increase the turbulence of the fluid and thus increase the heat transfer efficiency.

An easier case, i.e. with small temperature changes on each side (Figure 1.11b), will most probably be solved with a short **single-pass** CBE. The plate pattern for easy jobs may very well be a **low-theta pattern** (cf. Figure 1.13), which leads to a low pressure-drop through the heat exchanger.

Asymmetric operation (Figure 1.11 c) occurs when the temperature change on one side of the CBE is much larger than on the other. The solution for this is a **two-pass over one-pass** CBE, as shown in Figure 1.12, or an asymmetric CBE. An asymmetric CBE uses a design with a mixture of high θ and low θ plates.

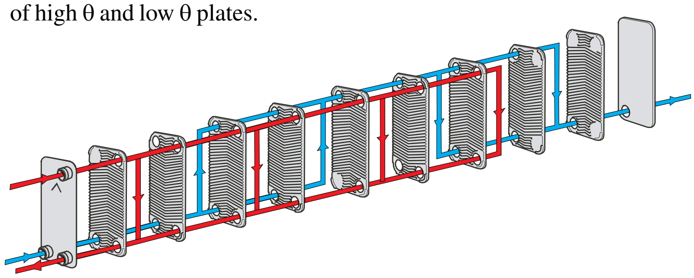


Figure 1.12 Two-pass over one-pass configuration.

1.8 Pressure Drop

When a flow is disturbed, a pressure drop (ΔP) is created, i.e. the flow pressure at the beginning of a passage is higher than at its end. Pressure drop is a phenomenon with both positive and negative consequences for the heat transfer process.

Pressure Drop Utilization

Excessive pressure drop is of course negative, because the flow must then be pushed through the CBE using a lot of pump power. High pump power can be achieved only with large pumps, which demand a large amount of electricity and thus make the operation expensive.

The positive result of pressure drop is the greater extent of turbulence obtained. Turbulence is desirable in heat exchangers, because it improves heat transfer (as discussed in chapter 1.4).

There are some proportional(~) relations that are useful to keep in mind when a CBE is to be designed:

[1] $v \sim \frac{1}{A}$

[2] $\Delta P \sim v^2$

[3] $k \sim \sqrt[3]{\Delta P} = \Delta P^{1/3}$

Relation [1] tells that decreasing the area, A, increases the flow velocity by the same factor. The interpretation of relation [2] is that if the velocity, for example, is doubled, the pressure drop is increased four times. The interpretation of relation [3] is that if the pressure drop, for example, is increased four times, the heat transfer coefficient becomes $4^{1/3} = 1.26$ times higher than the original value, i.e. a 26% increase. Note that these proportionalities only apply to fully turbulent flow.

Pressure Drop Relations

The pressure drop in a CBE channel mainly depends on different variables, shown in Table 1.1.

Table 1.1. Some pressure drop relations.

Variable	Proportional to:
Pressing depth	$1/\Delta P^3$
Plate width	$1/\Delta P^2$
Plate length	ΔP
Plate pattern	High-theta pattern means higher ΔP
Port size	Small port size means higher ΔP

The plate pattern is one of the tools the designer can use to increase or decrease the pressure drops through the CBE. A high θ pattern leads to bigger pressure drop than a low θ pattern. In practice, the high θ pattern is much more common due to the greater turbulence it creates. The difference between high θ and low θ patterns is visualized in Figure 1.13.

Figure 1.13 High-theta (left) and low-theta (right) plates.

2 Compression Cycle

The purpose of a refrigerant system is to transfer heat from a cold chamber that is at a lower temperature than that of its surroundings. Heat is absorbed at a low temperature and rejected at a higher temperature as work is supplied.

This chapter explains the various phenomena that occur in a compression cycle (such as compression, condensation, expansion, evaporation, superheating and sub-cooling), and describes the different components necessary for such an operation.

2.1 The Pressure-Enthalpy Diagram

The pressure-enthalpy diagram (log P/h diagram) is a very useful tool for refrigerant technicians. First, an explanation of how the diagram is built up is given, and then its use is described.

Figure 2.1 shows the principle of a log P/h diagram, and indicates the refrigerant's various thermodynamic states. This diagram can be seen as a map of the refrigerant. The area above and to the left of the saturation line for liquid (A-CP in Figure 2.1) is the area where the refrigerant is sub-cooled, i.e. the temperature is lower than the saturation temperature for the pressure range in question. The area above and to the right of the saturation line for gas (CP-B in Figure 2.1) is the area where the gas is superheated, or overheated, i.e. the gas has a higher temperature than the saturation temperature at that pressure. The area below the saturation lines for liquid and gas (A-CP-B in Figure 2.1) represents the conditions where the refrigerant can change its state of aggregation from liquid to gas or vice versa. Hence, there is a mixture of gas and liquid.

The practical meaning of the critical point (CP) is that at temperatures higher than this, the refrigerant cannot be condensed, no matter how high the pressure. Therefore, compression refrigeration systems normally operate at temperatures below the critical one.

Lines of constant temperature (isotherms) are vertical in the sub-cooled liquid region, horizontal (i.e. parallel to the constant pressure lines) in the liquid + vapor mixture region, and drop steeply towards the enthalpy axis in the superheated gas region (see Figure 2.2). The constant pressure lines (isobars) are parallel to the x-axis.

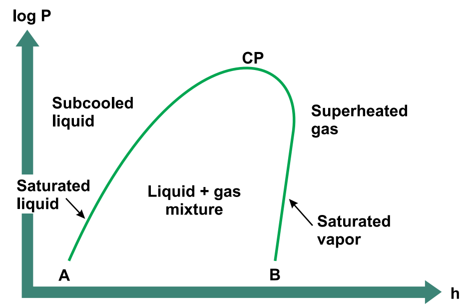


Figure 2.1 The log P/h diagram for a typical refrigerant.

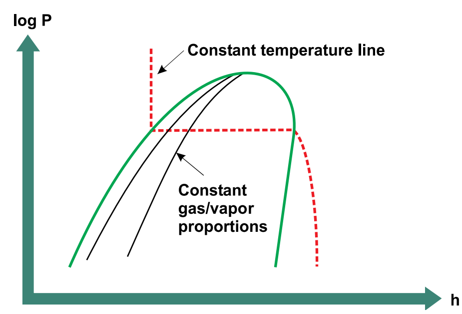


Figure 2.2 The log P/h Diagram with temperature lines.

2.2 Basic Components

The science of refrigeration is based on the fact that liquid can be vaporized at any desired temperature by changing the pressure around it. Water under normal atmospheric pressure of 1.01 bar will boil at 100˚C. The same water in a closed vessel under a pressure of 4.6 bar will not boil until its temperature has reached 149˚C. Liquids boiling at low temperatures are the most desirable media for removing heat, i.e. refrigerants. Comparatively large quantities of heat are absorbed when liquids are evaporated. Many of the liquids used as refrigerants in refrigeration systems have boiling points below –18˚C under ordinary atmospheric pressure. An example is ammonia, which boils at -33˚C.

Refrigeration can be achieved with ammonia without any equipment whatsoever. If liquid ammonia is poured into an open container (an **evaporator**) surrounded by ordinary air at ordinary atmospheric pressure, it will immediately begin to boil at -33˚C. There will be a continuous flow of heat from the surrounding air, through the walls of the container to the boiling ammonia (see Figure 2.3). Moisture from the air will condense and freeze on the exterior of the container. Such a system would work satisfactorily as far as cooling alone is concerned, but the cost of replacing the lost ammonia would be high. The ammonia, or any other refrigerant, is therefore used repeatedly. Additional equipment is needed for this pur-

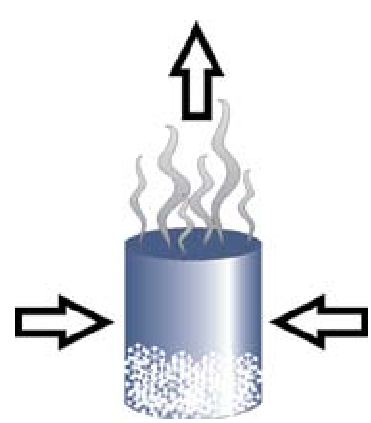


Figure 2.3 A very simple evaporator.

pose. The refrigerant must be delivered to the evaporator as a liquid because it absorbs heat best by vaporizing. Because the refrigerant leaves the evaporator as a vapor, it needs to be condensed to a liquid before it can be used again. To condense a refrigerant, the latent heat given off by the vapor must be transferred to another medium, such as water or air, in a **condenser** (see Figure 2.4). The surrounding medium must be at a lower temperature than the condensing temperature of the refrigerant.

As the temperature of the available water or air is always higher than that of the boiling refrigerant in the evaporator, the refrigerant cannot be condensed as it leaves the evaporator. To condense the vapor, its pressure must be increased to a point where its condensing temperature is above the temperature of the water available for the condensing process. For this purpose, a **compressor** is needed (see example in Figure 2.5).

The compressor and condenser are needed to enable the same refrigerant to be used again and again. The cost of compressing and condensing the vaporized refrigerant is far less than the cost of continuously buying new refrigerant to replace that used.

To maintain the difference in pressure between the condenser and the evaporator caused by the compressor, an **expansion valve** is needed in the cycle. The expansion valve separates the high-pressure part of the system from the low-pressure part. Only a small trickle of refrigerant liquid flows through the valve. In fact, the valve is always adjusted so that the rate at which liquid passes through it is the same as the evaporation rate. An expansion valve is shown in Figure 2.6. The simplest refrigeration system therefore consists of an evaporator, a compressor, a condenser and an expansion valve (see Figure 2.7).

The refrigerant boils in the evaporator at a constant low pressure and temperature. Heat is removed from the fluid being cooled. After leaving the evaporator, the vaporized refrigerant flows through the compressor. In the compressor, the pressure of the vaporized refrigerant is raised to a point at which it can be condensed by some relatively warm fluid, e.g. water. The compressor removes the refrigerant vapor. This creates such a low pressure in the evaporator that the evaporation temperature is kept below the surrounding temperature. The work input for this process is represented by W in Figure 2.7.

After being compressed, the vapor enters the condenser and is condensed at constant pressure and temperature. Latent heat is transferred from the condensing vapor through the walls of the condenser. The expansion valve has two functions: maintaining the pressure difference between the condenser and the evaporator, together with the compressor, and regulating the volume of refrigerant going to the evaporator.

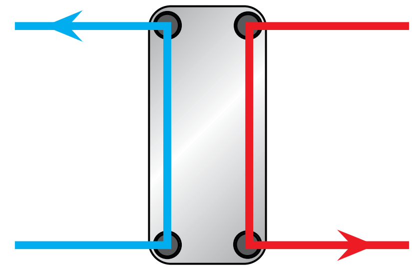


Figure 2.4 Heat exchanger working as a condenser.

Figure 2.5 Scroll compressor.

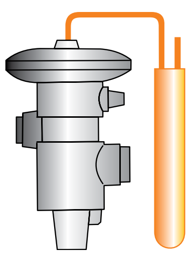


Figure 2.6 Expansion valve.

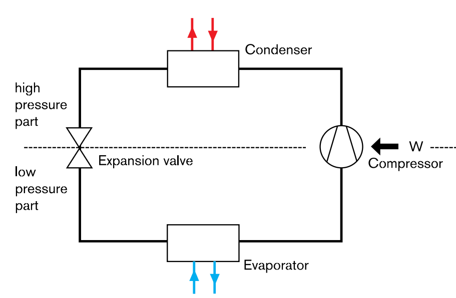


Figure 2.7 The compression refrigeration cycle.

2.3 The Basic Cycle in a log P/h diagram

Figure 2.8 shows the fundamental process that describes the ideal refrigeration cycle. The following are assumed for the fundamental process:

- No sub-cooling of the liquid or superheating of the gas
- Ideal compression, i.e. it occurs at constant entropy

- No pressure drop losses

The fundamental process can be divided into the four parts described below, the numbers referring to the numbers in Figure 2.8:
(1-2): The **compression** is performed at constant entropy.
(2-3): The **condensation** is performed at constant pressure and thus follows the isobars.
(3-4): The **lowering of the pressure** is performed at constant enthalpy (h).
(4-5): The **evaporation** is performed at a constant pressure and thus follows the isobars, i.e. the pressure at states 1 and 4 is equal to the evaporator pressure (temperature). State 1 is determined by the temperature of the gas leaving the evaporator.

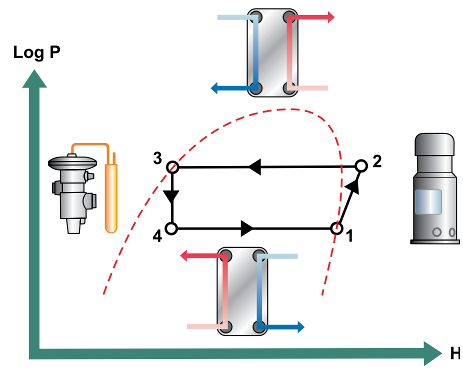


Figure 2.8 The basic refrigerant cycle (also called the Carnot cycle) in a log P/h diagram.

2.4 The Complex Cycle in a log P/h diagram

In reality, the pressure drops that occur in the evaporator, condenser and piping must be considered. There are also mechanical and electrical losses in the compressor. The consequences are increased operational and maintenance expenses. However, some measures can be taken to minimize the costs. The main ones are discussed below.

Superheating of the Refrigerant Gas

In the fundamental process, the gas entering the compressor was assumed to be dry and saturated. In reality, the gas is overheated as shown in Figure 2.9 (point 1.2). The overheating is the difference between the temperatures at points 1.1 and 1.2 in the figure and is created in the end of the evaporator. It is a practical necessity to allow the refrigerant vapor to become superheated to prevent the carry-over of liquid refrigerant into the compressor, where it may cause severe damage due to its incompressibility. It may also contaminate the lubricants. The level of superheating should be kept to a minimum to minimize both the work to be done by the compressor and the necessary heat transfer surface in the evaporator.

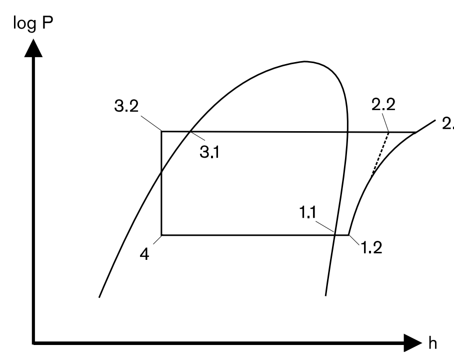


Figure 2.9 The real refrigerant cycle in the log P/h diagram. The overheating is the difference between the temperatures at points 1.1 and 1.2. The sub-cooling is the difference between the temperatures at points 3.1 and 3.2.

Sub-Cooling of the Refrigerant Liquid

In the fundamental process, the liquid leaving the condenser was just on the saturation line for liquids. The pressure drop in the pipes, filters, etc., before the expansion valve is negligible, but still causes “flash gas”, i.e. vaporization of a small part of the liquid. The condensed liquid is therefore sub-cooled to a temperature below that of the saturation temperature corresponding to the condenser pressure, for two reasons: the cooling capacity of the refrigeration process is increased and the risk of gas bubbles in the flow fed to the expansion valve is avoided. (Gas bubbles in the inlet flow to the expansion valve disrupt the regulation mechanism.) The sub-cooling is the difference between the temperatures at points 3.1 and 3.2 in Figure 2.9, and is generated in the condenser or in a separate heat exchanger after the condenser.

The Compression Process

In the real refrigeration process, compression does not follow the lines of entropy (see Figure 2.9) as it does in the ideal, fundamental process. This means that the compression work increases. The ratio of the theoretical to the real compression work is called the **isentropic efficiency**.

Efficiency Definitions

The basic compressor-driven refrigeration cycle consists of one compressor, two heat exchangers (a condenser and an evaporator) and a throttling device (expansion valve). These components form the circuit in which the refrigerant circulates. The cycle operates between the two pressure levels p_1 and p_2, and the temperatures T_1 and T_2, where $T_1>T_2$ (see Figure 2.10).

The refrigerant receives energy in the cold chamber at a temperature below that of the surroundings. The energy at rejection is of a higher quality than in the cold chamber, because of its higher temperature. This energy can be used for heating purposes. Plants designed entirely for this purpose are called heat pumps. The term "heat pump" is appropriate because energy is transferred against a natural temperature gradient from a low temperature to a higher one. It is analogous to the pumping of water from a low level to a higher one against the natural gradient of gravitational force. Both actions require an input of energy for their accomplishment. There is no difference in operation between a refrigerator and a heat pump. However, in a refrigerator the desired effect is the removal of energy from the cold chamber, represented by Q_2 (refrigerating effect) in Figure 2.11. In a heat pump, it is the energy to be rejected by the refrigerant, Q_1, for heating purposes (see Figure 2.11) that is desired. Both actions also follow the first law of thermodynamics:

"When a system undergoes a thermodynamic cycle, the net heat supplied to the system from its surroundings plus the network input to the system from its surroundings is equal to zero."

Applying the first law of thermodynamics to the refrigerant system in Figure 2.11 gives the following equations:

$$\sum Q + \sum W = 0 \quad \text{or:}$$

$$Q_1 + Q_2 + W = 0 \quad \text{therefore:}$$

$$(1) \qquad W + Q_2 = -Q_1$$

The power input, W, is important because it is the quantity that has to be paid for and constitutes the main item of the running costs.

Refrigerator and heat pump performances are defined by means of the coefficient of performance, COP, which is defined as:

$$(2) \qquad COP_{hp} = \frac{Q_1}{\sum W} = \frac{T_1}{T_2 - T_1}$$

$$(3) \qquad COP_{ref} = \frac{Q_2}{\sum W} = \frac{T_2}{T_2 - T_1}$$

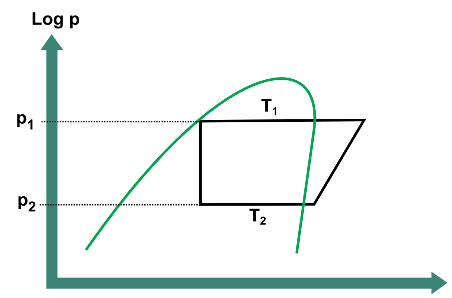


Figure 2.10 Log P/h diagram with temperature levels shown.

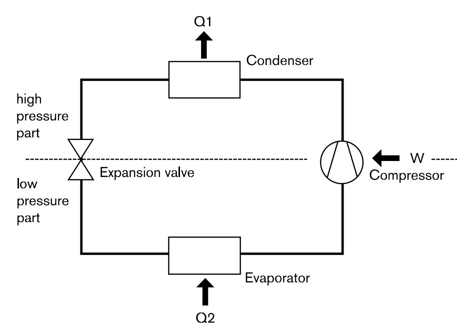


Figure 2.11 The energy flow in a refrigerant system.

2.5 Other Components

A refrigeration system contains a minimum of four key components: compressor, condenser, expansion valve and evaporator. However, in practice the systems are much more complicated. A number of components make the system more efficient, reliable or controllable, as shown in Figure 2.12. Many of these components must be selected with care, because they can have a negative influence on the system if they are incorrectly selected or mounted. The most important components are described below. Information about problems that could affect the refrigeration process and the performance of heat exchangers and expansion valves has been *italicized.*

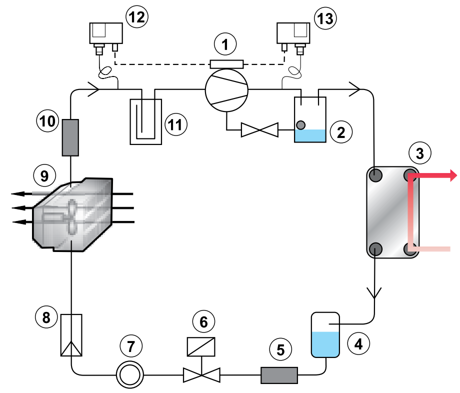


Figure 2.12 A detailed refrigerant system containing the components: (1) Compressor, (2) Oil separator, (3) Condenser, (4) Receiver, (5) Filter dryer, (6) Solenoid valve, (7) Sight glass, (8) Expansion device, (9) Evaporator, (10) Suction line filter, (11) suction accumulator, (12) Low-pressure gauge, (13) High-pressure gauge.

Oil Separator

All refrigeration compressors except centrifugal compressors contain oil, which lubricates the compressor and forms seals between the moving parts during the compression. The oil is important to achieve high efficiency in the compressor, but it affects the heat transfer in the system negatively.

There will always be some oil in the discharge gas from the compressor. Screw compressors usually have much more oil carry-over than other types.

To minimize the amount of oil that must be transported around the system, an **oil separator** can be introduced (see Figure 2.13). It is important to remember that there is no such thing as a 100% efficient oil separator, which means that all systems must be dimensioned to ensure adequate oil transport. The oil separator will only delay the failure of a system without proper oil transport. *Oil transport can often be a problem during part-load operation when the velocities are low. This can create problems during continuous operation at part load.*

The oil separator is mounted in the high-pressure pipe between the compressor and the condenser. When the high-pressure gas enters the oil separator, the gas velocity decreases. This causes the oil that is transported by the refrigerant gas to be captured by the metallic filter in the inlet. The oil then forms droplets, which fall down to the bottom of the oil separator. When the oil level is high enough, a float-valve is opened that allows the high pressure to force the oil back to the compressor. As a final precaution against oil being swept into the system, there is a strainer in the outlet that also captures oil.

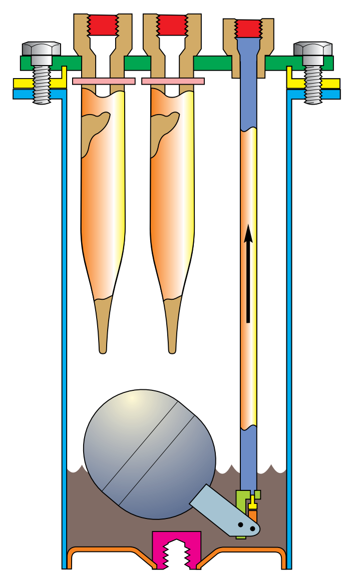


Figure 2.13 Oil separator.

At high evaporation temperatures, such as in air conditioning systems, oil separators are common only in screw compressor systems. This is because the small amount of oil from scroll compressors and reciprocating compressors will be carried around the system without any significant effect. At lower temperatures, the viscosity of the oil increases greatly, and consequently its impact on the heat transfer increases. The problem of oil carry-over increases with higher viscosity oils and lower density gases.

Refrigerant Liquid Receiver

Where an evaporative condenser, an air-cooled condenser, or a tube-within-a-tube condenser without sub-cooling is employed, a **receiver** is necessary to collect the condensed refrigerant used by the system. The receiver also stores the entire charge of the refrigerant during the period when the system is pumped down. The receiver is usually just a small steel tank with appropriate shut-off and relief valves (see Figure 2.14).

The receiver is placed in the **liquid line** (i.e. the part of the refrigeration cycle where the refrigerant is in the liquid phase) between the condenser and the filter (cf. Figure 2.12). It should be filled partly with liquid and partly with vapor in all operating conditions. A receiver will be needed only in systems where the evaporator volume is large compared with the condenser volume and a wide operation range is desired or pump-down operation is used. The receiver can also be used as a service vessel to hold the charge while work is performed on other parts of the system. Unlike most systems, those using air-to-refrigerant evaporators operating under varying conditions often require a receiver.

Although there are advantages in having a receiver, in many systems there are strong reasons to avoid one. Because the receiver contains both gas and liquid, little or no sub-cooling can be achieved at the outlet, which is an important difference from a system without a receiver. With little or no sub-cooling, all pressure drops or heat transfer to the liquid line will cause flash gas, which means the expansion valve has no opportunity to function correctly. To achieve substantial sub-cooling, a separate sub-cooler is required after the receiver.

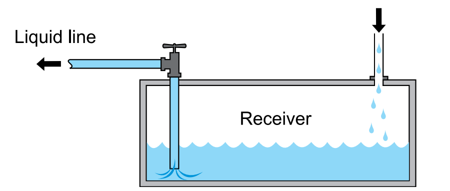


Figure 2.14 Liquid receiver.

Filters and Driers

Moisture, water vapor or foreign matter may cause problems in any refrigeration system. Moisture may freeze in the orifice of the expansion valve, cause corrosion of metal parts and wet the motor windings of a semi-hermetic compressor. Eventually, this can result in motor burnout and oil sludge. Foreign matter, on the other hand, may contaminate the compressor oil and become lodged in valve parts, rendering the valve inoperative. Various types of devices are used to remove water vapor and foreign matter from the refrigerant stream, some of which are described below.

Liquid line filter driers are positioned in the liquid line to protect the expansion valve from particle contamination and to absorb potential humidity in the refrigerant. Different desiccant materials are available to suit the various types of refrigerants.

This type of filter drier should always be installed in the system. If not, the expansion valve could become blocked by particles or by ice formation due to humidity. Damage to the compressor could also follow. The combination of humidity and high discharge gas temperatures will accelerate decomposition of the oil. This also increases the risk of compressor failure.

If a filter in the liquid line becomes blocked, there will be a pressure drop causing the liquid refrigerant to boil and create flash gas, which will disrupt the operation of the expansion valve.

Suction line filters are positioned in the **suction line** (i.e. before the compressor) to protect the compressor from possible contamination.

However, they are not always installed. After burnout, when the electric motor in a hermetic or semi-hermetic compressor has failed, they are often installed temporarily or permanently.

If a filter in the suction line becomes clogged, the suction pressure and capacity will decrease, which is often mistaken for an evaporator problem.

Oil filters are often installed in the oil return line between the oil separator and the compressor. *If this filter is blocked it can lead to excessive oil carry-over, which affects heat transfer in the evaporator.*

Figure 2.15 Filter dryer.

Pressure Control

Pressure gauges, permanently installed to monitor compressor suction and discharge pressures, are convenient when maintaining the system. Because of the pulsating nature of the refrigerant gas flows within a reciprocating compressor system, it is advisable to connect gauges to the system using throttle valves. These valves not only provide a shut-off when gauge readings are not required, but also provide a means for throttling the lines to prevent gauge fluctuation when readings are to be taken.

Compressor shut-off valves or service valves are safety devices to protect the system from excessively low or high compressor pressures, i.e. low/high evaporation temperatures. A compressor shut-off valve is shown in Figure 2.16. If the pressure drops below a pre-set level, the compressor is stopped. The valve can also be used to start the compressor when the pressure reaches a pre-set level. The valve also acts as a safety device that protects the system from excessively high pressures by controlling the compressor. It measures the pressure at the compressor outlet, and the compressor is stopped if this pressure becomes too high.

This component is apparently ideal for freeze protection by preventing low evaporation temperatures. However, for practical reasons the low-pressure control must often be by-passed during start-up and will therefore not protect against freezing. Unfortunately, the start-up is the situation with the highest risk of freezing (cf. chapter 8.4).

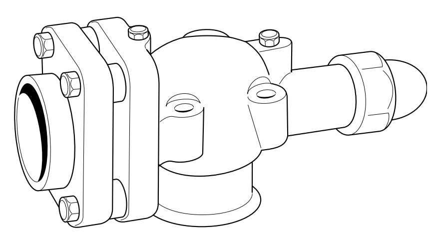


Figure 2.16 Compressor shut-off valve.

Sight Glass with Humidity Indicator

A **sight glass**, see Figure 2.17, should be positioned in the liquid line in every system (cf. Figure 2.12). It is generally located immediately before the expansion valve, and provides a means for viewing the liquid flow.

If a gas/liquid mixture is detected in the sight glass, this indicates that the sub-cooling is insufficient. This will disturb the operation of the expansion valve and may result in severe superheating or ***system hunting****. The presence of gas in the liquid may be due to the following:*

1. *No sub-cooling being achieved in the condenser*
2. *A high pressure-drop in the liquid line, caused by e.g. a blocked filter drier or a solenoid valve with high pressure-drop*
3. *Heat transfer to the liquid line from the surroundings*

In a system without a liquid receiver, (1) can be solved by charging more refrigerant. When doing so, the extra refrigerant will collect as

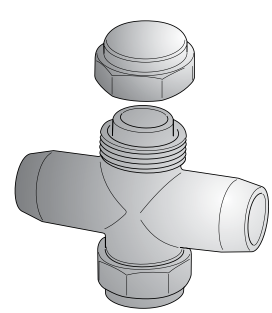


Figure 2.17 Sight glass.

liquid in the condenser. The liquid level will then rise in the condenser and the sub-cooling will thereby increase. Note that in a system containing a liquid receiver, the sub-cooling cannot be increased by charging more refrigerant. Problem (2) is solved by changing the filter or modifying the system in other ways to reduce the pressure drop. Problem (3) can be solved by insulating the liquid line or increasing the sub-cooling. If this is not possible, the condensing pressure may have to be increased at the expense of a higher energy cost.

The sight glass also includes a humidity indicator. A color scale shows the degree of humidity in the refrigerant and hence warns when it is time to change the filter drier.

Solenoid Valves

A **solenoid valve** (see Figure 2.18) is an on/off valve that is often positioned in the liquid line before the expansion valve. The solenoid valve is electronically controlled, and closes at the same time as the compressor is switched off. The solenoid valve will maintain the pressure difference between the condenser and the evaporator side during off periods and hence prevent liquid from flowing into the evaporator.

Solenoid valves can be direct or pilot operated. The direct type is usually only manufactured in small sizes. *Pilot operated valves require a certain pressure drop to open. In this case, there is always a risk that flash gas will disturb the operation of the expansion device in systems with limited sub-cooling.*

Figure 2.18 Solenoid valve.

Suction Line Accumulator

Suction line accumulators are installed to avoid liquid carry-over to the compressor, which could affect lubrication and, in the worst-case scenario, lead to damage if liquid or oil foam enters the compression chamber. This may cause the expansion device to malfunction. It is also a problem during start-up and after defrost, when large amounts of liquid refrigerant can be flushed back to the compressor.

The pressure drop in a suction line accumulator is, as for all components in the suction line, critical. Pressure drop in the accumulator can be mistaken for poor evaporation performance.

3 Compressors

The function of a compressor is to remove the vapor produced by the evaporator and to deliver it at a required higher pressure. The compressor can be compared to a heart pumping the blood (the refrigerant) inside the body (the compression cycle). In the basic compression cycle, the compressor is positioned between the evaporator and the condenser. Compressors can be installed in either single or multistage configuration, and can be connected to each other in series or in parallel.

This chapter discusses the most common working principles of different compressors, and presents the intimate relationship between the compressor and the evaporator.

3.1 General Function and Theory

For a liquid inside a vessel to boil, the pressure over the liquid surface most correspond to the boiling temperature. The gas produced due to the addition of heat must be removed continuously if the same boiling point, i.e. the same pressure, is to be maintained.

In a refrigerant system, the "vessel" referred to above is the evaporator, and the device removing the gas is the compressor. The compressor pumps gas from the evaporator and compresses it, i.e. increases its pressure. The energy required for the compression normally comes from electricity.

The red compression line in Figure 3.1 symbolizes an ideal isentropic compression, i.e. compression at constant entropy, in which there is no heat exchange with the surroundings. In reality, however, there are always some heat losses from the vapor due to mechanical friction in the equipment, i.e. flow frictional work between the fluid and the wall, and potential leakage in the compressor.

Figure 3.1 illustrates a comparison between a realistic case (blue) and the theoretical isentropic compression (red). The difference between isentropic and actual compression can be expressed by the isentropic compressor efficiency, η_{IS}.

$$[1] \qquad \eta_{IS} = \frac{W_{ISENTROPIC}}{W_{ACTUAL}} = \frac{h_2 - h_1}{h_3 - h_1}$$

The efficiency of a compressor can be indicated in various ways. Another example is **volumetric efficiency**, η_{VOL}, which is the ratio between the actual vapor volume and the theoretical maximum volume that can be contained in the compressor cylinder:

$$[2] \qquad \eta_{VOL} = \frac{V_{ACTUAL}}{V_{THEORETICAL}}$$

It is always desirable to achieve high compressor efficiency to minimize the compressor work and maintain conditions in the refrigerant system. The compression ratio shows the ratio between the high-pressure and low-pressure sides of the compressor:

$$[3] \qquad \text{Compression ratio} = \frac{P_{high}}{P_{low}}$$

A high compression ratio, i.e. a large pressure difference between the low-pressure and the high-pressure sides, requires more compressor work.

3.2 Compressor Types

There are several types of compressors, as listed in Table 3.1.

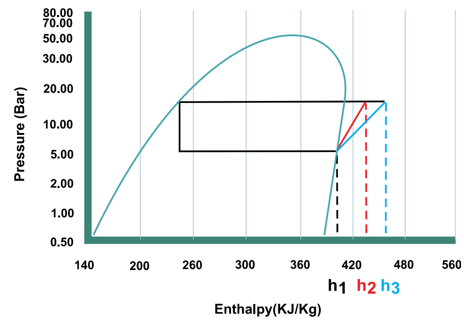


Figure 3.1 Comparison between isentropic and real compression in a log P/h diagram.

Table 3.1 Compressor types.

Positive displacement compressors	Dynamic compressors
Reciprocating or piston type compressors	Centrifugal or turbo compressors
Screw compressors	
Scroll compressors	
Rotary compressors	

The working principles of displacement compressors and dynamic compressors differ significantly. In **positive displacement compressors**, a certain volume of gas is trapped in a space that is continuously reduced by the compressing device (piston, scroll, screw or similar) inside the compressor. The reduction in volume increases the pressure of the vapor when the compressor is operating. The principle of a **centrifugal compressor**, also called a turbo compressor, is different. Here, the gas is compressed by being accelerated by an impeller. The pressure is further increased in the diffuser, where the speed is transformed to pressure. Centrifugal compressors are of interest for very large capacities, where the inlet flows may be approximately 2000 m^3/h or more. CBE evaporators and condensers cannot handle such large capacities, so they are not compatible with centrifugal compressors. However, CBEs may well be used as oil coolers for centrifugal compressors.

In addition to their different working principles, compressors can also be distinguished according to their basic type of construction, as shown in Table 3.2.

Table 3.2 Classification of compressors according to their size.

Type	Type of construction	Size
Open compressor	No shell	Large
Semi-hermetic compressor	Shell that can be opened	Medium
Hermetic compressor	Closed shell	Small

In an **open compressor**, the motor and the compressor housing are mounted separately. Because the open compressor lacks a seal around it, there is risk of refrigerant leakage. The advantages are that the compressor components are easily accessible for maintenance and the costs of a shell can be avoided.

In a **semi-hermetic compressor**, the motor and the compressor housing are located in a two-piece shell. The covers are bolted together, allowing the cover to be opened for servicing, etc. Semi-hermetic compressors are generally a little more expensive than hermetic compressors, due to the bolts and O-rings needed to join the covers.

A **hermetic compressor** also houses both the motor and the compressor housing inside a shell. However, the steel shell is welded, which provides a true hermetic seal against the surroundings. It is impossible to open the welded shell of a hermetic compressor, and the compressor must therefore be scrapped in the event of damage to the motor or compressor.

The reason for the general size grouping is to show the possibilities for maintaining and repairing expensive compressors, which is less important for small, mass-produced hermetic compressors.

Reciprocating Compressors

Reciprocating compressors (see Figure 3.2), also called piston compressors, are still widely used but have faced increasing competition from other compressor types in recent decades.

Inside the reciprocating compressor housing, one piston moves up and down in each cylinder. When the piston is at its lowest point, superheated gas enters the compressor through the inlet valves. When the piston moves up, the inlet valve closes and the gas pressure increases, due to the reduced volume. The compressed gas leaves the compressor when the pressure is high enough to open the exit valve. The downward piston action initiates a new intake of gas through the valves.

The advantage of reciprocating compressors is the relatively simple working principle and construction. The main component, a circular cylinder with a suitable piston, can be manufactured quite easily with good accuracy. A disadvantage of reciprocating compressors is that they have many moving parts, which makes it almost impossible to avoid vibrations. Another disadvantage is the "dead space". When the piston is at its top position, some of the compressed gas will be trapped in the space between the top of the piston and the cylinder roof. The gas in the dead space results in lower volumetric efficiency, because less fresh gas is compressed on each piston stroke than the total volume of the cylinder could actually admit.

The valves controlling the flow of gas to and from the compressor are sensitive to droplets in the gas. If a considerable amount of liquid enters the compressor housing, a very high pressure can be built up when the piston reaches its top position, which may cause severe damage to the valves or crankshaft. This phenomenon is called **liquid hammer**.

Figure 3.2 Reciprocating compressor (Courtesy of Danfoss)

Screw Compressors

Thanks to the improvements in screw compressors in recent years, they have become more common in air-conditioning and mid-range refrigerant applications. They will probably become even more popular, and replace many large (from 50 kW) reciprocating compressors. Screw compressors are produced in two different configurations: the **twin-screw compressor**, also called the Lysholm type after its inventor, and the **single-screw compressor** (see Figure 3.3).

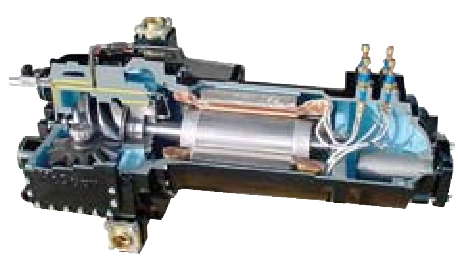


Figure 3.3 Single-screw compressor (Courtesy of McQuay International)

The twin-screw, the most common type, is composed of two rotors with complementary profiles referred to as screw and slide rotors, or male and female rotors. The rotor profiles are designed to decrease the volume between them continuously from the inlet to the outlet of the compressor. Unlike reciprocating compressors, screw compressors have no dead space. The refrigerant is fed from the low-pressure to the high-pressure side with a continuously decreasing volume, i.e. continuously increasing pressure. Screw compressors therefore have neither suction valves nor pressure valves, only a non-return valve to ensure that there is no return flow of refrigerant when the compressor is stopped.

Screw compressors can work at a high compression ratio because the oil, in addition to its lubrication and sealing functions, also absorbs compression and friction heat during the process. Proper oil cooling is therefore essential in a screw compressor, and can be provided either by the injection of refrigerant into the compressor or by a separate oil cooling system. CBEs are widely used as oil coolers.

Scroll Compressors

The advantages of scroll compressors have been known since the early years of the 20th century. The reason for their not being introduced on a large scale until the 80's was the difficulty of producing scrolls, which requires very high precision.

Scroll compressors capture the gas in the volume formed between one fixed and one orbiting scroll. The orbiting scroll is driven by an electric motor, which rotates a shaft. Note that the scrolls perform an orbiting motion. They do not rotate.

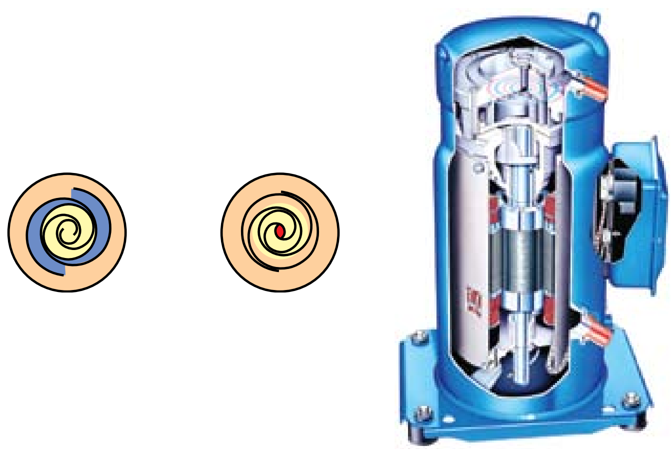


Figure 3.4 Scroll compressor (Courtesy of Danfoss)

Figure 3.4 explains the scroll compressor function. Superheated gas (blue) enters at the outer ends of the spirals and is compressed on its way through the scrolls due to the orbiting motion of one of the spirals. The discharge hole, where high-pressure gas (red) leaves, is located in the center of the scrolls.

Scroll compressors are available in both open and hermetic design. They have several advantages over reciprocating compressors:

- The absence of suction and discharge valves eliminates pressure drops and consequential noise and vibrations.
- The scrolls have no dead space, which results in volumetric efficiencies close to 100%.
- Fewer moving components, leading to a lower failure rate.
- They are relatively insensitive to liquid droplets in the suction gas from the evaporator.

3.3 Compressor Lubrication

Compressors are lubricated for three main purposes:

- To reduce frictional wear on bearings and other moving parts of the compressor
- To cool the refrigerant gas during compression
- To seal against refrigerant gas leakage

Different compressors use different lubrication techniques. In screw compressors, the oil is often pumped into the moving parts, while pistons and scrolls often employ splash lubrication using oil from a vessel in the bottom of the compressor.

If the viscosity of the oil-refrigerant mixture is too low, it leads to incomplete or inefficient separation of the metal surfaces, which increases friction and wear. Various anti-wear additives can counteract this to some extent, but this solution cannot be used to its full extent in refrigeration systems due to the risk of reaction between the additives and the refrigerant.

With high-viscosity oil-refrigerant mixtures, there may be problems, such as obstructed flow, that may lead to poor pumping efficiency. In order to work properly, oil-refrigerant mixtures should have a dynamic viscosity that is sufficiently high to give satisfactory sealing and lubrication in the compressor. In addition, the mixture must be thermally and chemically stable, so as not to react with components and materials in the refrigeration system.

Lubricating oil may have negative impacts on other parts of the refrigeration system. An oil separator is therefore often mounted directly after the compressor outlet to reduce the flow of lubricant into the condenser and evaporator. Heat transfer will be impaired if oil droplets become trapped in these components. The refrigerant is protected from most of the oil, because the oil separator continuously returns lubricant to the crankcase of the compressor.

Partly miscible oils and refrigerants may separate in the condenser. If so, a refrigerant-rich phase is carried over to the expansion valve, while the oil accumulates in an oil-rich phase in the refrigerant reservoir. This may restrict the return of oil to the compressor, leading to insufficient lubrication. In the evaporator, the lubricant is subject to low temperatures, which may lead to problems with wax formation and phase separation. If the solubility of the refrigerant in oil at low temperatures is low, there may be problems with returning the oil to the compressor.

3.4 Compressor Performance

A change in the evaporation or the condensing temperature influences the operating conditions for the compressor. Any change in temperature affects the density of the refrigerant, which alters the compression ratio between the low-pressure and high-pressure sides. The influence of changes in the evaporation and condensation temperatures on compressor performance is discussed in this section.

High temperature in the evaporator is equivalent to high pressure and high vapor density. This means that 1 kg high-pressure vapor occupies less volume than 1 kg low-pressure vapor. In a refrigerant system, the mass flow of high-pressure vapor into the compressor is therefore larger at each displacement than the mass flow of low-pressure vapor. To maintain a specific suction pressure, i.e. to maintain a specific evaporating temperature, the evaporator must be designed to vaporize the same mass of refrigerant as is compressed in the compressor.

If the entering water temperature, **EWT**, and the leaving water temperature, **LWT**, increase e.g. 1 K from 12°C and 7°C, respectively, to 13°C and 8°C, respectively, the mean temperature difference, MTD, will increase (see Figure 3.5). Hence, a larger amount of refrigerant than before will evaporate in the evaporator. However, the compressor still removes the same amount of vapor as before the change in water temperature. Excess gas that is not removed by the compressor therefore remains inside the evaporator. The accumulation of excess vapor in the evaporator leads to higher pressure and temperature on the refrigerant side. The increased vapor pressure means the vapor density also increases. Consequently, a larger mass of refrigerant becomes compressed on every compressor stroke, i.e. the compressor capacity will increase if EWT and LWT increase 1 K. However, the evaporator and the compressor will subsequently find a new operating point where equal masses of refrigerant vapor are produced by the evaporator and removed by the compressor. Thus, when the conditions in a refrigerant system change, the compressor and the evaporator together will find a new operating point.

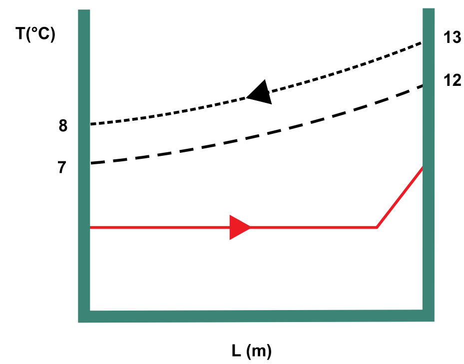


Figure 3.5 The MTD, i.e. the difference between the evaporation temperature profile and the water side temperature profile, becomes larger when the EWT and LWT increase 1 K.

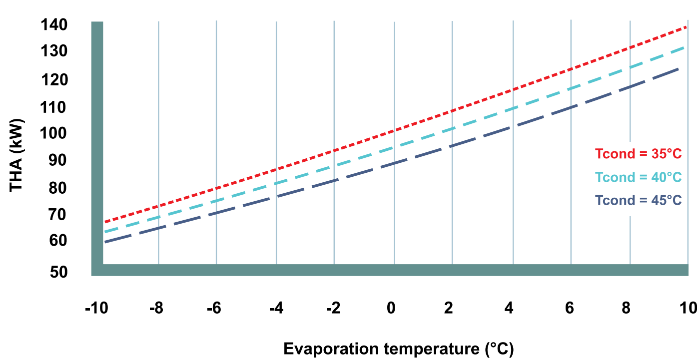


Figure 3.6 How change in evaporation temperature and condensation temperature affect the total heat of absorption for a specific compressor type.

Figure 3.6 shows three operating lines for a compressor at different evaporation temperatures but a constant condensation temperature for each compressor line. The compressor suction performance corresponds to a certain cooling capacity (**THA**) at each pressure ratio. Increasing the evaporation temperature at a constant condensing temperature leads to increasing compressor performance.

Just as every compressor has its own characteristic operating line, every evaporator has its own characteristic operating line. Figure 3.7 shows that the evaporator performance decreases when the evaporation temperature increases. One of the compressor lines (T_{cond} = 40°C) in Figure 3.6 is also plotted in Figure 3.7. The compressor operating line crosses every CBE operating line only once. The intersection point, marked with a circle in Figure 3.7, determines the evaporation temperature and thus the cooling performance of the specific compressor/CBE combination. This highlights the importance of matching the compressor and the CBE correctly to achieve the desired operating conditions.

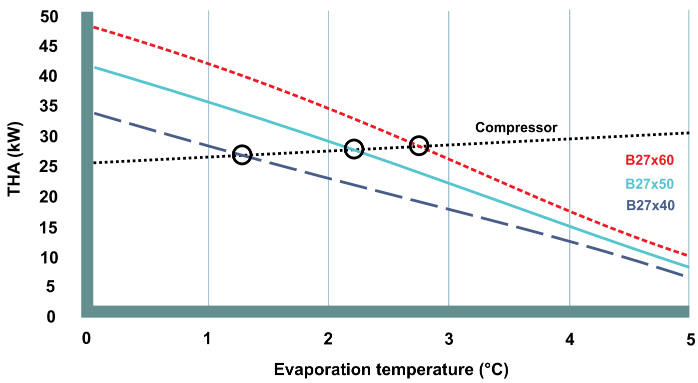


Figure 3.7 Operating points for three different CBE evaporator/compressor combinations at specific operating conditions.

The condensing temperature can also fluctuate for various reasons. One reason is that suction pressure differences can affect the pressure ratio of the compressor, which leads to an altered condensation pressure, i.e. a different condensation temperature. Other reasons could be changes in the flow or temperature of the cooling water to the condenser.

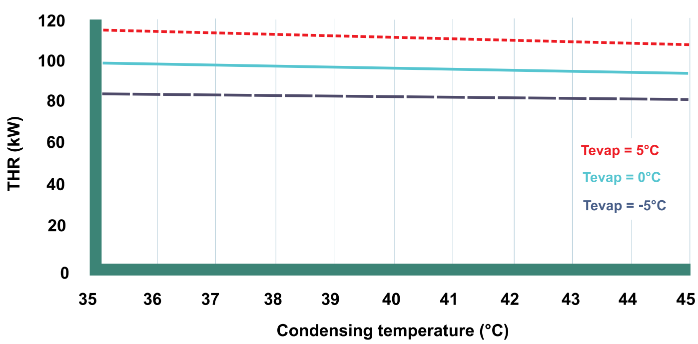


Figure 3.8 How change in evaporation temperature and condensation temperature affect the total heat of rejection for a specific compressor type.

Figure 3.8 is similar to Figure 3.6, with the differences that the characteristic operating line is plotted for constant evaporation temperatures and varying condensation temperatures. Note that the effect of increased condensing temperature on the compressor heating capacity (**THR**) is less than that of increased evaporation temperature on the compressor cooling capacity (THA) (see Figure 3.6). The compressor heating capacity decreases only slightly when the condenser temperature increases.

Figure 3.9 shows the operating points for three different SWEP condensers. It can be concluded from Figures 3.6 to 3.9 that a change in the evaporation temperature affects the cooling/condensing capacity more than a change in the condensation temperature. Thus, to maintain the designed total system capacity, it is more important to maintain the designed evaporation temperature than the designed condensation temperature.

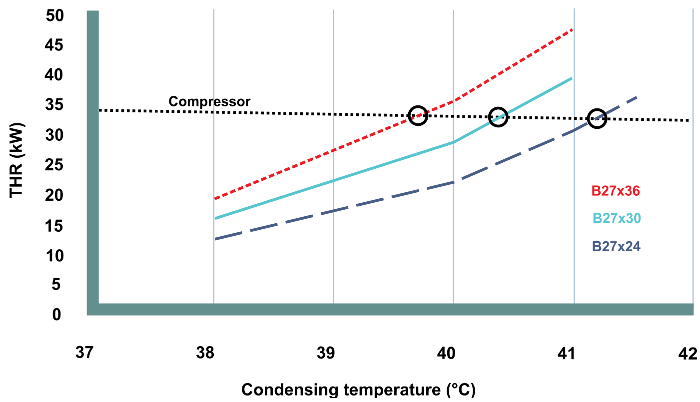
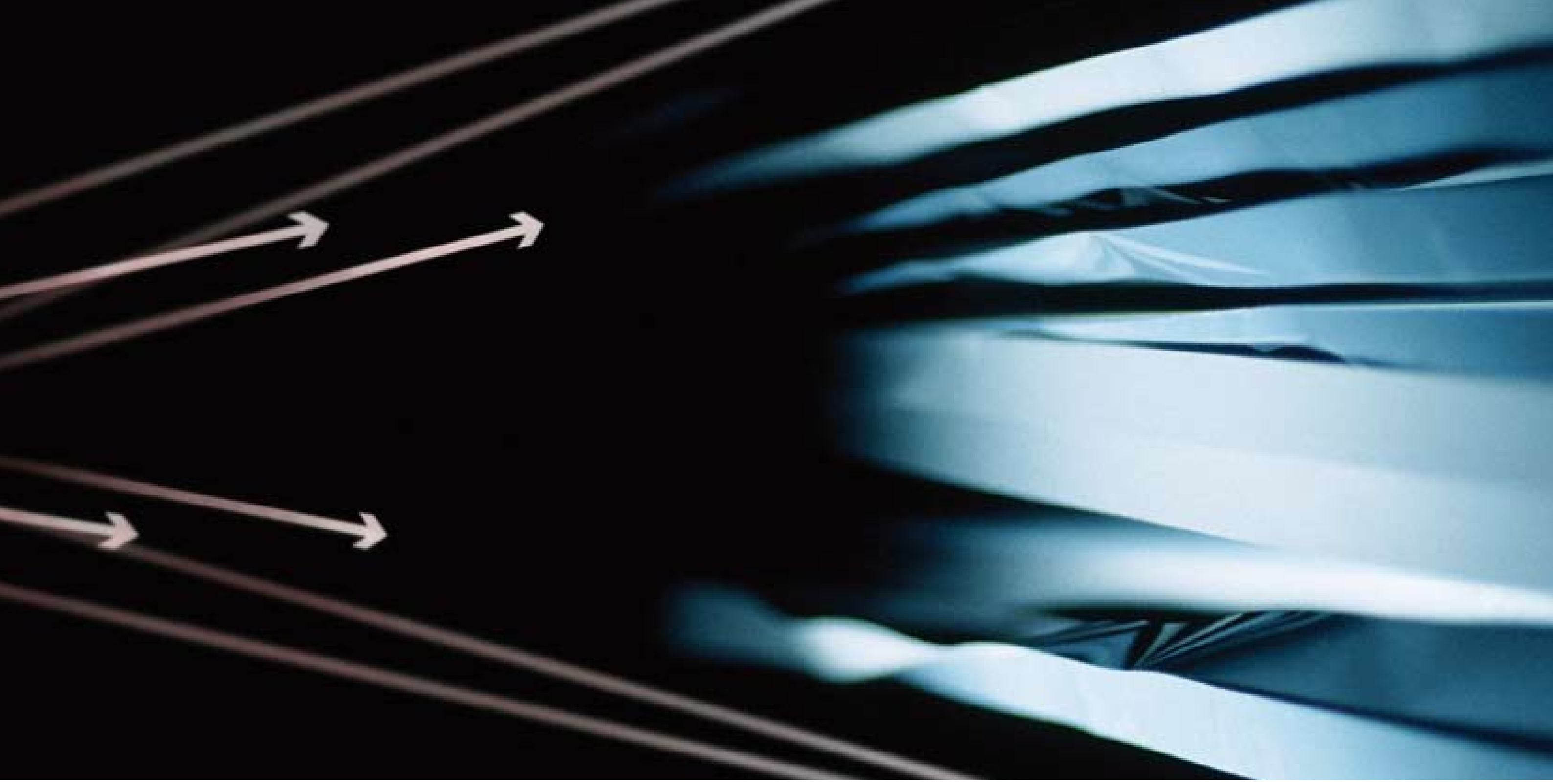


Figure 3.9 Operating points for three different CBE condenser/compressor combinations at specific operating conditions.

Table 3.3. Impact from changes in evaporation and condensation temperatures on cooling capacity (Q_2), or total heat of absorption (THA), condenser heating capacity (Q_1), or total heat of rejection (THR), and compressor power input (W). Also shown are the coefficients of performance for a chiller (COP_{REF}) and a heat pump (COP_{HP}). The table is valid for T_{EVAP}=2°C and T_{COND}=40°C.

The impact on:	**Change in operating conditions** **1°C decrease in evaporation temp.**	 **1°C increase in condensation temp.**
THA (Q_2)	-3.5%	-1%
THR (Q_1)	-3%	<-1%
W	<-1%	+2%
COP_{REF}	-3.5%	-3%
COP_{HP}	-3%	-2%

The values in Table 3.3 have been calculated in design software for compressors.



4 Expansion Valves

The expansion valve is situated in the liquid line between the condenser and the inlet of the evaporator. It operates on the opposite side of the system, relative to the compressor. Whereas the compressor operates to increase the pressure and pump the refrigerant through the system, the expansion device releases the pressure between the high-pressure condensation side and the low-pressure evaporation side.

There are numerous types of expansion valves, depending on the demand for control and the type of evaporator, i.e. flooded or direct expansion. This chapter discusses the characteristics of different expansion techniques and their suitability for CBE evaporators.

4.1 General Function and Theory

Expansion valves serve two purposes:

1. **Controlling the amount of refrigerant entering the evaporator:** As much of the evaporator surface as possible should be covered with liquid refrigerant without liquid being carried over to the compressor. If the capacity of the evaporator increases, the expansion valve should allow a larger flow of refrigerant, and vice versa. A smaller refrigerant mass flow results in a higher level of superheating, because less surface area is required for evaporation.
2. **Maintaining the pressure difference between the condenser (high pressure) and the evaporator (low pressure):** The pressure difference created by the work of the compressor is maintained by the expansion device.

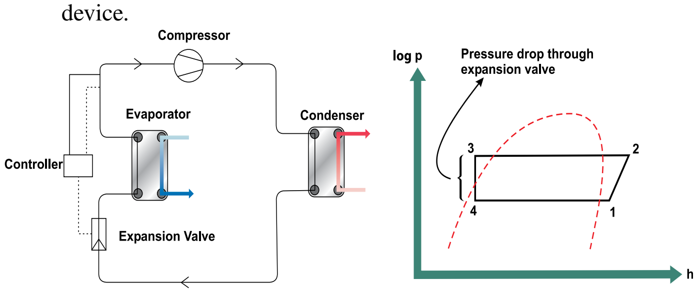


Figure 4.1 A refrigerant system as a flowchart and in a log P/h diagram.

Expansion valves do not directly control the evaporation temperature. Instead, they regulate the superheating by adjusting the mass flow of refrigerant into the evaporator, and maintain the pressure difference between the high-pressure and low-pressure sides. The evaporation temperature depends on the capacity of the compressor and the characteristics and efficiency of the evaporator. The function of an expansion valve is shown in Figure 4.1.

There are seven main types of expansion devices:

1. Thermal expansion valves (TEVs)
2. Manual valves
3. Capillary tubes
4. Automatic valves
5. Electronic expansion valves
6. Low-pressure float valves
7. High-pressure float valves

4.2 Thermal Expansion Valves (TEVs)

Thermal expansion valves, or thermostatic expansion valves, are the expansion devices used most commonly with CBE evaporators. TEVs are popular expansion devices due to their simplicity and availability, and their relatively good sensitivity and accuracy in regulation. The large choice of expansion valve sizes and bulb charges means the capacity and temperature ranges are very good. The disadvantage of TEVs is the necessity for relatively high superheating, which "steals" heat transfer area from the evaporation process.

The TEV strives to maintain a stable level of superheating inside the evaporator under all conditions by adjusting the mass flow of refrigerant in response to the evaporator load. This is achieved by a membrane inside the valve housing, which compares the temperature before and after the evaporator. To be able to compare the pressures before and after the evaporator, the TEV has to be combined with another device, a bulb. The difference in pressure between the saturation pressure of evaporation and the pressure of the bulb is balanced across a membrane inside the head of the valve. Movement of the membrane controls the position of a needle and hence the mass flow of refrigerant entering the evaporator. The components of a TEV and a bulb are shown in Figure 4.2. The function of the bulb is shown in Figure 4.3. The bulb, which transmits the corresponding pressure of the superheated gas, consists of a hollow metal container filled with a refrigerant fluid. A capillary tube connects the bulb to the valve housing. The bulb is fitted in direct contact with the suction pipe, close to the compressor inlet.

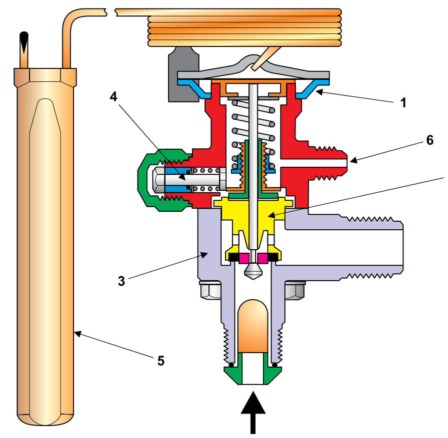


Figure 4.2 A thermostatic expansion valve. (1) Membrane housing, (2) Interchangeable adapter, (3) Valve housing, (4) Spindle for adjusting static superheat, (5) Refrigerant-filled bulb, (6) Port for external equalization.

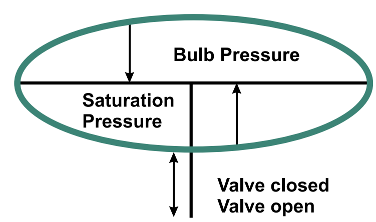


Figure 4.3 The effect of the bulb on the membrane inside the TEV.

If the superheating increases, the pressure inside the bulb will increase, because more refrigerant inside the bulb evaporates. The increased pressure is transmitted through the capillary tube, and depresses the membrane inside the head of the TEV. This moves the needle, opening the valve orifice and thus increasing the refrigerant mass flow. The balance across the membrane is adjusted with a spring that may be adjustable manually or set at the factory. The stiffer the spring, the higher the level of superheating required to open the valve.

If the saturation pressure increases instead, the bulb will still detect an increased temperature. More refrigerant will boil off, but the increased pressure below the membrane will balance the higher pressure above the membrane. Thus, there will be no change in the needle position.

The following example illustrates the bulb's balancing effect on a TEV system. An increased mass flow of refrigerant requires more heat surface area to evaporate, and therefore results in less superheating. The outlet gas temperature will therefore decrease. This in turn will cool the bulb, resulting in the condensation of some bulb refrigerant and thus decreasing the pressure on the membrane. The force of the spring will close the valve slightly, and less refrigerant will be allowed into the evaporator, again increasing the superheating. The system will soon find a balance.

There are three different types of bulbs:

- Liquid-charged bulbs
- MOP (Maximum Operating Pressure) bulbs – also called gas-charged bulbs
- Adsorption-filled bulbs

A **liquid-charged bulb** has a large charge of refrigerant and will never "run dry". It will always contain both liquid and gaseous refrigerant. The pressure inside the bulb increases as the superheating increases, due to additional evaporation. Historically, the refrigerant in the bulb was the same as the working refrigerant in the system (parallel-charged). However, better characteristics have been achieved by using different refrigerants (cross-charged), which is now the most common arrangement.

An **MOP bulb**, also called gas-charged, has a much smaller quantity of refrigerant mixture inside the bulb than a liquid-charged bulb. As the evaporation pressure increases, the suction pipe will become increasingly warm as a result. A limited refrigerant charge in an MOP bulb will be totally evaporated at a predefined pressure, the MOP pressure. When the liquid refrigerant mixture has boiled off, the pressure inside the bulb will not increase greatly even if the evaporating pressure does. The needle valve will not open further, thus limiting the maximum mass flow through the valve. The reason for this is to protect the compressor from electrical overload, especially during start-up when the evaporation pressure can be much higher than under normal operating conditions. A disadvantage of the MOP valve is that the bulb always has to be colder than the valve housing to prevent the limited refrigerant charge from migrating and condensing at the membrane surface. If the MOP bulb were instead warmer than the valve housing, the MOP valve would close even if the operating pressure were well below the maximum operating pressure.

TEVs may also have an **adsorption charge,** where the bulb also contains a solid adsorbent such as charcoal or silica gel. The adsorbed refrigerant reacts more slowly to temperature changes than direct-charged bulbs, and gives a slower response. This can sometimes help to stabilize oscillation tendencies. However, adsorption-filled bulbs work best over a limited range, which is why they are often specially designed for the operating conditions.

Adjusting the Superheating

Superheating is the energy added to saturated gas, resulting in a temperature increase. During the evaporation of a liquid refrigerant, the temperature depends only on the boiling temperature of that refrigerant. Increasing the temperature (superheating) is possible only after obtaining 100% vapor.

The spring inside the TEV acts on the needle to keep the valve closed if the TEV detects insufficient superheating. A minimum level of superheating is required to allow the pressure from the bulb to start pushing back the spring and thus opening the valve. This is called the **static superheating** (B-C in Figure 4.4). The spindle on the side of the expansion valve regulates the static superheating. A loose spring gives less static superheating, because the valve opens earlier. A stiff spring requires more static superheating, because the valve opens later. The valve curve in Figure 4.4 is shifted to the left with less static superheating and to the right with more static superheating.

The additional superheating required to open the valve for operation is called the **opening superheating,** and should be optimized for the **nominal operation point** of the system (C-D in Figure 1.4). The opening superheating is determined by the construction of the TEV and cannot be altered in the system. Adding the static and opening superheating gives the **working superheating**, which is the real superheating that can be measured in the system. The expansion valve is normally slightly oversized, and will reach maximum capacity when it is fully opened. This can be achieved only with higher operating superheating (point D in Figure 4.4).

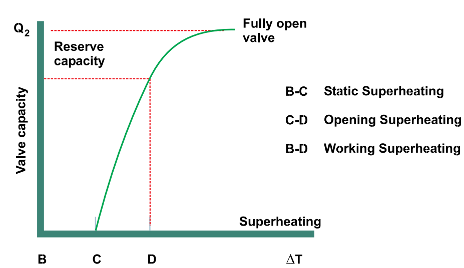


Figure 4.4 The correlation between the amount of capacity being used and the amount of superheating this causes in the evaporator.

Increasing or decreasing the working superheating for a system can be accomplished only by altering the static superheating, as shown in Figure 4.5. The performance curve of the TEV will then shift to the right or left with more or less static superheating, respectively.

The maximum refrigerant flow through a TEV depends on the size of the valve and the pressure difference over it. If the valve is too small, the nominal refrigeration capacity cannot be reached even if the valve is fully open. The TEV is normally selected to allow a slightly higher (approximately 20%) refrigeration capacity than the nominal. However, to open the valve maximally, a higher bulb pressure is needed to push back the spring. The reserve capacity is therefore used at the expense of increased working superheating.

Influence of Superheating on Evaporation Temperature

The evaporation temperature is influenced by the operation of the expansion valve due to the change of superheating and mass flow, as shown in Figure 4.6. Small changes in superheating have only a small impact on the evaporation temperature. However, a very high level of superheating will cause a large decrease in the evaporation temperature. In this situation, the suction gas temperature will approach the inlet water temperature, and a large part of the heat transfer surface must be utilized for the superheating (shown at D in Figure 4.6).

Stability

It is important to adjust the superheating to a suitable level. If the superheating is too small, it may cause instability in the evaporator (**hunting**). Excess liquid refrigerant may flow over and enter the compressor where it can cause problems such as **foaming**, where droplets of refrigerant enter the oil sump and are immediately evaporated. The turbulence may create a refrigerant/oil foam that disrupts operation. Excess liquid refrigerant that enters the compressor may also create pressure shocks inside the compression chamber. Liquid refrigerant droplets splashing onto the shaft will dissolve in the oil and decrease the lubrication effect. Eventually, the shaft bearing may be exposed and wear rapidly. These factors may reduce the life expectancy of a compressor considerably, although the sensitivity to liquid refrigerant carry-over varies greatly with different compressor techniques. If the superheating is too high, on the other hand, it will lead to high compressor discharge temperatures, decreasing the life span of the oil. Furthermore, a high level of superheating will require an unnecessarily large heat transfer area and/or depress the evaporation temperature, resulting in a decreased COP.

Thermal expansion valves must always operate with a minimal working superheating to achieve stable regulation. The minimum stable signal (MSS) depends on the type of TEV, the characteristics of the evaporator present and the mutual positions of the expansion valve, the evaporator and the bulb. It is therefore difficult to predict the minimum superheating for stable operation. In practice, the adjustment is made by starting operation with a stable superheating and then systematically loosening the spring until instability occurs. Choosing a slightly higher level of superheating will assure stable operating conditions.

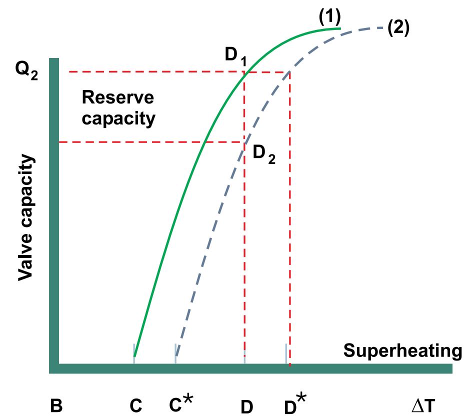


Figure 4.5 The valve characteristics before and after changing the static superheating by tightening the regulating screw of the valve.

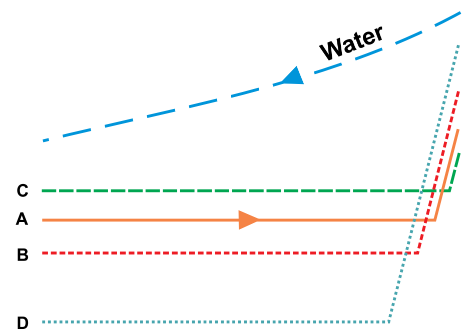


Figure 4.6 Temperature profiles for different levels of superheating. A: Normal superheating. B: High superheating. C: Low superheating. D: Very high superheating.

In Figure 4.7, valve 1 is too large. With just a small increase in the superheating, the needle will open to allow a large volume of refrigerant to pass. The feedback signal is too strong relative to the increase in superheating, and the system may become unstable. It would be possible to stabilize the valve by increasing the static superheating, because the TEV line would then shift to the right, away from the MSS line. However, this would require a higher working superheating, which would lower the evaporation temperature and reduce the operating economy. Valve 2 is perfect, touching the MSS line at exactly the point of nominal load with a reasonable level of superheating. Valve 2 may still open slightly more to allow for a momentary capacity increase. Valve 3 is too small, because it can deliver the nominal duty only with increased superheating. There is also no extra capacity potential.

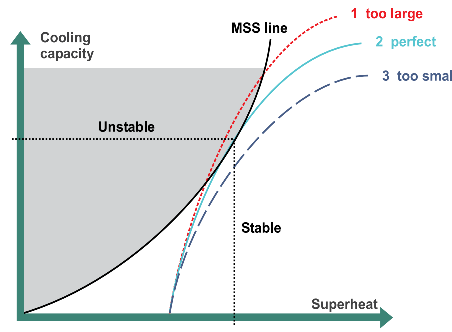


Figure 4.7 The influence of valve size on stability.

External and Internal Pressure Equalization

To estimate the level of superheating correctly, the temperature of the refrigerant gas should be compared with the saturation temperature at the same point, i.e. the evaporator outlet. The internally equalized expansion valve compares instead the temperature measured in the bulb at the outlet of the evaporator with the pressure just after the expansion valve.

If there is a large pressure drop between the expansion valve and the measuring point of the bulb (e.g. if there is a distribution device in the evaporator), the difference between the measured temperature and the estimated saturation temperature will be too small. The valve will overcompensate by closing, resulting in an unreasonably high level of superheating and an excessively large proportion of the evaporator surface being used for superheating the refrigerant. At the nominal cooling capacity, the superheating will be too high and will thus occupy too much of the heat transfer area to maintain the set evaporation temperature. The total system performance will decline as a result.

The following example illustrates the discussion above. A SWEP evaporator with a distribution device is used with a TEV with internal pressure equalization. The following is assumed: 1 bar pressure drop over the fluid distributor rings, T_{EVAP}=2°C and the TEV is pre-set at 5K superheating. The pressure immediately after the expansion valve (state ‘a’ in Figure 4.8) corresponds to a saturation temperature of approximately 7°C. Trying to adjust to 5K superheating, the temperature of the gas leaving the evaporator will be 7+5=12°C (state ‘c’). This results in a superheating of 10K instead of the intended 5K (state ‘d’).

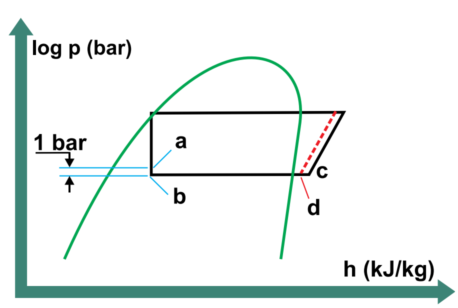


Figure 4.8 Pressure drop caused by a fluid distribution system in the evaporator.

To operate the TEV with the correct measurement of the superheating for evaporators with a high pressure-drop on the refrigerant side (i.e. with distribution devices), it is necessary to use a TEV with **external pressure equalization**. An additional pressure tube is then installed at the evaporator outlet downstream of the bulb in the suction line. The static pressure is fed back to the expansion valve, acting on the opposite side of the membrane from the bulb. Because of this extra pressure equalization from the outlet of the evaporator, the pressure from the bulb will give a response only to the actual superheating.

As can be seen in the example above, it is particularly important to use this type of expansion valve with a SWEP evaporator with built-in fluid distributor rings, which cause an additional pressure drop of normally 0.5-2 bar. The pressure drop over the SWEP refrigerant distribution system is not considered to be part of the CBE pressure drop. The expansion valve and the distribution system operate together, creating the total pressure drop between the condensing and evaporating pressure levels.

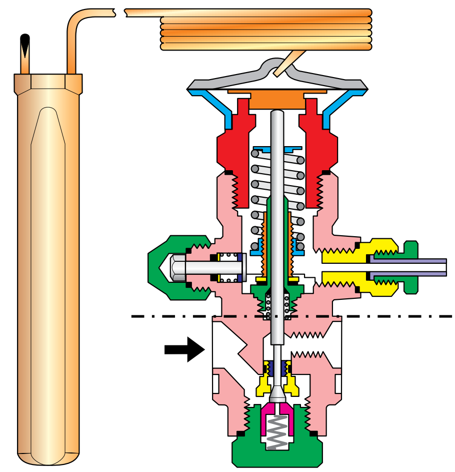


Figure 4.9 Externally equalized thermostatic expansion valve.

Sizing the Thermal Expansion Valve

The expansion valve is selected to ensure that a sufficient mass flow of refrigerant can pass while maintaining a stable and reasonable superheating for regulation of the evaporator. The TEV should function over the full operating range of the system without instability or capacity problems. Different manufacturers use different recommendations, but a TEV size allowing 20% over the nominal capacity is normally selected.

Important data to take into account when selecting an expansion valve are the nominal cooling capacity, the condensation pressure and the evaporation pressure. If the evaporator is equipped with a distribution device, its approximate pressure drop should also be noted. The required pressure drop over the TEV is calculated as follows.

(1) $$\Delta P_{TEV} = P_{Co} - P_{Ev} - \Delta P_{Sys}(-\Delta P_{Dist})$$

Where:

ΔP_{TEV} = Designed pressure drop over the TEV

P_{Co} = Condensation pressure

P_{Ev} = Evaporation pressure

ΔP_{Sys} = Pressure drop over other system components

ΔP_{Dist} = Pressure drop over other distribution system

Programs or selection tables from expansion valve manufacturers can be used to find a suitable size based on the calculated pressure drop over the TEV and the nominal refrigeration capacity. Note that the extra pressure drop over the distribution system, ΔP_{Dist}, will decrease the actual pressure difference over the TEV, which often means that a larger expansion valve or nozzle insert is required.

The expansion valve is pre-set with a static superheating based on a nominal condensing pressure. If the system condensing pressure is higher, the pre-set static superheat will become lower and vice versa. The nominal capacities of the TEVs are displayed in selection tables for the pre-set static superheating and a nominal level of subcooling. Again, if the subcooling of the system exceeds the nominal value, the tabulated cooling capacity should be divided by a correction factor. Always read the technical information from the TEV manufacturers and follow their instructions.

The nominal cooling capacity of the evaporator is determined by the balance between compressor and evaporator performance.

The most complicated selection is when the system operating conditions change considerably with fluctuating condensation and evaporation pressures or with large variations of cooling capacity, i.e. multiple or variable compressors. The selection of an expansion valve is normally a compromise, because no valve will be ideal for the whole operating range. However, a correctly selected valve will give a wide operating range.

4.3 Manual Throttles

The manual throttle, or hand expansion valve, is a manually operated needle valve (see example in Figure 4.10). The needle position is fixed, and the mass flow through it depends on the pressure difference across the valve. The hand expansion valve is normally used as a supplementary safety valve installed in a bypass line. It is also commonly used to control the flow rate through oil bleeder lines. The manual throttle is a non-regulating valve and should not be used as an expansion valve with a CBE evaporator, because any changes in operating conditions would instantly change the evaporation process inside the rapidly responding plate heat exchanger. Regulating the flow manually would require immediate adjustment, which is not practically feasible.

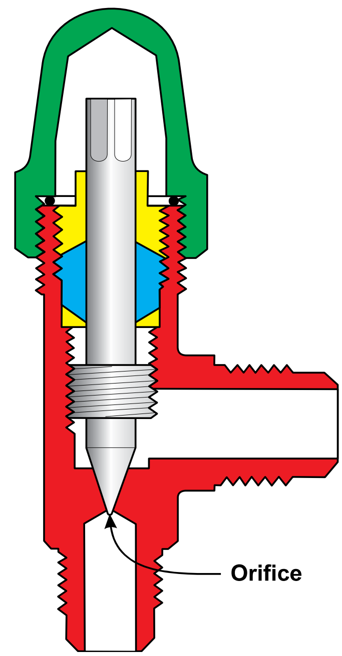


Figure 4.10 The hand expansion valve.

4.4 Capillary Tubes

Capillary tubes are the simplest of all refrigerant flow controls, with no moving parts. They normally consist only of a copper pipe, diameter 0.5 to 1.5 mm and length 1.5 to 6 m. The expansion function is caused simply by the pressure drop induced by the long, narrow tube. The mass flow through the tube depends on the pressure difference between the condensing and evaporating sides.

Capillary tubes can be found on small, high-volume commercial systems such as household refrigerators, but can also be used for larger systems if the operating conditions are relatively stable. The capillary tube is vulnerable to clogging, which is why a filter drier and filter are normally mounted before the inlet. A refrigeration system with a capillary tube installed is shown in Figure 4.11.

The low-pressure side of a refrigerant system with a capillary expansion device must be able to hold the whole refrigerant charge. When the compressor stops, the refrigerant will migrate to the cold, low-pressure side. Often, the low-pressure side is equipped with a liquid separator, which acts as a receiver, just before the compressor.

The refrigerant charge must also be carefully considered for capillary tube systems. An overcharged system will back up condensate into the condenser. This will eventually flood the condenser totally if the overcharge is sufficiently large or if there is a large change in operating conditions. Undercharge, on the other hand, will result in starvation of the evaporator, with hunting as a result.

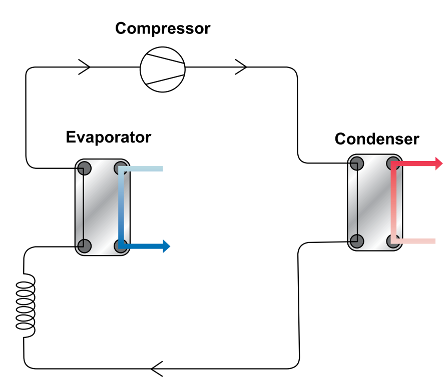


Figure 4.11 The refrigerant cycle with a capillary tube installed.

4.5 Automatic Valves

Automatic valves operate with a constant counter-pressure that regulates the mass flow of refrigerant through the valve to obtain a constant evaporation pressure, as shown in Figure 4.12. The automatic valve, or constant pressure valve, can be used in systems with small variations in heat load and a maximum of one evaporator. The technology is old-fashioned, and is now often replaced by the more advanced thermal expansion valve.

An automatic valve could be used successfully with a CBE evaporator. However, the difference in cost compared with a thermal expansion valve would be less than the cost of the extra heat surface area in the CBE evaporator that would be needed to compensate for variations in superheating.

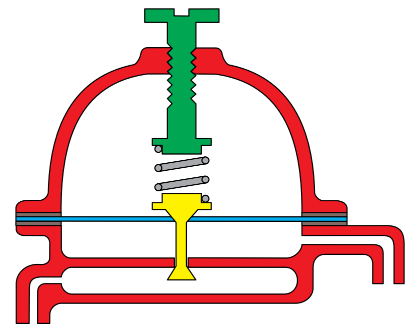


Figure 4.12 The automatic expansion valve (constant pressure valve).

4.6 Electronic Expansion Valves

Electronic expansion valves have been increasing in popularity, and the technology has evolved to cope with increasingly sophisticated demands. However, the cost of electronic valves, which includes the sensors, regulator, actuator and the valve itself, is still much higher than for the simple and mechanical thermal expansion valve. Electronic expansion valves are therefore mostly found on very large systems and systems with a high demand for precise regulation. Two different models of electronic valves are discussed in this section: **modulating electronic expansion valves**, with a continuously adjusting orifice and **electronically controlled ON/OFF valves**, with a solenoid valve that is opened and closed periodically.

Modulating Electronic Expansion Valves

Modulating electronic expansion valves are controlled by temperature or pressure sensors. The electronic regulating unit can be programmed to correct for differences in temperature and pressure at any point of the system. Because the electric actuator reacts only to signals from the regulator, there are good possibilities for achieving a lower level of superheating than with a thermal expansion valve. The signal follows the MSS signal continuously, because any hunting tendencies will be compensated immediately. Pressure differences between the valve and the sensor caused by refrigerant distribution systems are also corrected. Furthermore, the same valve can be used for different refrigerants after reprogramming. The electric actuator controls the shutter to adjust the orifice area continuously to allow a higher or lower mass flow of refrigerant to pass, depending on the signals from the regulator.

This type of valve can handle large variations in operating conditions, e.g. changes in pressure difference and cooling capacity. The biggest disadvantage of electronic valves is the relatively high cost and complexity of components. Programming the regulator box is not trivial, and system performance with a poorly adjusted electronic expansion valve may very well be lower than with a thermal expansion valve.

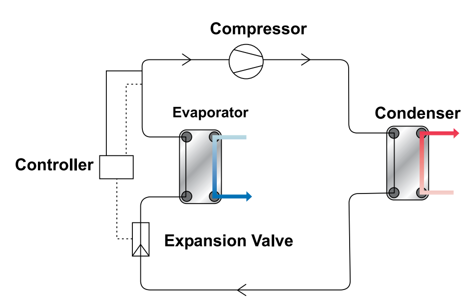


Figure 4.13 Electronically controlled expansion valve (continuous control).

Electronically Controlled ON/OFF Valves

The electronic ON/OFF valve is actually an electronically controlled solenoid valve that functions both as an expansion valve and as a solenoid valve. When functioning as an expansion valve, ON/OFF control is used. During one cycle period, typically 6 seconds, the valve is opened and closed once. This type of valve is normally not recommended for use with CBE evaporators, because the sudden surge involved when the solenoid valve opens will create thermal and mechanical stresses in the evaporator that can lead to a reduced life expectancy. A system with an electronically controlled ON/OFF valve is shown in Figure 4.14.

The capacity of the expansion valve, i.e. the amount of refrigerant flowing through it, is determined by the relationship between the opening and closing times. A regulator controls the opening and closing of the valve in order to reach the correct level of superheating. The inputs to the regulator are the temperature and pressure at the evaporator outlet. The inputs could also be the inlet and outlet evaporator temperatures, as for an electronic valve with continuous control.

When the demand for refrigerant is high (high cooling capacities), the valve remains open for almost the entire 6 seconds. When the demand is very low (low cooling capacities), the valve opens only for a fraction of the 6 seconds. When the compressor is shut off, the valve closes and functions as a solenoid valve. The variations in opening and closing times over the cycle periods are shown in Figure 4.15.

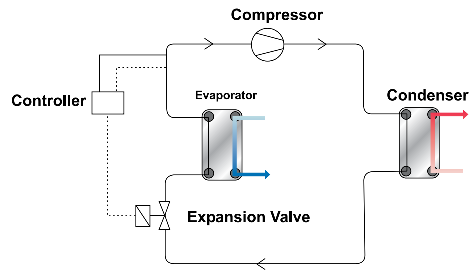


Figure 4.14 Electronically controlled expansion valve, ON/OFF type.

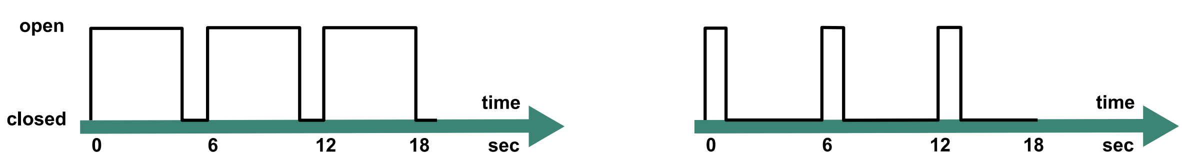


Figure 4.15 Opening/closing time for high demand (left) and low demand (right).

In a system where a CBE condenser is used and no receiver is installed in the liquid line, the on/off operation of this type of valve can cause problems. This type of system is shown in Figure 4.14. Unless enough refrigerant is charged in the system, there is a risk of the condenser draining during the opening period and a gas/liquid mixture being fed to the valve. This will cause a decrease in the valve capacity, i.e. the refrigerant flow will decrease, and insufficient refrigerant will pass through. The result may be system hunting, or excessive gas superheating. To prevent this potential problem, the system must be charged with a sufficient amount of refrigerant.

The electronic ON/OFF valve can operate satisfactorily even with large variations in operating conditions, such as changes in pressure difference or cooling capacity. The valve capacity is adjusted simply by changing the relation between the opening and closing time. This allows the minimum stable superheating to be found for a wide range of operating conditions.

4.7 Low-Pressure Float Valve

The means of expansion for **flooded evaporators** differ from those for **direct expansion (DX) systems** (see chapter 6, Evaporators). Most common are refrigerant flow controls of the float type. A refrigeration

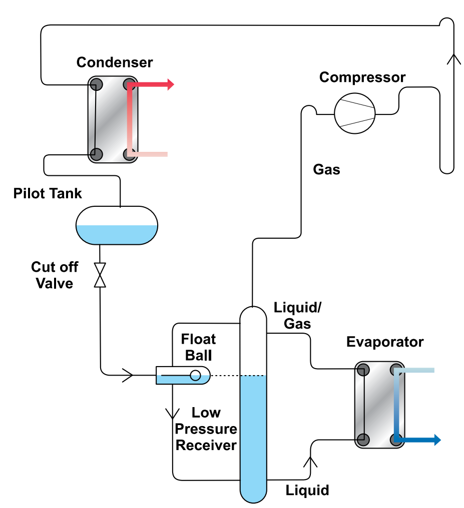


Figure 4.16 Refrigerant system with a low-pressure float control valve.

system with a **low-pressure float valve** is shown in Figure 4.16. Figure 4.17 shows the details of a low-pressure float valve. The low-pressure float valve controls the liquid level, and is normally mounted in a chamber parallel to the liquid/vapor separator. For **thermosiphon systems**, this modifies the effect of the force of gravity, which drives the refrigerant into the evaporator. If the level of refrigerant in the separator increases, the valve will close and vice versa. Balance is maintained when the refrigerant flow into the evaporator is equal to the vapor flow from the evaporator. For **forced circulation systems**, the circulation pump will control the degree of evaporation.

Instead of using a mechanical connection between the float and the needle, it is possible to connect the float to an electronic switch that controls the flow via solenoid valves. The function is the same as described above.

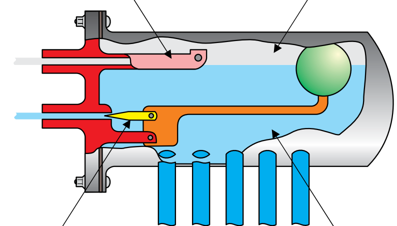


Figure 4.17 Low-pressure flow valve.

4.8 High-Pressure Float Valve

Also used as an expansion device for flooded systems, the high-pressure float valve is located on the high-pressure side of the system and is in open connection to the condenser. It controls the evaporator level indirectly by maintaining a constant level of refrigerant inside the float chamber. The evaporator level therefore depends on the total refrigerant charge of the system, and must be adjusted to the system. If the charge is too great, it will lead to excessive flooding of the evaporator, while if the charge is too small it will lead to starvation. The function of a high-pressure float valve is shown in Figure 4.18.

The mechanical high-pressure float valve is now often replaced with an electronic alternative where the float provides impulses to control a solenoid valve.

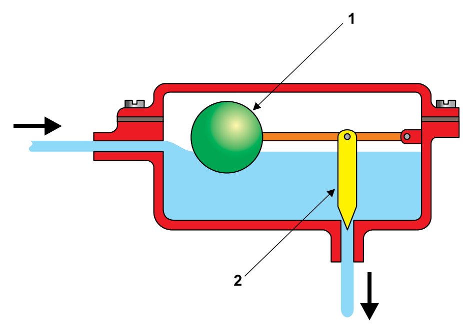


Figure 4.18 High-pressure float valve. The float (1) floats on the high-pressure liquid and the needle forms a constriction (2). The float chamber is in open connection with the condenser, and the condensed refrigerant enters the chamber continuously. When the liquid level increases, the needle opens the flow to the evaporator.

5 Refrigerants

This chapter studies the properties of substances used as working fluids in compression cycles, and discusses the classification of refrigerants. Some of the most common refrigerants are presented, as well as the special phenomena occurring with the use of zeotropic refrigerant mixtures. Finally, the use of secondary refrigerants and oil in refrigerant are discussed.

5.1 Refrigerant Criteria

A working fluid in a compression refrigeration system must satisfy a number of requirements that can be divided into two groups:

1. The refrigerant should not cause any risk of injuries, fire or property damage in case of leakage.
2. The chemical, physical and thermodynamic properties of the refrigerant must suit the system and the working conditions at a reasonable cost.

The criteria can be specified more precisely as follows:

Chemical:	• Stable and inert
Health, safety and environmental:	• Non-toxic • Non-flammable • Benign to the atmosphere, etc.
Thermal:	• Critical point and boiling point temperatures appropriate for the application • Low vapor heat capacity • Low viscosity • High thermal conductivity
Other:	• Satisfactory oil solubility/miscibility • High dielectric strength of vapor • Low freezing point • Reasonable containment materials • Easy leak detection • Low cost

It is not possible to fulfill all the requirements above at the same time. The most important criterion is chemical stability within the refrigeration system. All the other criteria are meaningless if the refrigerant decomposes or reacts with the materials used in the system. However, the chemical criterion can also be a problem. When a refrigerant is emitted to the atmosphere, it should not be so stable that it can exist indefinitely. The ideal refrigerant would be totally stable in use within the system, but would decompose easily in the atmosphere without the formation of any harmful substances.

The thermodynamic and transport properties determine the performance of the refrigeration system. The pressure should be neither too high nor too low. A pressure below atmospheric is undesirable, because air can then be sucked into the system, resulting in problems such as inert gases in the condenser or ice plugs in the expansion valve. The **latent heat** multiplied by the vapor density gives an estimate of the capacity of a refrigeration system. A refrigerant with a high pressure gives a higher capacity, but a refrigerant with a low critical pressure compared with the pressure in the system leads to a low coefficient of performance (COP). This trade-off between high capacity and high efficiency must be considered when choosing the refrigerant. Another desired criterion is that the specific heat should be small compared to the latent heat of vaporization, in order to achieve small losses in the expansion device.

Oil is normally present in a refrigeration system, and the interaction between the oil and the refrigerant must be considered. High oil solubility is used in hermetic compressors, but immiscible oils are normally used when the working fluid is ammonia.

5.2 Designation of Refrigerants

According to an international agreement, refrigerants are represented by the letter R (as in Refrigerants) followed by a two- or three-digit number and, in some cases, one or two letters. The designation Rxyz is determined by the chemical composition of the molecule, as described below.

The methane, ethane and propane series:

Here, (x) gives the number of carbon atoms in the chemical formula, minus one.

- (x) = 0 is the methane series, but the 0 is ignored for these compounds. Examples are R12 and R22.
- (x) = 1 is the members of the ethane series, such as R114, R124 and R134a.
- (x) = 2 is the propane series, e.g. R290 (propane).

For these groups, (y) describes the number of hydrogen atoms plus one and (z) describes the number of fluorine atoms.

Zeotropic and azeotropic mixtures

- (x) = 4 refers to zeotropic mixtures. The components in the mixture have different boiling points, and thus the refrigerant mixture has a temperature glide. R407A and R407C are examples of such refrigerants (see section 5.7).
- (x) = 5 refers to azeotropic mixtures. These act like homogeneous substances with one specific boiling point, and therefore they have no glide. R502 and R507 are examples of azeotropic mixtures (see section 5.7).

Here, (y) and (z) are ordinal numbers.

High organic compounds

- (x) = 6 means that the composition is organic, e.g. butane, R600, and isobutene, R600a. This group has several subgroups, for example hydrocarbons, oxygen compounds, sulfuric compounds and nitrogen compounds.

The subgroups have been assigned different number series within the main group, so (y) and (z) describe the subgroup and order within the subgroup.

Inorganic compounds

- (x) = 7 refers to inorganic compounds, such as ammonia, R717, water, R718, and carbon dioxide, R744.

In this group, (y) and (z) are the molar mass.

Unsaturated organic compounds

- (x) = 11 stands for unsaturated ethane compounds, such as R1150 (ethylene).

- (x) = 12 stands for unsaturated propane compounds, such as R1270 (propylene).

The (y) and (z) are the same as for the ethane and propane series.

Letters at the end

The last letter, if any, in the designation number means different things:

- Lower-case letters describe the structure of the molecule. For example, R600 is butane and R600a is isobutane. These two compounds have the same chemical formula, but different spatial arrangements, and they therefore have slightly different properties.
- Capital letters describe specific mixing proportions of different components. For example, R407 A-E are mixtures of the refrigerants R32, R125 and R134a. R407A has the following mixing proportions: 20% R32, 40% R125 and 40% R134a, while R407C consists of 23% R32, 25% R125 and 52% R134a.

5.3 Environmental Impacts of Refrigerants

In addition to being toxic or explosive and thereby dangerous to people's health, there are other problems associated with refrigerants. Environmental aspects are increasingly being taken into consideration. Refrigerants can thus also be ranked according to their impact on the stratospheric ozone layer (the Ozone Depletion Potential, **ODP**) or as greenhouse gases (the Global Warming Potential, **GWP**).

Ozone Depletion Potential, ODP

Refrigerants containing chlorine or bromine contribute to the breakdown of the ozone layer (see Figure 5.1). The reaction is as follows:

$$Cl + O_3 \text{ -> } O_2 + ClO$$

However, the CIO molecule is unstable. It breaks down and reacts with ozone molecules (in accordance with the equation above) repeatedly until a more stable compound is created.

The ODP is the ratio of the impact on ozone of a chemical compared with the impact of a similar mass of **CFC-11** (R11). Thus, the ODP of CFC-11 is 1.0 by definition. Other CFCs and **HCFCs** have ODPs ranging from 0.01 to 1.0. The **halons** have ODPs ranging up to 10. Carbon tetrachloride has an ODP of 1.2, and methyl chloroform's ODP is 0.11. **HFCs** have zero ODP because they do not contain chlorine.

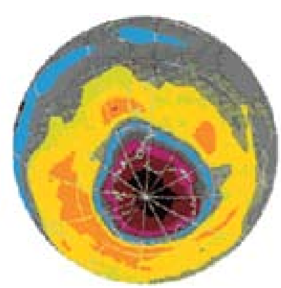


Figure 5.1 The image recorded by the American satellite Nimbus-7 on 3 October 1990 already shows a severe reduction in the ozone layer over the South Pole.

Global Warming Potential, GWP

Due to their stability in the atmosphere, CFCs as well as HCFCs and HFCs are often very effective greenhouse gases. The GWP factor is used to reflect their impact on global warming.

The GWP is the ratio of the warming caused by a substance to the warming caused by a similar mass of carbon dioxide. Thus, the GWP of CO_2 is 1.0 by definition. CFC-12 has a GWP of 8500, while CFC-11 has a

GWP of 5000. Various HCFCs and HFCs have GWPs ranging from 93 to 12100. Water, a substitute in numerous end-uses, has a GWP of 0.

When using GWP values from different sources, it is important to consider that the values may differ due to different integration times or calculation models.

Another measurement of the impact on global warming is the **TEWI** value (Total Environmental Warming Impact). This includes not only the direct impact of any release of the refrigerant (GWP), but also the impact during the generation and use of primary energy in the system.

5.4 Refrigerant Types

Refrigerants are divided into groups according to their chemical composition. Following the discovery that some of these chemical compounds may be harmful to the environment, they are being replaced with more environmentally friendly alternatives (see Figure 5.2). The process is not easy, and although there are alternatives to old refrigerants, the new ones are usually not flawless.

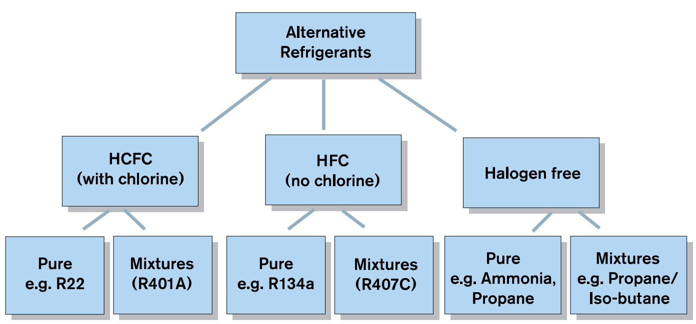


Figure 5.2 Alternatives to the "old" refrigerants.

In the following section, different groups of refrigerants are discussed, some examples are given and their fields of application are described.

CFC = ChloroFluoroCarbons

Chlorofluorocarbons are refrigerants that contain chlorine. They have been banned since the beginning of the 90's because of their negative environmental impacts. Examples of CFCs are R11, R12 and R115. The conversion of equipment and systems using CFCs has not yet been completed. On the contrary, the illegal market for this type of refrigerants flourishes worldwide, and it is estimated that no more than 50% of CFC systems worldwide have been upgraded.

HCFC = HydroChloroFluoroCarbons

The slow phase-out of CFCs shows it is a costly process. However, and more importantly, it also shows the problems and indecisiveness surrounding the availability of HCFCs, which were officially indicated as temporary (until 2030) substitutes for CFCs. The hasty actions of the European Union that culminated in the ban of HCFCs, immediately for refrigeration and soon (2004 at the latest) for air conditioning, has upset the industry's programs and plans.

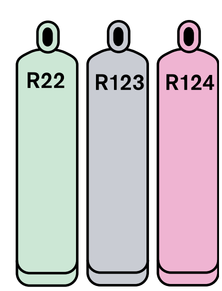


Figure 5.3 Some HCFC refrigerants.

The HCFCs contain less chlorine than CFCs, which means a lower ODP (see section 5.3). Examples of hydrochlorofluorocarbons include R22, R123 and R124 (see Figure 5.3).

HFC = HydroFluoroCarbons

The hydrofluorocarbons are refrigerants that contain no chlorine and are not harmful to the ozone layer (ODP = 0, see section 5.3). However, their impact on global warming is very large compared with traditional refrigerants. The most common HFC refrigerants available since the ban on HCFCs are presented in Table 5.1 (see also Figure 5.4):

Table 5.1 The most common refrigerants among halogenated hydrocarbons.

Refrigerant	Series	Safety	Glide
R32	Methane series	A1	
R125	Ethane series	A1	
R134a	Ethane series	A1	
R245ca R245fa	Propane series	A2	
R404A	Zeotropic mixture (44% R125, 52% R143a, 4% R134a)	A1/A1	1K
R407C	Zeotropic mixture (23% R32,25% R125, 52% R134a)	A1/A1	7K
R410A	Zeotropic mixture (50% R32, 50% R125)	A1/A1	near 0K
R507A	Azeotropic Mixture (50% R125, 50% R143a)	A1	
R508B	Azeotropic mixture (46% R23, 54% R116a)	A1	

Some comments on the refrigerants presented in the table are given below:

- **R32** and **R125** are seldom used as single refrigerants, but only in mixtures with particularly favorable thermodynamic properties.
- **R245c** and **R245fa** are used almost exclusively in the United States and in a rather experimental way.
- **R404A** has been developed as an alternative to R502 for refrigerators and freezers.
- **R134a** was the first HFC introduced in refrigeration and air conditioning with great success, because it requires almost no changes in the equipment designed for R22. However, it offers a very limited efficiency, about 40% lower than that obtained with R22. Consequently, the manufacturer has two choices: either to accept a substantial reduction in the thermal capacity in a given system, or to increase its dimensions (and cost) to achieve the same capacity. For this reason, R134a is used mainly in large systems (over 250 kW) that can afford the higher costs.
- **R407C** is, like R134a, thermodynamically similar to R22 and works as a “drop in” refrigerant. However, unlike R134a, which is a pure compound, R407C has a glide of 7 K, making it barely usable

in small residential (household) equipment. There are two reasons to justify such a limitation: residential equipment is more subject than other equipment to sudden accidental losses, and it is usually serviced on site. In the event of a sudden leakage, a 7K glide may result in changes in the proportions of the mixture, because the relative losses of its most volatile components will be disproportionately high. If a standard refill is used, there is no guarantee that the new refrigerant mixture has the same proportions as it had before the leakage. Due to its high glide, this refrigerant is used only in medium-capacity systems (50-250 kW), which are usually serviced by skilled personnel.

- **R410A** has very attractive thermodynamic properties, higher energy efficiency than R22, no glide and hence no problem with the mixture remaining after charge loss and refill. However, it has an operating pressure almost double that of R22, and therefore requires a redesign of the whole system with larger compressors, expansion valves, etc.
- **R507A** is used successfully in industrial and commercial refrigeration.
- **R508B** is less frequently used in low temperature cycles. R507A and R508B have favorable thermodynamic properties and no problems with temperature glides, because they are azeotropic mixtures.

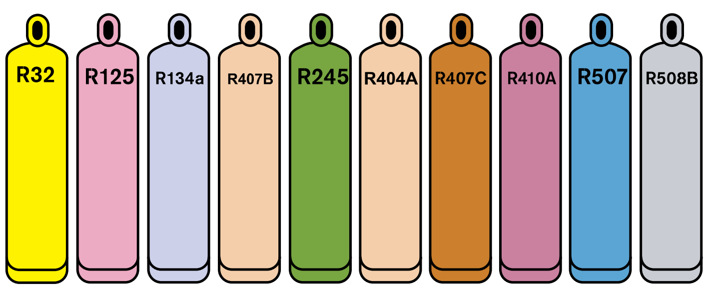


Figure 5.4 Some HFC refrigerants.

FC = FluoroCarbons

Fluorocarbons (Figure 5.5) contain no chlorine and are not harmful to the ozone layer. However, they are extremely stable, and they have a high GWP (cf. section 5.3). R218 is an example of a fluorocarbon, and FCs are also present in the mixtures R403 and R408.

HC = HydroCarbons

Hydrocarbons are a very limited solution to the environmental problems associated with refrigerants. They are harmless to the ozone layer (ODP = 0) and have hardly any direct green house effect (GWP<5), but they are highly flammable. The use of HCs as refrigerants is confined to Europe, because many other countries elsewhere have banned the use of flammable gas in the presence of the public. According to the standards ISO 55149 and EN 378.2000, this should apply also in Europe. However, the standard IEC 355.2.20 allows the use of HCs in household refrigerators with refrigerant charges up to 150 g.

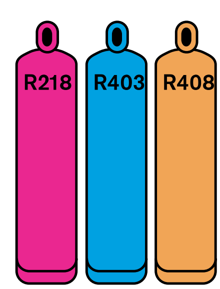


Figure 5.5 Some fluorocarbon refrigerants.

This standard has opened the way for some European refrigerator manufacturers to produce household refrigerators with flammable isobutene, R600a. These have been accepted enthusiastically by environmentalists, and have achieved great success in the market.

NH_3 = Ammonia

Ammonia, **R717**, is an attractive refrigerant alternative. It has been used in refrigeration systems since 1840 and in vapor compression since 1860. In terms of its properties, it should be considered a high-class refrigerant. Furthermore, its ODP and GWP are 0. However, although it is a self-alerting gas, i.e. leaks can easily be detected by the smell, ammonia is very hazardous even at low concentrations because the smell often causes panic. This is the main reason why ammonia was withdrawn from applications for use by unskilled people and retained only for industrial applications. It is also quite common in commercial refrigeration, although safety regulations require that it be used with a **secondary distribution loop**. Obviously, this secondary loop reduces the efficiency.

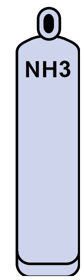


Figure 5.6 Ammonia.

CO_2 = Carbon Dioxide

R744, carbon dioxide, has several attractive characteristics: non-flammable, does not cause ozone depletion, very low toxicity index (safety A1), available in large quantities, and low cost. However, it also has a low efficiency and a high operating pressure (approximately 10 times higher than R134a). For the two latter reasons, efforts are needed to improve its refrigeration cycle and related technology, particularly heat exchangers and expansion devices. A major forthcoming CO_2 application seems to be air conditioning in the automotive industry. Heat pumps could also benefit from CO_2 due to the higher temperature that can be obtained even at very low ambient temperatures.

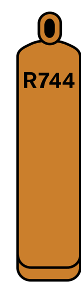


Figure 5.7 Carbon dioxide.

Summary Table

Table 5.2 Summary table of the different types of refrigerants.

Prefix	Meaning	Atoms in the Molecule
CFC	chlorofluorocarbon	Cl, F, C
HCFC	hydrochlorofluorocarbon	H, Cl, F, C
HFC	hydrofluorocarbon	H, F, C
HC	hydrocarbon	H, C

5.5 Health and Safety Classification

The health and safety classification of refrigerants affects the field of application, system construction, etc. The classification, by ASHRAE (standard 34), uses two classes (A and B) and three groups (1, 2 and 3) according to their health hazards and flammability, as shown in Table 5.2.

Table 5.2 The health and safety classification of refrigerants, as defined by ASHRAE.

Class/ Group	Description
A	The toxicity of the refrigerant is not identified at concentrations below 400 ppm by volume.
B	Evidence of toxicity identified at concentrations below 400 ppm by volume.
1	No flame propagation in air at 18°C and 101 kPa. This group has the least severe restrictions, but the maximum amount of refrigerant in the system (g/room volume) is limited. The purpose of this limitation is that if there is a pipe fracture, the concentration of refrigerant in the room is kept below the hazardous level.
2	Lower flammability limit than group 1, but the toxicity is the largest hazard. These refrigerants may not be used for comfort refrigeration in direct systems. There is also a limit on the amount of refrigerant in the system, depending on the category to which the room in question belongs.
3	Highly flammable. This group has the most severe restrictions. Depending on the filling amount, a special machine room is required, and special licenses are needed in some cases. If a refrigerant is not classified, it is considered as a group 3 refrigerant until classification.

5.6 Oil and Refrigerants

Mineral oil is the compressor lubricant used with CFCs and HCFCs. Due to the different properties of HFCs, mineral oil cannot be used as a lubricant for HFC systems. For example, the miscibility and solubility of HFC and mineral oil differ from those of mineral oil and CFCs or HCFCs. If HFCs and mineral oils were used together, both the oil return to the compressor and the heat transfer in the evaporator and condenser would be impaired.

To achieve an acceptable miscibility between the refrigerant and the lubricant, polyolester oils (POEs) are normally used with HFCs. Polyvinyl ethers are also being introduced in some applications. In automotive air conditioning systems, polyglycol (PAG) is used by most manufacturers, but some manufacturers recommend POE for retrofits. POEs are very **hygroscopic**, and will therefore absorb water if given the chance. If too much humidity enters the oil, there is a risk of acid formation, which could attack the components of the system. Polyolester oils used with the new refrigerant alternatives also seem to have a cleaning effect. This could lead to a possible problem in converted (retrofitted) systems. Deposits are dissolved and collect in the expansion valve, with consequent blocking of the refrigerant flow. Installing a proper and efficient filter drier in the liquid line can prevent both the humidity and the blocking problem.

When converting an existing system to HFCs, the mineral oil is removed and replaced by polyolester oil. If too much mineral oil is left in the system after converting, it may be deposited in the evaporator and impair the boiling heat transfer.

A refrigerant mixture consists of two or more components. The extent to which these dissolve in the compressor oil varies for the different components. In systems where the refrigerant charge is small in comparison to the oil volume, this difference in solubility may lead to problems. For example, if a mixture has a small percentage of one component that dissolves more readily in the oil than the other components, this will influence the physical properties and thereby the system performance.

5.7 Azeotropic/Zeotropic Refrigerants

A refrigerant may be either a pure compound or a mixture (blend) of two or more refrigerants. Examples of pure refrigerants are R12, R22 and R134a. Examples of mixtures are R502, R404A and R407C. A mixture can behave either as a pure refrigerant (azeotropic mixtures), or differently (non-azeotropic, or zeotropic, mixtures).

Azeotropic Mixtures

Although it contains two or more refrigerants, at a certain pressure an azeotropic mixture evaporates and condenses at a constant temperature. Because of this, azeotropic mixtures behave like pure refrigerants in all practical aspects. Figure 5.8 shows that the temperature is constant in the liquid-vapor mixture region for a given pressure.

Non-Azeotropic/Zeotropic Mixtures

Zeotropic mixtures have a gliding evaporation and condensing temperature (see Figures 5.9 and 5.10). When evaporating, the most volatile component will boil off first and the least volatile component will boil off last. The opposite happens when gas condenses into liquid. Figure 5.8 shows that for a given pressure, the temperature will change in the liquid-vapor mixture

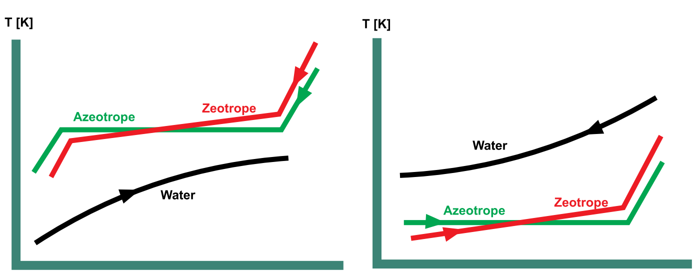


Figure 5.9 Condenser and evaporator temperature program, counter-current flow.

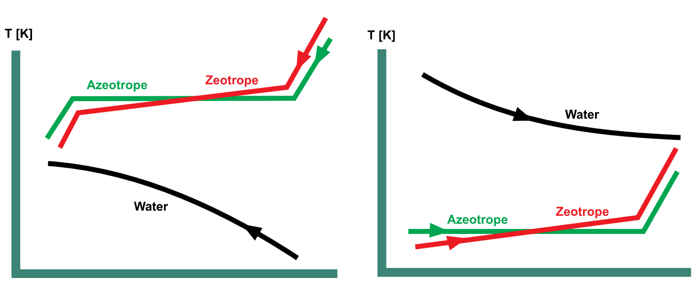


Figure 5.10 Condenser and evaporator temperature program, co-current flow.

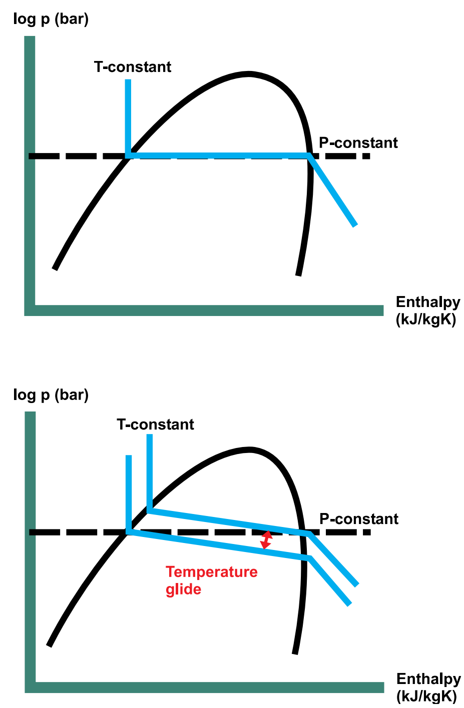


Figure 5.8 Upper: Pure refrigerant or azeotropic mixture (no glide). Lower: Non-azeotropic (zeotropic) mixture (glide).

region. This results in a gliding evaporation and condensing temperature along the heat transfer surface. In practice, the saturation temperature at the inlet of the evaporator will be lower than at the outlet. In the condenser, the saturation temperature at the inlet will be higher than at the outlet.

Heat Exchangers and Refrigerants with Glide

To exploit the temperature glide optimally, it is necessary for the heat exchanger to operate with a counter-current flow. This gives an advantage for CBE heat exchangers compared with Shell & Tube (S&T) heat exchangers. S&Ts do not operate with a truly counter-current flow, and test results have shown a decrease in capacity for this type of heat exchanger that is not experienced with CBEs.

For refrigerants with glide, e.g. R407C, it has been indicated that a substantial sub-cooling is required to receive pure liquid from the condenser. Thus, if the condenser is operating with co-current flow, an increased condensing temperature could be the result.

During the evaporation and condensation of zeotropics, there is a negative effect on the heat transfer coefficient due to mass transport phenomena in the refrigerant. High turbulence and good mixing in the heat exchanger neutralizes this negative effect, which also suggests that CBEs have an advantage over S&Ts.

Another problem with zeotropics can occur if refrigerant liquid is allowed to collect somewhere in the circuit, e.g. in suction line accumulators, flash tanks, receivers or pool boiling/flooded evaporators (often S&T). A change in the composition of the refrigerant circulating through the system could follow from this, resulting in unpredictable performance. To avoid this, all components should have a continuous flow of refrigerant without the opportunity to collect any liquid. Hence, flooded evaporators will probably disappear from systems containing mixtures with glide.

A considerable leak of refrigerant in the liquid-vapor region can lead to the same problems, i.e. changed composition of the remaining refrigerant charge, giving unpredictable system performance.

5.8 Secondary Refrigerants

Secondary refrigerants allow the amounts of environmentally harmful primary refrigerants to be minimized and contained in a restricted area. Examples of secondary refrigerants include water, air, hydrocarbons, ammonia and carbon dioxide, which are more environmentally benign than traditional refrigerants such as HCFCs. They are safer (some are even incombustible and non-toxic) and generally suitable for refrigeration systems. Brines are often chosen as secondary refrigerants for large refrigeration systems, such as those supplying supermarkets, the most common brines being water-glycol solutions, water-ethanol solutions and acetate solutions.

The traditional method of refrigeration for medium- and low-temperature display cases in supermarkets is to circulate primary refrigerants

(CFCs, HCFCs and HFCs) from a centrally located plant room. The charges of primary refrigerants for such systems are large, and they are contained in extensive ducts with many potential sources of leakage. The refrigerants are expensive and have negative environmental impacts when released to the atmosphere. One way of reducing the amount of these refrigerants and the risk of leakage is to chill a secondary refrigerant, such as glycol or brine, in a centralized plant room and circulate this rather than the primary refrigerant.

Secondary loop systems thus employ two separate heat transfer loops: one for the primary refrigerant and one for the secondary refrigerant (see Figure 5.11). The secondary refrigerant is circulated through the display cases in the supermarket using a pump. The primary refrigerant is contained within the primary loop in the machine room and does not enter the retail sales floor. A heat exchanger is required to cool the secondary refrigerant with the primary refrigerant. Because the primary refrigerant is no longer in close proximity to the customers, it is possible to use alternative fluids, such as ammonia. A number of supermarket installations with CO_2 as the secondary refrigerant have been installed with good results. Supermarket systems using ice slurry (a mixture of water, ice and ethanol) have also been evaluated in some European countries.

The disadvantage of this kind of system is the thermodynamic penalty from using an extra heat exchanger and a pump for the secondary loop, both of which add to the system's power consumption. This penalty may be reduced by new developments in refrigeration technology.

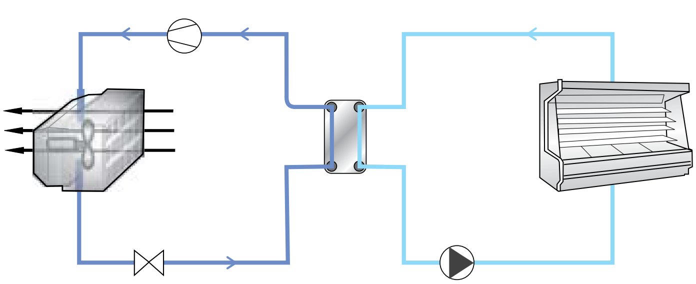


Figure 5.11 A secondary refrigerant used in a secondary loop.

6 Evaporators

The evaporator is one of the five essential components in the refrigeration system, together with the condenser, compressor, expansion valve and refrigerant. In the evaporator, the refrigerant boils off by absorbing energy from the warmer secondary fluid, thus reducing the temperature. The secondary fluid may be a gas or a liquid, depending on the system.

6.1 General Function and Theory

The Evaporation Process

When sub-cooled liquid refrigerant at high pressure (state (a) in Figure 6.1) expands through the expansion valve, the pressure and thus the saturation temperature decrease (b). The amount of flash gas formed after the expansion valve decreases with the level of sub-cooling and the evaporator inlet pressure. The mixture of liquid and gas from the expansion valve enters the evaporator and starts to boil, because heat is transferred from the warmer secondary fluid (b-c). The evaporating refrigerant absorbs energy from the secondary fluid, whose temperature is reduced. After full evaporation, when 100% of the refrigerant has become saturated vapor (c), the temperature of the vapor will start to increase, i.e. the vapor will become superheated. The refrigerant flow leaving the evaporator will be 100% superheated vapor (d).

The total energy absorbed by the refrigerant is often called the Total Heat of Absorption (THA). It consists of the latent energy of evaporation (b-c in Figure 6.1) plus the sensible energy of superheating (c-d in Figure 6.1). The refrigerant vapor is superheated mainly to ensure that dry gas enters the compressor. Many control systems, such as thermal expansion valves, also regulate by means of the outlet temperature, and so superheating is necessary to achieve stable control of the evaporation process.

Evaporator

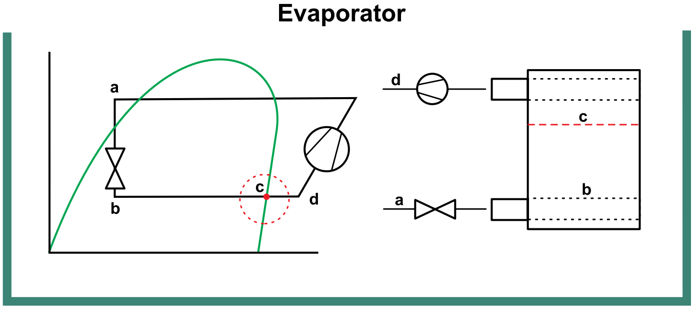


Figure 6.1 A log P/h diagram showing the evaporation process (left) and a CBE evaporator sketch displaying the corresponding thermodynamic states (right).

The evaporation temperature of a pure refrigerant corresponds to a certain pressure level and remains constant unless the pressure is changed. In reality, however, the evaporation temperature is never constant through the evaporator. Inside an evaporator, the increased velocity of the liquid/gas refrigerant mixture will induce a pressure drop, which thus reduces the saturation temperature. Refrigerant mixtures consisting of refrigerants with different boiling temperatures will increase in temperature during the boiling process; the refrigerant is said to "**glide**".

A good evaporator is able to provide a good, stable boiling process with a small temperature difference between the refrigerant and the secondary fluid. A low temperature difference means that a higher evaporation temperature is possible, which corresponds to a higher pressure. Decreasing the pressure difference from the low-pressure side (evaporator) to the high-pressure side (condenser) will decrease the energy use in the compressor. The higher evaporation pressure will also increase the density of the refrigerant gas. For each stroke, the compressor will therefore transport more refrigerant through the system. Lower electricity consumption and higher refrigeration capacity will increase the total system efficiency (COP)

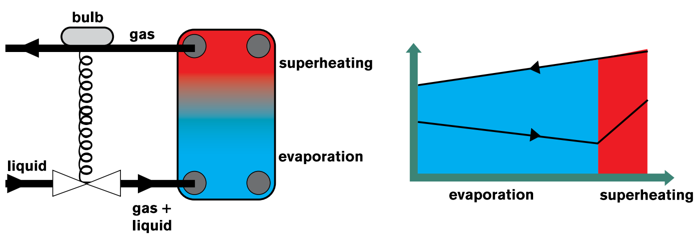


Figure 6.2 Simple image of a DX system and temperature program for an azeotropic refrigerant.

A **direct expansion (DX) system** is recognized by the expansion valve that lowers the pressure of the warm liquid condensate (see Figure 6.2). This creates a cold gas/liquid mixture that enters the evaporator. Normally, there is no collecting vessel after the evaporator, and thus the refrigerant must be superheated a few degrees before the compressor to avoid liquid refrigerant leaving the evaporator. DX systems require fewer components than **flooded systems** and are less expensive to build.

A **flooded (wet) evaporator** works instead with a low-pressure receiver situated after the expansion valve. The receiver separates vapor from the liquid. The receiver ensures that vapor is fed to the compressor and that 100% liquid is fed to the evaporator. Typically, not all refrigerant is boiled off after one pass through the evaporator, and the refrigerant has to be re-circulated. No superheating is therefore possible. The receiver will continuously separate the vapor from the expansion valve and the outlet of the evaporator and feed it to the compressor, while redirecting the remaining liquid to the evaporator as described in section 6.9. A flooded evaporator system is shown in Figure 6.3.

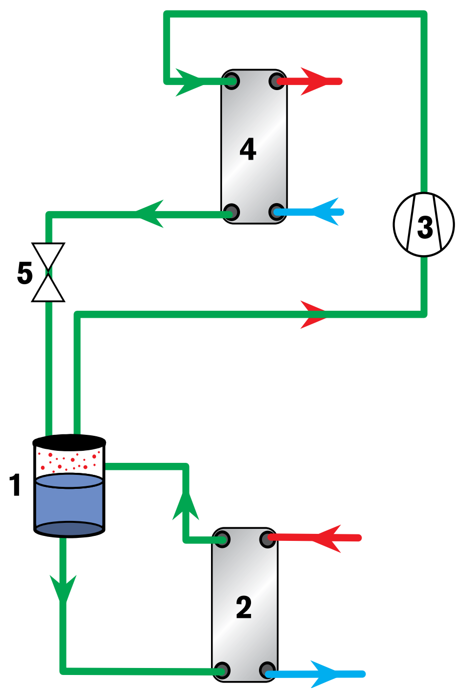


Figure 6.3 Simple image of a flooded system. (1) Low-pressure receiver, (2) Flooded CBE evaporator, (3) Compressor, (4) CBE condenser, (5) Expansion valve.

6.2 Boiling Regimes

The theory of boiling is complex and not yet fully understood. The boiling process depends on factors such as mass flow, vapor content and the temperature difference between the refrigerant and the heating surface. The available research is based on empirical experiments on smooth tubes. Understanding the evaporation process inside plate heat exchangers is further complicated by the swirling flow that the plate pattern induces.

However, the processes of evaporation could be explained by a combination of the available research on pool boiling and flow boiling. SWEP has developed algorithms based on these theories and empirical laboratory tests that resemble the heat transfer.

Pool Boiling

The heat transfer coefficient for pool boiling has a characteristic curve that displays **heat flux** versus the temperature difference between the evaporating and the secondary media (see Figure 6.4). As the temperature difference increases, the energy flow per heat transfer area (kW/m^2) will increase until an unstable maximum value for heat transfer is reached, the **critical heat flux**. At this point, vapor will form to such an extent that it hinders contact between the liquid refrigerant and the heat transfer area. As the temperature difference increases, a continuous gas film will form and the heat flux will reach a minimum. A further increase in the temperature difference will increase the heat flux only slowly, because the heat transfer coefficient will depend on radiation. High heat transfer coefficients after this point are achieved only with a temperature difference of >800K, where CBEs are not a suitable heat transfer solution. Preferably, a CBE evaporator should be designed to operate below the critical heat flux to avoid the unstable performance of partial film boiling and the low heat transfer coefficient of complete film boiling.

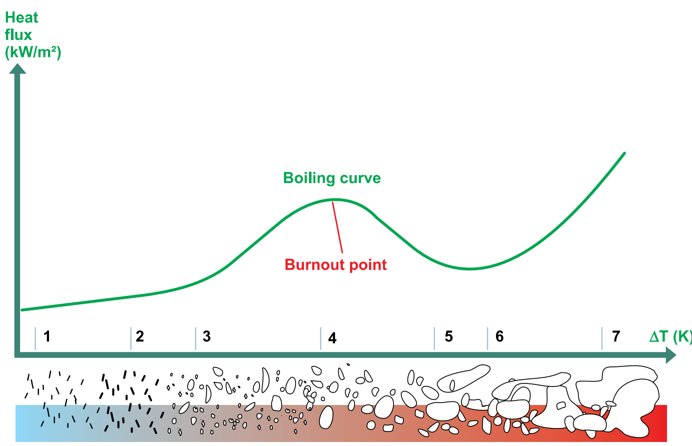


Figure 6.4 Heat transfer for pool boiling.

The different boiling regimes in Figure 6.4 are explained below:

1. **Free convection evaporation.** Heat is transferred between the wall and the refrigerant without bubble formation. The liquid close to the surface wall becomes slightly superheated and evaporates in the interface between liquid and gas.
2. **Sub-cooled nucleate boiling region.** Heat transfer between the wall and the refrigerant is sufficiently large to create bubbles, but they collapse in contact with the liquid bulk.
3. **Nucleate boiling region.** This is the most important boiling region for technical applications. The temperature difference required to enter this region is approximately 3K. Superheated liquid overcomes the surface tension, forming unstable vapor bubbles that collapse in contact with the sub-cooled refrigerant liquid. The additional turbulence caused by forming and collapsing bubbles increases the heat transfer. The heat transfer increases as the temperature difference between the refrigerant and the secondary medium increases to reach a maximum critical heat flux at the **burnout point**.
4. **Partial film boiling.** Increasing the temperature difference further, beyond the burnout point, will cause the refrigerant to evaporate too

quickly to allow new liquid refrigerant access to the heat transfer surface. The partial vapor blanket acts as insulation, causing the heat transfer coefficient to drop greatly and reducing the overall heat flux. This is an unstable region that should be avoided, because performance is uncertain and may fluctuate substantially.

5. **Complete film boiling.** With a very high temperature difference, a stable film of refrigerant vapor will form on the heat transfer area. The film will act as an effective insulation layer hindering any direct contact between the heat surface and the liquid refrigerant. Although the heat transfer coefficient reaches a minimum here, this evaporation region is preferable over the unstable partial film-boiling region, because predictions of the heat transfer coefficient are more reliable.
6. **Radiation.** At very high temperatures, the heat flux will increase again due to radiation.

Flow Boiling

The theory of flow boiling inside tubes or channels is more complex than that of pool boiling. The gas phase has a much lower density than the liquid phase, and thus the evaporation of the refrigerant causes an acceleration of the fluid. Different flow regimes have been identified in experiments with different mechanisms of heat transfer. The different flow regimes are shown in Figure 6.5.

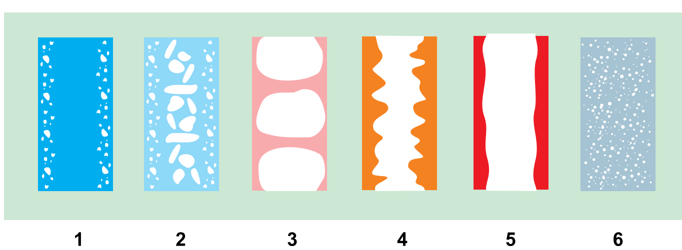


Figure 6.5 Flow regimes for flow boiling.

The different flow regimes are discussed below:

1. **Sub-cooled boiling.** The temperature difference is not sufficient to initiate stable vapor bubbles that will collapse in contact with sub-cooled liquid. If the vapor content is small (<5-8%), the heat transfer mechanism will be similar to that of pool boiling.
2. **Bubble or emulsion flow.** Vapor nucleates form and grow as a result of evaporation at the gas/liquid interface. For evaporators with small temperature differences (< 3K), this flow regime dominates the initial part of the evaporator before enough vapor is formed to increase the turbulence.
3. **Slug flow.** Vapor forms spontaneously and merges into big bubbles. This flow is typical of evaporators with conventional temperature differences (5-10 K) between the refrigerant and the secondary fluid.

4. **Annular flow (regions 4 and 5).** The vapor phase accelerates and
5. forms a "chimney" that pushes the liquid phase upwards. Annular flow is often found at the top of CBE evaporators, where the

vapor quality is high. However, the liquid phase remains in contact with the surface area, ensuring a high heat transfer coefficient.

6. **Mist flow:** the vapor velocity becomes high enough to tear the liquid film from the heat transfer area; this results in greatly reduced heat transfer coefficients.

For a CBE evaporator working in a direct expansion (DX) system, refrigerant vapor is already present at the inlet. Flow regimes 2-5 are therefore most applicable to CBEs. Flooded evaporators also have flow types 1 and 2 at the beginning.

Flow boiling heat transfer mechanisms are explained as a combination of nucleate and convective boiling from the pool boiling theory. The heat transfer coefficient for nucleate boiling is higher than for convective boiling, but nucleate boiling requires a larger temperature difference between the refrigerant and the secondary medium. To simplify the relation between convective and nucleate flow boiling, it is possible to assume that the heat transfer coefficient is the net effect of the two mechanisms, as shown in Figure 6.6.

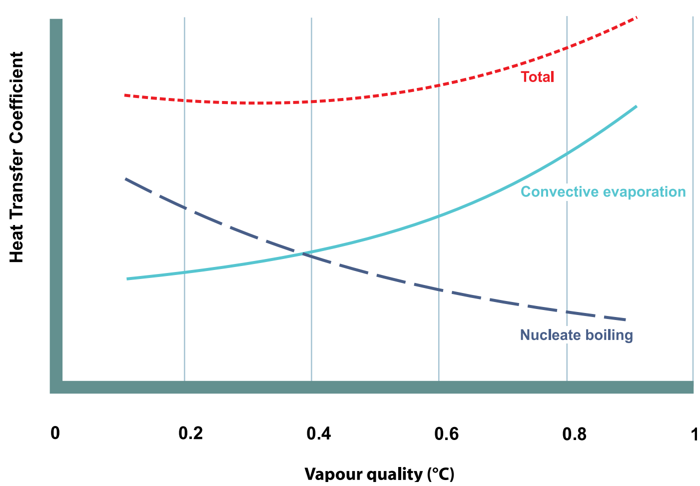


Figure 6.6 Total heat transfer coefficient as the sum of convective and nucleate boiling.

At the beginning of the evaporator, where there is normally only 15-30% vapor, nucleate boiling is the dominant heat transfer mechanism. When the vapor content is high, i.e. higher up in the evaporator or in evaporators with a very high inlet vapor quality, the convective heat transfer mechanism becomes dominant. This is easily understood, because convective boiling depends on the total liquid/gas interface area, which increases with the vapor quality.

As Figure 6.7 indicates, the heat transfer coefficient for an evaporator operating in the convective regime is almost independent of heat flux, but highly influenced by the mass flow and turbulence in the liquid phase. At a certain heat flux, the vapor pressure is sufficiently high to overcome the surface tension, and nucleate boiling is initiated. At this point, the heat transfer coefficient increases with increasing heat flux.

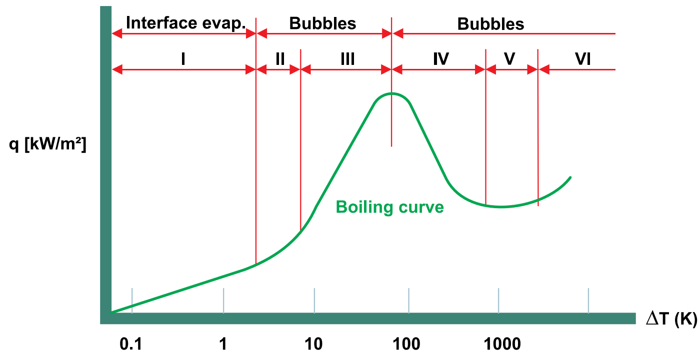


Figure 6.7 The heat transfer coefficient for convective and nucleate boiling. In zone I, the heat is transferred purely by convection; superheated liquid rises to the liquid/gas interface where evaporation takes place. In zone II, nucleate boiling bubbles condense in superheated liquid. In zone III, the nucleate boiling bubbles rise to the gas/liquid interface. In zone IV, there is partial nucleate boiling and an unstable nucleate film. In zone V, the film boiling is stable, and in zone VI, radiation dominates the heat transfer.

6.3 Boiling Process

The boiling of a refrigerant is a complicated process. It changes with induced pressure drop, the type or blend of refrigerants, the temperature difference between the hot and cold sides and other factors. The following subchapters will discuss the most important factors influencing the evaporation process.

Temperature Profile inside the Evaporator

The boiling temperature of a refrigerant at a certain pressure is called the **saturation temperature**. At the saturation temperature, any additional energy absorbed by the refrigerant will transform liquid to gas. If the pressure is constant, the temperature will remain at the saturation temperature until no liquid remains. Only at the point where no liquid refrigerant is present can the temperature of the vapor increase, i.e. become superheated. A popular example to illustrate this is water, which boils at 100°C at sea level. The temperature will not increase even if the water boils vigorously. Hence, 100°C is the saturation temperature of water at a pressure of 1 atm. To change the evaporation temperature, the pressure must be changed. At the top of a mountain, where the atmospheric pressure is lower, the saturation temperature of water is lower, and it starts to boil at temperatures lower than 100°C.

Figure 6.8 illustrates the temperature profiles inside an evaporator for an evaporating and superheating refrigerant compared with the secondary fluid. Refrigerant a evaporates at a higher evaporating pressure and thus saturation temperature than refrigerant b.

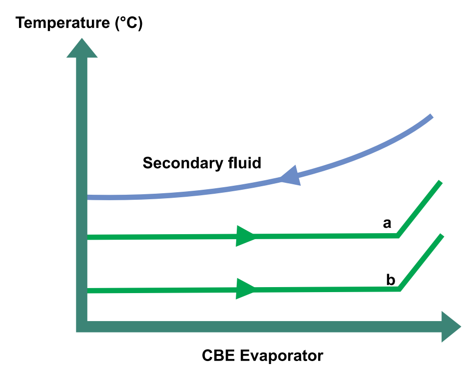


Figure 6.8 The temperature profile inside a CBE evaporator depends on the pressure level of the refrigerant: (a) is at high pressure and (b) is at low pressure.

Influence of Superheating on the Evaporation Process

The latent energy absorbed in the evaporation represents the majority, approximately 95%, of the total heat of absorption (THA). Superheating of the vapor represents the remaining heat transfer to the refrigerant.

The evaporation process accounts for most of the heat transfer area of the evaporator. Although the superheating only accounts for approximately

5% of the total heat of absorption, the gas heating process normally takes up 10-25% of the total heat transfer surface. This imbalance in the required heat transfer surface versus energy transfer can be explained by the differences in heat transfer coefficients between boiling liquid refrigerant and one-phase gas heating. During the evaporation process, the CBE wall is in contact with boiling liquid refrigerant made turbulent by the vapor formed. When the liquid phase has boiled off, the energy is instead transferred through the vapor, i.e. a one-phase process, with a greatly reduced heat transfer coefficient as a result. Because the temperature of the vapor increases, the driving force for heat transfer is gradually reduced.

A CBE designed with a very high level of superheating (see Figure 6.9) will therefore require a proportionately very large heat transfer area for vapor heating. Similarly, a CBE designed with a normal level of superheating that is forced to operate with too much superheating will have less heat transfer area available for evaporation. The result is a decreased evaporation temperature, which leads to reduced system capacity and efficiency.

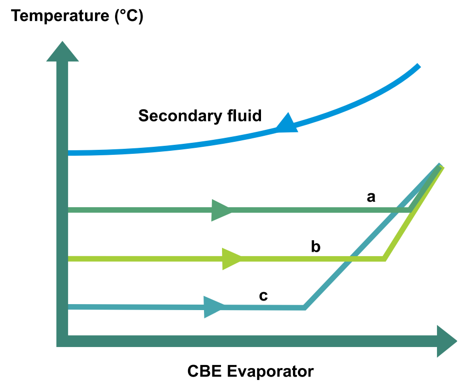


Figure 6.9 Temperature program for evaporator with (a) low, (b) normal and (c) high levels of superheating.

Effect of Pressure Drop

When the refrigerant flows through the winding channels inside the CBE, turbulence is created and pressure drop is induced. Hence, the pressure will decrease as the refrigerant passes through the CBE. As stated above, the evaporation temperature is constant at constant pressure for a pure refrigerant. Decreasing pressure at a later stage in the evaporator means a decreasing saturation temperature, because the force required to hold the refrigerant molecules together as a liquid will be smaller. The evaporation temperature therefore decreases as evaporation progresses through the unit.

The pressure drop through the CBE channels depends on the characteristics of the refrigerant used and many other factors. The pressure drop increases with higher inlet vapor quality, mass flow and level of superheating. A lower evaporation temperature, which corresponds to a lower evaporation pressure, also results in a higher pressure-drop due to the larger specific volume of the vapor formed.

The pressure drop is proportional to the velocity squared, so a higher vapor content in the refrigerant will result in a larger pressure drop. Because the specific volume of the vapor increases with decreasing pressure, a lower evaporation temperature also gives a higher pressure-drop for the same mass flow.

For an azeotropic refrigerant, i.e. a refrigerant without glide, the pressure drop will result in the inlet saturation temperature being higher than the outlet evaporation temperature (see Figure 6.10). The outlet saturation temperature, which is often referred to as the *evaporation* temperature, is in fact the minimum evaporation temperature inside the CBE evaporator. Normally, the saturation temperature differs by only one or a few degrees over the evaporator.

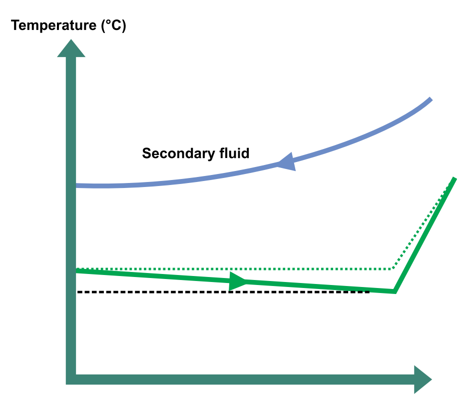


Figure 6.10 The effects of pressure drop on the saturation temperature.

6.4 Glide Refrigerants

The relation between a pure refrigerant and its saturation temperature was discussed above. If a mixture of two or more refrigerants with different evaporation temperature is used instead, the phenomenon of glide may occur. This is a non-azeotropic, or zeotropic, mixture.

When a non-azeotropic mixture boils, the vapor formed first will be rich in the refrigerant with the lowest boiling temperature. The composition of the liquid will therefore change during the evaporation process, with an increasing concentration of the refrigerants with higher evaporation temperatures. Consequently, the saturation temperature will increase, reaching a maximum at the outlet saturation temperature, just before the superheating. This is the opposite situation to a pure refrigerant, where the outlet saturation temperature is the minimum refrigerant temperature.

Because the evaporation temperature of a refrigerant with glide is *not* constant at constant pressure, but will change with the composition of the refrigerant mixture, there will be a start (low temperature) and stop (high temperature) evaporation temperature. To distinguish these, the temperature at which boiling starts is called the **bubble point** and the point where all liquid has evaporated is called the **dew point**. The dew point temperature corresponds to the outlet evaporation temperature of a pure refrigerant.

Some refrigerant mixtures have a very small temperature glide of less than 1K, for example R404A, which has a temperature glide at 0°C of 0.5K. Such refrigerants are called near-azeotropic and the glide can normally be neglected for all practical purposes.

Potential Problems

The thermodynamic properties of interest in heat transfer applications are less favorable for refrigerant mixtures with glide than for pure refrigerants. There are three possible reasons for this:

1. The least volatile refrigerant in the mixture at any time accumulates at the heated surface because the more volatile refrigerants boil off. This increases the bubble point locally near the heated surface and thus also creates a temperature difference between the wall and the bulk liquid.
2. The viscosities and conductivities of the refrigerants are usually less favorable than the weighted average of the components themselves.
3. The start of nucleation in the boiling process may become weaker, leading to fewer nucleation sites. Furthermore, there is a risk of partial distillation of the most volatile components if there is an open vessel to which vapor is led, i.e. a low-pressure receiver or a **pool boiling evaporator.** The refrigeration composition will then vary throughout the system, causing a lower evaporation pressure and a higher condensation pressure, with reduced system performance as a result.

If there is a leak, it is possible that a disproportionate amount of the most volatile component will escape. Refilling the system with new refrigerant without controlling the actual composition of the refrigerant mixture inside the system will create a refrigeration mixture with a new composition. Its thermodynamic properties are unknown, and may not be as good as those stated for the original composition.

Because of the low heat transfer and the risk of refrigerant separation, non-azeotropic refrigerants are not recommended for use with **recirculation type** or pool boiling type evaporators.

Glide in Direct Expansion (DX) Evaporators

For dry expansion evaporators operating with high turbulence and true counter-current flow, e.g. CBE evaporators, non-azeotropic refrigerants are much easier to handle. They may even have some advantages over pure refrigerants.

The high degree of turbulence induced by the winding CBE channels minimizes the effect of local accumulation and temperature increase close to the heat exchanger wall.

Because of the glide, the evaporation temperature will increase through the evaporator with increasing vapor quality. If the heat exchanger is arranged in counter-current flow, the increase in boiling temperature will be compensated by the decrease in temperature of the secondary fluid (see Figure 6.11). Consequently, the mean temperature difference over the CBE evaporator is higher for zeotropic refrigerants compared with refrigerants with a constant evaporation temperature (see Figure 6.11).

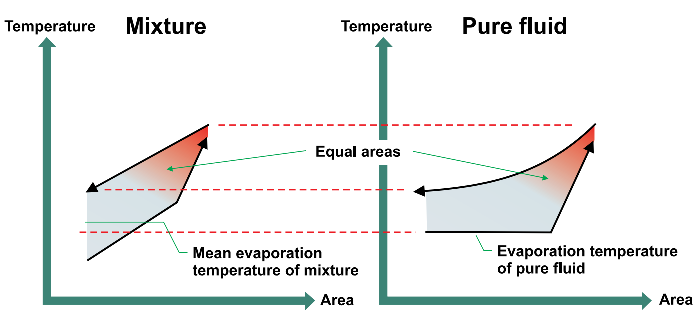


Figure 6.11 Temperature profiles in counter-current evaporator with a glide mixture (left) and a pure fluid (right).

The higher mean temperature difference for glide refrigerants can be exploited to obtain a higher heat flux without the penalty of a lower system efficiency (COP). In reality, this means that a smaller CBE evaporator can be used. Alternatively, the heat transfer surface is kept constant and higher system efficiency is obtained by an increased evaporation pressure and, consequently, higher bubble and dew points.

A potential problem that should be considered is the heat surface required for superheating. Because the temperature difference at the end of the evaporation process is lower for zeotropic mixtures than for refrigerants without glide, the superheating will require more heat transfer area. For a normal level of superheating, 5-7K, the difference is negligible. However, if the evaporator is designed with a very small temperature approach or if a high level of superheating is necessary, the difference increases.

Effects of Pressure Drop

The effects of pressure drop on non-azeotropic (glide) refrigerants are identical to the effects on azeotropic (no glide) refrigerants, except that the glide of the refrigerant and the temperature decrease due to the pressure drop counteract each other. As discussed above, the saturation temperature increases gradually during the boiling process for non-azeotropic refrigerants, whereas the induced pressure drop will act to reduce the evaporation temperature by decreasing the saturation temperature (see Figure 6.12).

To a certain extent, pressure drop acts positively on the superheating, because the temperature difference in the end of the evaporator will increase due to the decrease in saturation pressure. As shown in Figure 6.12, it is possible to remove the effect of the temperature glide totally only if the pressure drop is large enough. However, it is difficult to obtain such a large pressure drop inside a CBE. To reach this condition for a common non-azeotropic refrigerant, R407C, would require a channel pressure drop of 6.1 bar at 0°C, excluding the pressure drop over the distribution system.

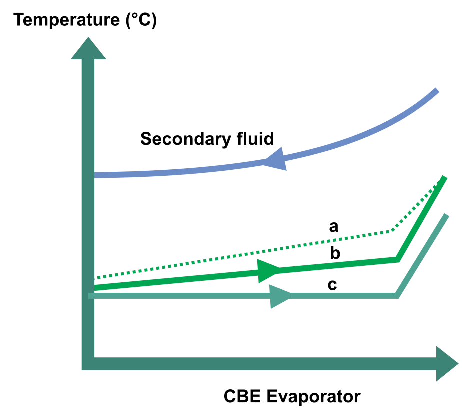


Figure 6.12 The effects of pressure drop on a non-azeotropic refrigerant: (a) temperature profile when no pressure drop occurs; (b) temperature profile for normal pressure drop; (c) temperature profile for extreme pressure drops.

6.5 Co- versus Counter-Current Flow

To make the best use of the temperature difference between the evaporation media and the secondary fluid, the evaporator should operate in counter-current mode, as discussed in chapter 1.6. Furthermore, the more uniform temperature difference in a counter-current evaporator results in a less vigorous evaporation process than in a co-current evaporator. In a co-current evaporator, the large temperature difference in the beginning (see Figure 6.13) will result in a very vigorous boiling process, which creates a large volume of vapor in an early stage of the evaporator. Because the large vapor volume accelerates the flow inside the CBE channels, the induced pressure drop will increase substantially. Pressure drops in co-current evaporators are much larger than in counter-current flow arrangements.

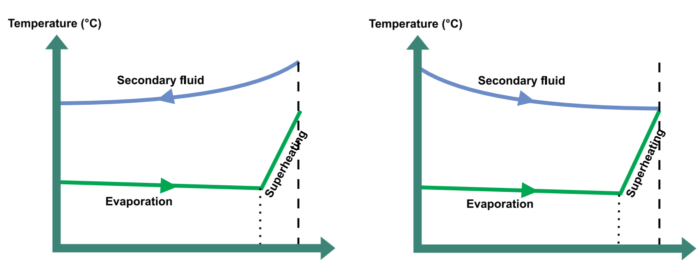


Figure 6.13 Temperature profile for counter-current (left) and co-current evaporator (right).

The temperature glide of non-azeotropic refrigerants has a positive effect on the mean temperature difference for counter-current evaporators, which is reversed for co-current flow. For a counter-current evaporator with a glide refrigerant, the increasing saturation temperature is balanced by the decreasing water temperature, giving a favorable temperature profile. In contrast, a co-current flow arrangement has a negative influence on the temperature profile, with the temperature glide of refrigerant and water temperature converging. At the end of the evaporator, where superheating takes place, the temperatures of the secondary medium and the refrigerant run the risk of crossing each other. Figure 6.14 shows these effects.

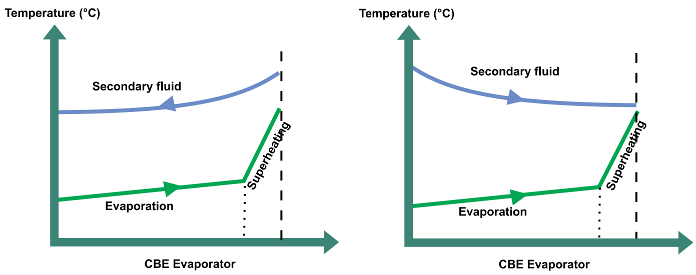


Figure 6.14 Temperature profile for non-azeotropic evaporator operating in counter-current (left) and co-current (right) mode.

Reversible chillers, often called heat pumps, are designed to provide both cooling and heating depending on the requirements. This arrangement imposes special demands on the design of the evaporator and condenser in the refrigeration cycle. The refrigerant flow is reversed by means of a three-way valve, which makes the evaporator operate as a condenser and vice versa.

When mounting an evaporator in a reversible system, it has to be decided whether it will operate in counter-current or co-current mode. When reversing the system, the refrigerant flows in the opposite direction during operation as a condenser. The mounting depends on whether the demand is greater for heating or for cooling. If it is for cooling, the evaporator should work in counter-current mode. If it is for heating, the condenser should work in counter-current mode. There are some special applications where the most beneficial temperature profile is achieved if the evaporator is connected in co-current mode. For example, when using refrigerants that have a high pressure-drop over the evaporator, causing the saturated temperature to drop greatly as it passes the heat transfer surface (see Figure 6.15). Typical applications where co-current flow should be considered are low-temperature or **cryogenic** applications.

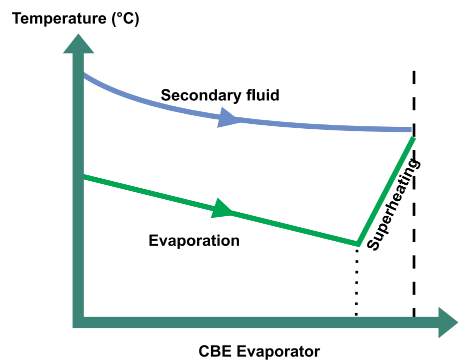


Figure 6.15 Temperature profile of evaporator with extremely high pressure-drop for an azeotropic refrigerant.

6.6 SWEP Distribution System

The brazed heat exchanger was launched as an evaporator for refrigerants by SWEP in the early 80's. However, a problem arose as the demand for higher capacities and larger plate packs increased. The operational efficiency and stability as an evaporator decreased for CBEs that had the large number of parallel channels necessary to cope with high refrigerant flows.

The problem of maldistribution was quickly shown to be caused by the dynamic pressure differences between the inlet and outlet ports. The evaporator's inlet conditions differ from its outlet conditions, because the inlet flow is a mixture of liquid, vapor and lubricant oil, while the outlet flow is dominated by vapor. Thus, the profile of the dynamic pressure gradient differs between the refrigerant inlet and outlet. It will therefore be easier for the gaseous refrigerant to enter the earlier channels than the later ones. The result is an uneven evaporation profile with lower heat transfer, decreased stability of operation and lower utilization of the heat transfer surface (see Figure 6.16).

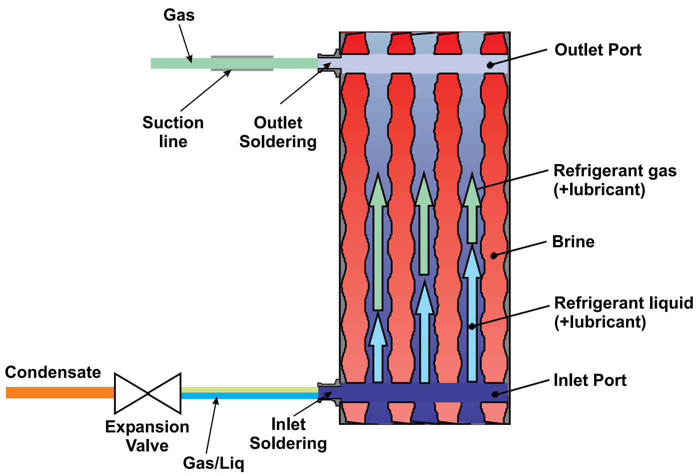


Figure 6.16 Uneven distribution of refrigerant gas.

Approximately 95% of the total heat of absorption inside the evaporator comes from the latent energy of boiling the liquid refrigerant. In the channels where a surplus of liquid refrigerant is injected, the temperature will be greatly reduced, which may result in problems with freezing (see Figure 6.17). The heat transfer area of the heat exchanger is not utilized optimally.

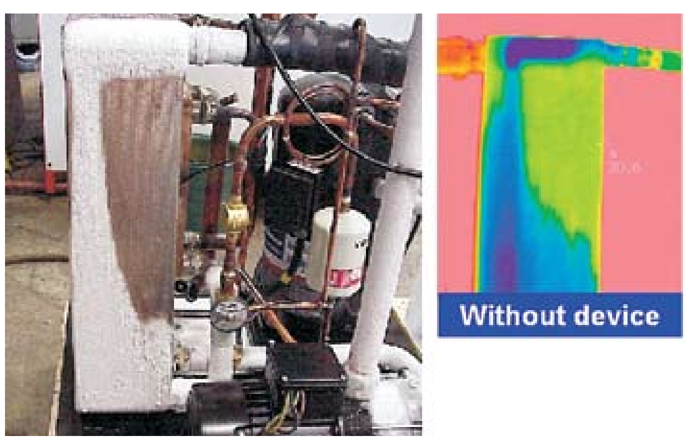


Figure 6.17 Examples of freezing problems. The picture to the right was taken with a heat camera.

SWEP has developed a reliable distribution system. A distribution device, as shown in Figure 6.18, is placed at the inlet of each channel, forming a smooth, consistent tube for the refrigerant mixture. Each device has a carefully defined radial drilled hole, through which the refrigerant flow is forced. The pressure drop induced by forcing the flow through the holes neutralizes the effects of the changing dynamic pressure in the top of the unit. The position of each device at the entrance to each refrigerant channel ensures even distribution. The design of the **V-ring** is optimized for each model, and production control is very advanced in order to ensure correct assembly and maintain the tolerance for the distribution hole. The result is the SWEP range of dedicated evaporators, the V-models. They ensure stable and highly efficient heat transfer for large plate packs and high refrigerant flows.

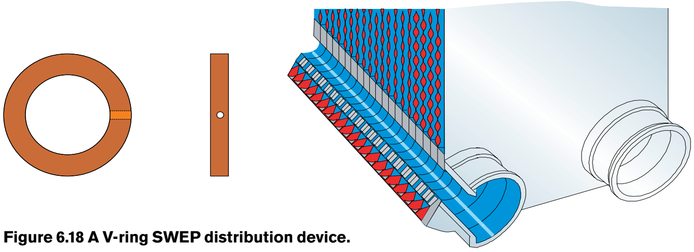


Figure 6.18 A V-ring SWEP distribution device.

Influence of the Inlet Connection

The SWEP distribution device is most effective when there is a homogeneous mixture of liquid and vapor at the entrance to the CBE. Directly after the expansion valve, vapor and liquid are completely mixed. They stay in this homogeneous state provided the flow velocity is high enough to create the necessary turbulence. If the velocity is low, i.e. the pipe dimension is too large, phase separation will occur. The refrigerant flow into the evaporator separates into a fast vapor stream and a slow liquid stream, giving less predictable performance. If the pipe diameter is instead very small, the high velocity will induce a high pressure-drop, resulting in energy losses and a reduced COP.

The recommended refrigerant velocity inside the expansion pipe (F3) to maintain a homogeneous mixture is 10-25 m/s. The inlet connection should never be larger than the inlet port diameter of the F3 port (see Figure 6.19), because this increases the risk of phase separation. Due to the distribution device, the inlet port size (F3) is smaller in a V-type evaporator than in a **B model**.

Figure 6.20 shows the recommended position of the expansion valve relative to the inlet port of a SWEP CBE evaporator. The best position is at a height equal to or greater than that of the evaporator inlet. If this is not possible, the selection of the correct pipe size and the distance from the thermal expansion valve (TEV) to the inlet port becomes more critical.

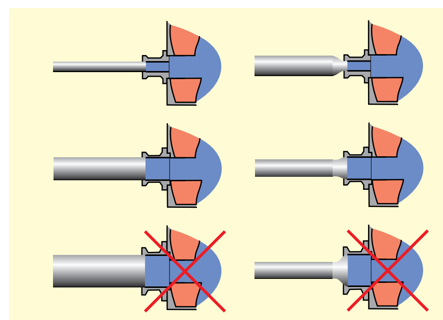


Figure 6.19 Suggestions for different inlet (F3) connections. The inlet connection can be smaller than the inlet port (top) or the same size (middle), but never larger (bottom).

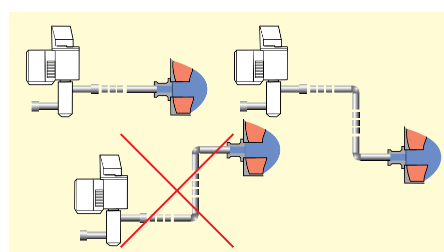


Figure 6.20 Recommended positions for the expansion valve.

Influence of Outlet Connection

The refrigerant vapor leaving the evaporator should have a velocity high enough to force out the small volume of compressor oil that circulates in the system. The oil will otherwise accumulate, adhere to the channel walls and reduce the heat transfer coefficient, resulting in lower evaporation temperature and reduced system capacity. A common recommendation is to design the outlet pipe to attain 5-10 m/s for vertical pipes and 2.5-5 m/s for horizontal pipes to ensure oil return. The higher velocity should be used for evaporation at low temperatures, where the viscosity of the oil is increased.

An undersized connection and suction pipe (see Figure 6.21) will result in unnecessarily high vapor velocities. In the suction pipe, velocities above 25 m/s lead to considerable energy losses, thus lowering the total COP for the system, especially if the suction pipe is long or convoluted. A larger connection and suction pipe will reduce the velocity and restore the performance.

The outlet velocity stated in SSP is the maximum vapor velocity in the CBE port, not corrected for a possibly smaller connection. When this port velocity exceeds 20-25 m/s, the induced pressure drop inside the port will further destabilize the dynamic pressure balance inside the evaporator, increasing the risk of boiling instability. It should be noted that selecting a larger connection than the port diameter does not result in better performance, because a high pressure-drop is induced inside the CBE. Instead, the port velocity can be reduced by allowing double gas exits or selecting a CBE with a larger port diameter.

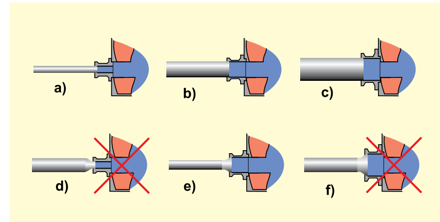


Figure 6.21 Outlet connection variations: (a) Smaller connection than port, (b) Connection size the same as port size, (c) Connection size larger than port, (d) Connection size smaller than port with larger pipe, (e) Connection size the same as port with smaller pipe, (f) Connection size larger than port with smaller pipe.

Effects on the Thermal Expansion Valve (TEV)

The pressure drop over the SWEP refrigerant distribution system should be considered as part of the expansion valve pressure drop rather than part of the CBE pressure drop. The expansion valve and the distribution system operate together, creating the total pressure drop between the condensing and evaporating pressure levels. The pressure drop over the distribution device depends on the operating conditions, and is normally 1-2 bar. This pressure drop occurs after the expansion valve but before the active surface area of the evaporator, which has two consequences:

1. The pressure drop over the distribution system decreases the total pressure drop over the expansion valve. During the design of the expansion valve, the lower pressure drop required can result in the selection of a TEV one size larger.
2. There is a large pressure difference between the inlet of the expansion valve and the position of the temperature **bulb**, so an **externally equalized expansion valve** is required to compensate for the pressure difference. The extra capillary tube evens out the differences in saturation temperatures so that the pressure drop over the distribution device does not affect the outlet evaporation temperature.

Figure 6.22 illustrates the co-operation between the SWEP distribution device and the expansion valve. Figure 6.23 can be used to illustrate the pressure levels just outside and inside the SWEP evaporator.

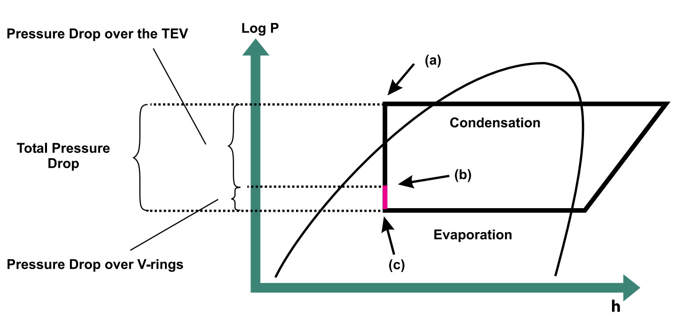


Figure 6.22 The pressure drop over the fixed orifice of the SWEP distribution device, (b)-(c), varies with operating parameters such as summer and winter conditions, capacity and inlet vapor quality. The total pressure drop, (a)-(c), is almost the same for all operating conditions, however, because the expansion valve controls the outlet evaporation pressure (TSAT,OUT). Measuring the temperature after the expansion valve, point (b), will therefore not give relevant values.

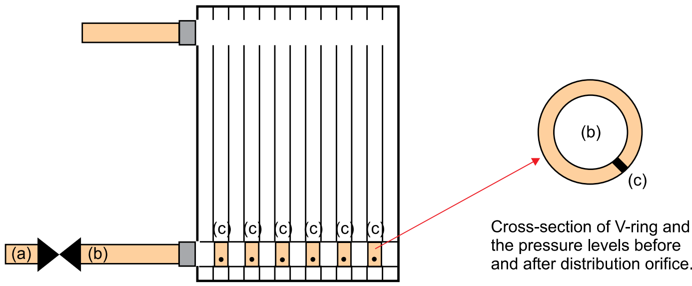


Figure 6.23 Pressure levels before and inside a SWEP V-model evaporator. If the pressure drop over the distribution device is assumed to be 1.5 bar and the temperature 5°C at point (b) (corresponding to a pressure of 5.47 bar), the corresponding pressure at point (c) will be 5.47 - 1.5 = 3.97 bar. The temperature at point (c) will then be -4°C. If the pressure drop over the V-rings were 1 bar instead, the temperature in point (c) would be -1°C. Because it is impossible to predict the pressure drop over the distribution device exactly at a given operating condition, the temperature measurement at point (b) is very uncertain and cannot be given any significance.

6.7 Operating Point

The compressor is the "heart" of the refrigerant system, and is the component that limits the capacity of a refrigerant system. The task of the evaporator and condenser is to utilize the available energy potential fully, thereby maximizing the efficiency and economy of the useful power that can be extracted from the system. The effect of selecting the evaporator for a chosen compressor is discussed in this section.

Matching the Evaporator with a Compressor

As discussed in chapter 3.4, compressor performance depends on the evaporating and condensing pressures. The saturation pressure of the evaporator determines the density of the refrigerant gas at the inlet of the compressor, thus affecting the refrigerant mass flow per compressor revolution. The total pressure difference between the evaporating and condensing sides also affects the required compressor power consumption. The evaporator and condenser affect these pressure levels though their ability to transfer energy between the refrigerant and the secondary fluids.

The evaporator **operating point** is the equilibrium point at which the performance of the evaporator matches the performance of the compressor. The refrigerant mass flow is determined by the compressor, and at the operating point the refrigerant is evaporated at a stable saturation temperature.

The compressor curve in Figure 6.24 has a positive slope, indicating that the available cooling capacity increases with higher evaporation temperature. The evaporator performance curve has a negative slope, indicating that a smaller temperature difference between the refrigerant and the secondary fluid leads to less heat transfer per area. The intersection of these two curves signifies the operating point.

A compressor curve is defined for a certain level of superheating and a certain condensation temperature. Increasing the values of these parameters will decrease the capacity of the compressor, i.e. shift the compressor curve downwards. The evaporator performance should be calculated for the same level of superheating in order to remain comparable with the compressor line. The performance and size of the evaporator influence the

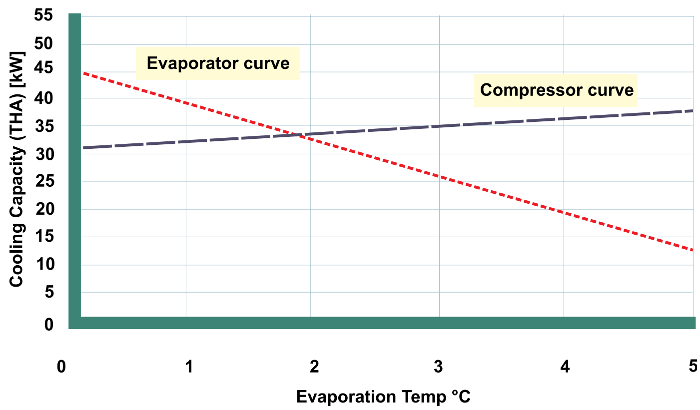


Figure 6.24 Evaporator curve vs. compressor curve. The operating point is found where the two curves meet.

total cooling capacity and performance of the system, because the operating points will differ. Figure 6.25 shows the performance curves of three different evaporators and a compressor curve.

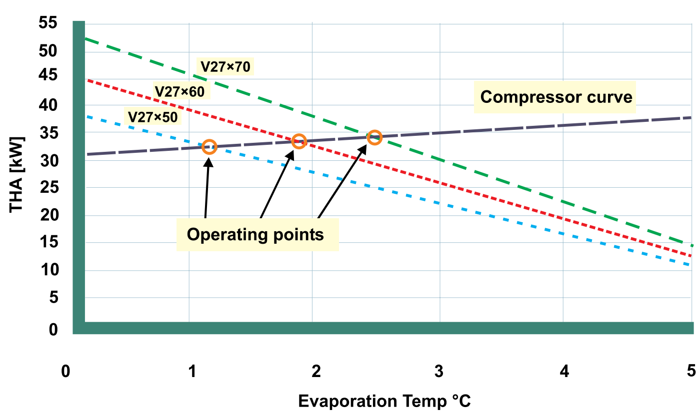


Figure 6.25 Different operating points for different evaporators operating with the same compressor.

Below are some examples of what happens if the heat exchanger is over- or undersurfaced. In this case, a CBE with 60 plates has 0% oversurface.

Table 6.1. Operating points of three different evaporators.

Evap.	Heat transfer area [m²]	Operating Point [°C]	Cooling capacity [kW]	Cooling capacity %	Oversurface in SSP at 2 °C	COP [-]
V27x50	2.88	1.2	32.2	97	-11%	3.88
V27x60	3.48	2.0	33.3	100	0%	4.00
V27x70	4.08	2.6	34.0	102	+12%	4.07

Although the active surface area is reduced from 60 to 50 plates, the reduction in cooling capacity is only 3%. The system performance (COP) falls by 3%.

It should be clear from this example that the refrigerant mass flow, controlled by the compressor, is the single most important parameter for determining the system performance. An underdimensioned evaporator will reduce the evaporation temperature but not alter the system capacity very much.

Oversurfacing the evaporator (in this case from 60 to 70 plates, corresponding to a 17% increase in the heat transfer surface), will not give a performance increase of the same magnitude since the compressor controls the system, as mentioned in the previous paragraph. Despite the large increase in heat transfer area, the cooling capacity increases by only 2%. The increased number of plates will also reduce the turbulence that is essential for stable heat transfer and the CBE's self-cleaning ability.

When determining or analyzing the operating point, it is very important to remember that the refrigeration system is dynamic. The compressor performance curve depends on the condenser performance and the level of superheating. If the conditions of the evaporator are changed, the slope and position of the evaporator curve are also changed, which will alter the operating point. Always make sure that the input data for the compressor and evaporator curve correspond in order to obtain realistic operating points.

6.8 CBE Evaporators

Since the copper brazed plate heat exchanger became commercially available in the early 1980's, new applications have been found every year. The refrigeration industry was one business area that very soon realized and accepted the advantages of this hermetic, pressure resistant, compact and highly efficient heat exchanger technology. SWEP has a very wide range of dedicated evaporators for single or dual refrigerant circuits.

All SWEP evaporator models have the refrigerant inlet at the lower left port, the F3 position (see Figure 6.26). The inlet can also be on the back, corresponding to the P3 position. The refrigerant outlet on all CBE evaporators is the top left port, the F1/P1 position

Each CBE has a specific port size, i.e. the diameter of the holes cut in the plates. To connect the piping to the heat exchanger, a connection is brazed on during the manufacturing process. This connection can have the same nozzle diameter as the port hole, but may also be different to meet the customer's requirements for expansion and suction pipes.

Higher demands on efficiency and compactness for air conditioning and refrigeration systems have improved the market penetration of CBEs as evaporators and condensers for increasingly large capacities. Compactness, a small refrigerant hold-up volume and true counter-current flow make CBEs the favored choice for many system builders.

With increasing system size, it is common to divide the total heat load between two independent refrigerant circuits. This increases the system safety and flexibility compared with using only one refrigerant circuit. Furthermore, most refrigerant systems are dimensioned to handle a nominal heat load higher than the normal cooling demand. The system consequently operates most of the time at part load with one or more compressors inactive. SWEP can

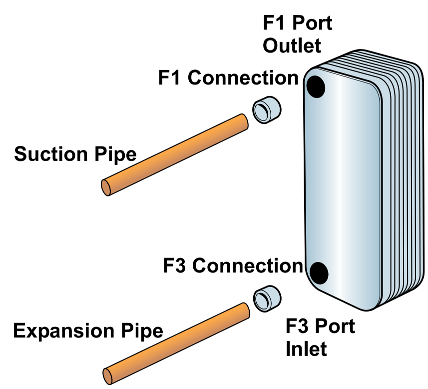


Figure 6.26 A CBE evaporator.

offer two dual circuit technology CBEs: the True Dual and the Back-to-Back dual. It is also possible to arrange two single circuit CBEs in parallel to obtain a dual refrigerant circuit system.

Single circuit

The basic CBE evaporator has one refrigerant and one cooling fluid side. The evaporating refrigerant flow enters at the lower left of the CBE, through port F3 or P3 (see Figure 6.27). The distribution device is located in this port for the V-line dedicated evaporator series. The evaporating flow should normally be upwards because of the lower density of the refrigerant gas formed. Inverting the evaporator is therefore not recommended without contacting SWEP. For a counter-current flow arrangement, the secondary fluid enters at the top right connection, F2 or P2, and exits from F4 or P4, flowing downwards. The arrangement should be reversed for co-current flow.

The secondary side always has one more flow channel than the refrigerant side. The first and last channels in a CBE therefore always contain the secondary fluid, surrounding the refrigerant channels (see Figure 6.28). This has two benefits: firstly, it ensures stable evaporation because all refrigerant channels receive the same heat transfer from the secondary side; secondly, it offers some insulation because it shields the CBE cover from the cold refrigerant.

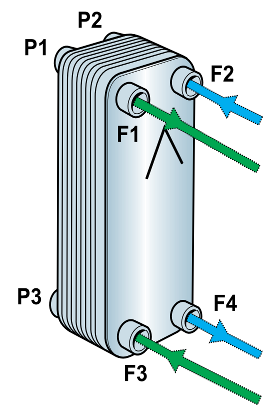


Figure 6.27 The CBE ports for a single circuit CBE.

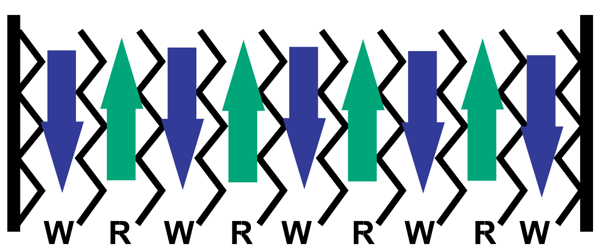


Figure 6.28 The flow channels inside a single circuit CBE. The secondary fluid (W) surrounds the refrigerant (R).

True Dual

The SWEP True Dual technology CBEs (see Figure 6.29) are equipped with two independent refrigerant circuits combined with a common secondary fluid circuit. The patented plate technology assures fully counter-current flow and full symmetry between the refrigerant circuits. The True Dual models are available with or without the SWEP distribution device, and for evaporator or condenser duty. The channels are arranged in such a way that each refrigerant channel (R1 and R2) of the two circuits is surrounded by secondary fluid channels (W). Thus, all secondary fluid channels are always in contact with at least one active refrigerant channel.

Capacity control: A True Dual CBE running with both circuits active operates no differently from a single circuit evaporator. Every water circuit is in contact with two operational refrigerant circuits (see Figure 6.30).

With one refrigerant circuit closed (half-load operation), all channels with secondary fluid will still be in contact with one active refrigerant channel. Thus, the temperature of the secondary fluid channels will be the same, and its leaving

Figure 6.29 A SWEP True Dual CBE.

temperature will be stable and homogeneous, just as for full load. Because the secondary fluid channels surround the active refrigerant channels, the evaporating process remains stable. The True Dual channel arrangement therefore maintains high-efficiency evaporation even at half load, completely eliminating any additional freezing risk.

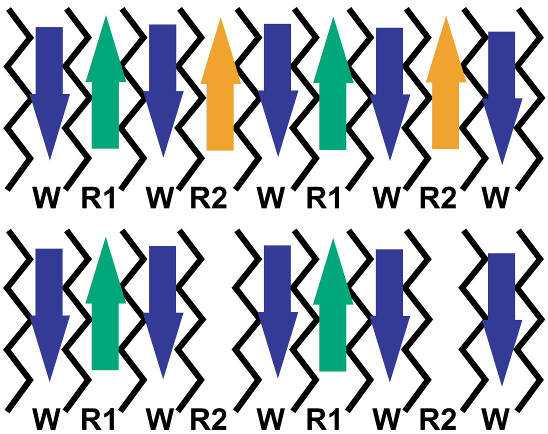


Figure 6.30 The flow channels through a True Dual CBE. The upper figure shows a CBE running on full load and the lower flow channels show a CBE running on half load (one refrigerant circuit is closed).

Combining the True Dual technology with **tandem compressors** (or variable speed compressors) is advantageous. In this arrangement, where each refrigerant circuit operates at half load (see Figure 6.31), 100% of the surface area is utilized for heat exchange. This capacity management method gives a higher thermal efficiency than operating one circuit at full load and closing down the other. As always, consideration should be given to ensuring oil return if the outlet vapor velocity becomes low.

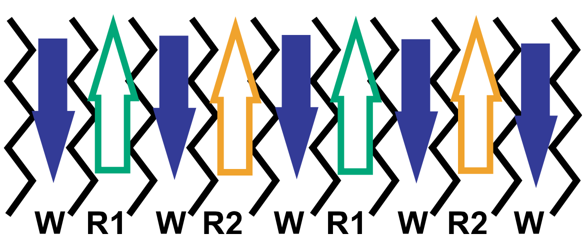


Figure 6.31 A True Dual with tandem compressors (parallel). Here, each refrigerant circuit is operated on half load without closing any channels.

The True Dual technology is based on alternating secondary fluid and refrigerant channels, so it is not possible to build an asymmetric True Dual with different numbers of refrigerant channels on each circuit; it must be a fully symmetric CBE. To meet asymmetric capacity demands on the refrigerant circuits, the True Dual models must be designed with respect to the largest capacity, resulting in oversurfacing for the small-capacity side.

System Design: The True Dual units have centered secondary fluid connections and refrigerant connections on either side (see Figure 6.32), which provides flexibility in the arrangement of the connections. The two refrigerant circuits can be located on either the front or the back, depend-

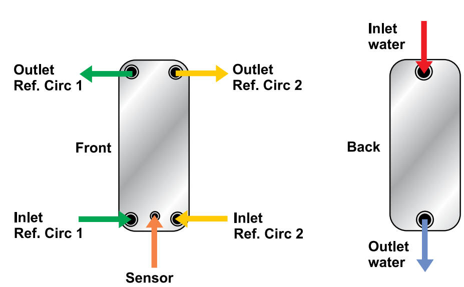


Figure 6.32 The connections on a True Dual CBE.

ing on the customer's demands. The centrally positioned water connections can be positioned on the same or opposite sides, facilitating mounting inside the refrigeration system.

The True Dual refrigerant system is often characterized by extreme compactness, small footprint and uncomplicated piping.

The True Dual symmetric channel arrangement assures an even distribution of secondary fluid inside the unit. It saves the system builder the trouble of arranging the pipes for secondary fluid symmetrically, as is necessary for systems with two single circuit CBEs in parallel. It is very common to mount the True Dual CBEs directly on the side of the cabinet, with secondary fluid connections facing outwards and refrigerant connections inwards. This facilitates the assembly of the refrigerant circuits and further minimizes piping for the secondary fluid circuit. As a result, the True Dual systems (an example is shown in Figure 6.33) have a lower assembly time and thus a higher production throughput. This is very important, especially for original equipment manufacturers (OEMs) with large production volumes.

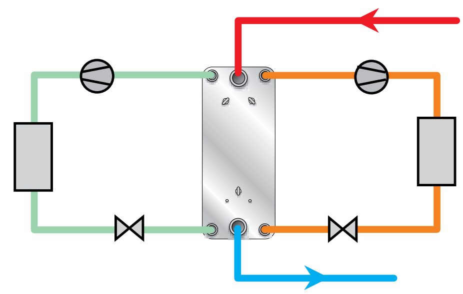


Figure 6.33 Example of a system solution with a True Dual evaporator.

Back-to-Back Dual

A Back-to-Back dual circuit CBE (see Figure 6.34) is based on two single circuit CBEs mounted in parallel, with a common water circuit divided between the two plate packs. The refrigerant circuits work independently and affect only the water channels in their respective plate packs. SWEP Back-to-Back duals are available with or without a distribution system.

Capacity control: For full load with both refrigerant circuits active, a Back-to-Back dual operates in the same way as a True Dual or single circuit CBE (see Figure 6.35). A Back-to-Back CBE operates at nominal load at high efficiency, with all secondary fluid channels in contact with refrigerant channels.

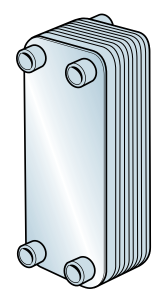


Figure 6.34 A Back-to-Back Dual CBE.

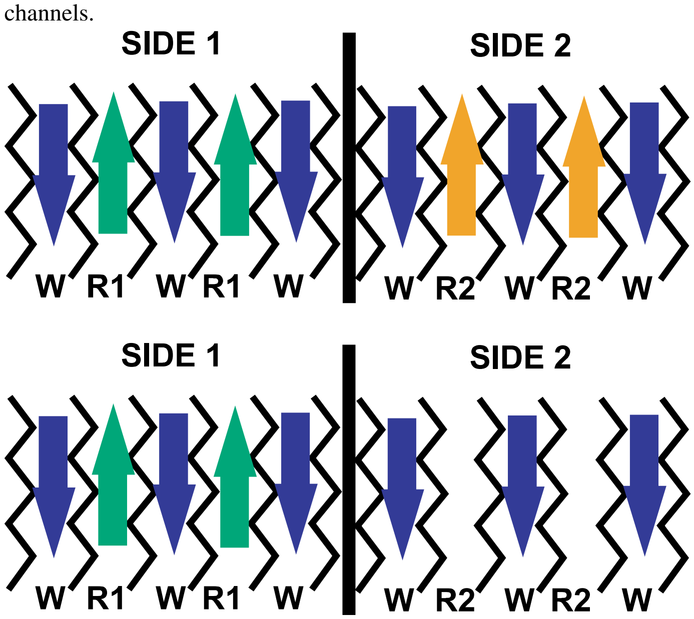


Figure 6.35 The flow channels in a Back-to-Back CBE. The upper flow channels show the CBE running on full load. The lower shows the CBE with only one refrigerant circuit active.

If one refrigerant circuit is closed, the secondary fluid channels in the inactivated plate pack will not be in contact with any refrigerant. The entering and leaving secondary fluid temperatures for this plate pack will therefore be the same, and the final temperature will be an average of the two secondary fluid streams leaving the CBE.

For a Back-to-Back evaporator with a fixed temperature for the leaving secondary fluid, the active side has to reach a much lower secondary fluid temperature to achieve an acceptable average leaving temperature for the secondary fluid. This will reduce the system performance due to the resulting drop in evaporation temperature, which is very likely to be below 0°C. This also dramatically increases the risk of freezing in those channels. For a Back-to-Back heat exchanger operating as a condenser, the decrease in system performance will still be visible but not as apparent as for the evaporator, because the high-pressure side has less influence on the system performance. There will of course be no additional risk of freezing on the condensing side.

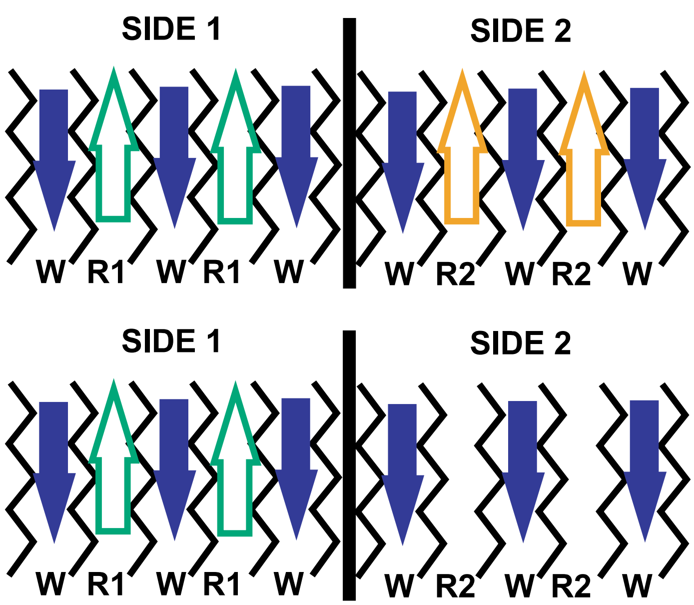


Figure 6.36 Tandem compressors are used in a Back-to-Back dual system for using all the heat surface area even when running on half load. In the lower flow channel picture, this is combined with the Back-to-Back possibility of using only one plate package.

Combining a Back-to-back dual with two sets of tandem compressors increases the performance and stability of operation at half load. All the heat surface area is utilized, and the leaving secondary fluid temperatures from both plate packs will be the same (see Figure 6.36). Precautions must still be taken to maintain the necessary channel and port velocities. However, trying to operate the Back-to-Back dual with one refrigerant circuit completely deactivated will again result in poor thermal performance and increased freezing risk, especially if the water flow is decreased to meet the part-load capacity. The reduced turbulence of the secondary fluid will combine with the decreased evaporation temperature, further increasing the freezing risk.

System Design: The Back-to-Back is less flexible than the True Dual in the arrangement of the water and refrigerant connections. Because the design is based on two parallel CBEs, brazed back-to-back, the refrigerant connections are always located on opposite sides of the unit (see Figure 6.37). The water connections will therefore always be adjacent to a refrigerant circuit, and more pipe routing is necessary to connect the refrigerant circuits to the compressors and the water circuit to the outside of the cabinet.

Nevertheless, the central water circuit will distribute the water evenly inside the Back-to-Back CBE, and no additional flow regulation is necessary to ensure an even flow over the two plate packs. Consequently, it is not possible to close one water circuit when operating at part load. The two refrigerant circuits of a Back-to-Back CBE are built with separate plate packs, and thus it is possible to design the unit for asymmetric chiller capacities. A higher number of plates in one plate pack will give proportionately more refrigerant and water channels.

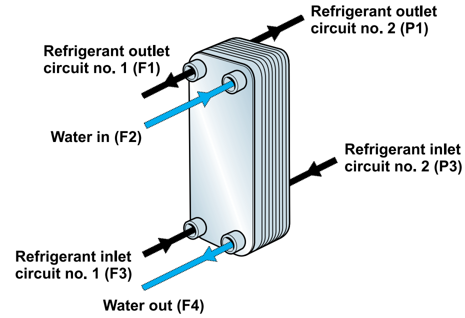


Figure 6.37 The connections and flows in a Back-to-Back CBE.

Single Circuits in Parallel

SWEP has a wide range of single circuit CBEs for refrigerant applications. The B-models are the standard type, while the V-models are equipped with the SWEP distribution device for evaporation duty. To take advantage of a dual refrigerant circuit system with single CBEs, two units can be arranged in parallel with the secondary fluid circuit divided between the two CBEs.

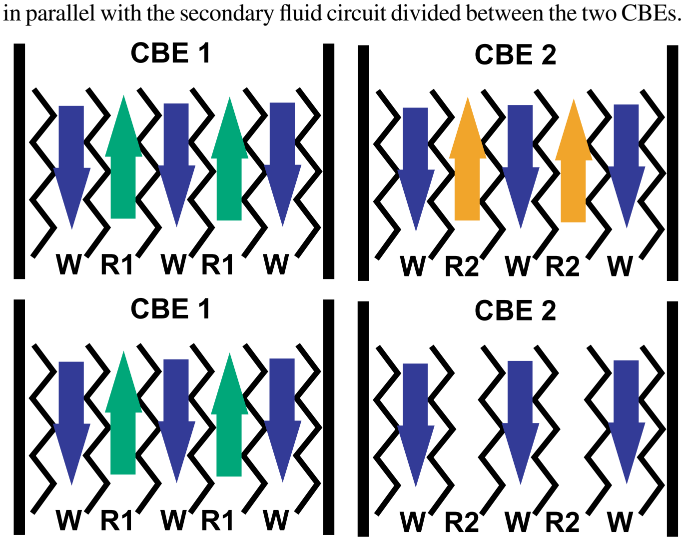


Figure 6.38 The upper flow channels show two single circuit CBEs in parallel, the lower the same CBE with one refrigerant circuit deactivated.

Capacity Control: At full load, two single circuit CBEs in parallel function as a Back-to-Back dual (see Figure 6.38). This arrangement requires that the water flow be divided proportionately before entering the CBEs. A high degree of symmetry in water piping is therefore necessary.

Deactivating one refrigerant circuit will leave the water in the inactive CBE without any heat exchange, similar to the operation of a Back-to-Back dual. The entering and leaving secondary fluid temperatures will be the same in the inactive CBE, and the final temperature will be the average of the two combined secondary fluid flows. The performance of the active refrigerant circuit will decline due to the consequent reduction of evaporation temperature.

By installing a flow control valve it is possible to completely close the water flow through the inactive CBE, thus maintaining high performance even at half load (see Figure 6.39). This arrangement requires a more complex pipe arrangement. In addition, the total water flow must be halved, because the pressure drop through the single active CBE will otherwise be squared.

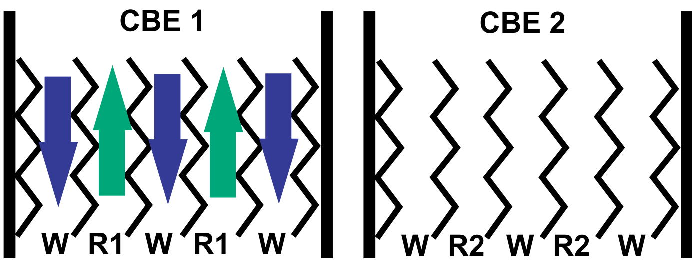


Figure 6.39 The same CBEs as in Figure 6.38, but with the refrigerant and water circuits shut off in the second CBE.

As with a Back-to-Back dual, it is possible to install two parallel CBEs with a variable compressor system (see Figure 6.40). All the heat surface area is then utilized even at half load, and the leaving secondary fluid temperatures from both plate packs will be the same. However, precautions must be taken to maintain the necessary channel and port velocities.

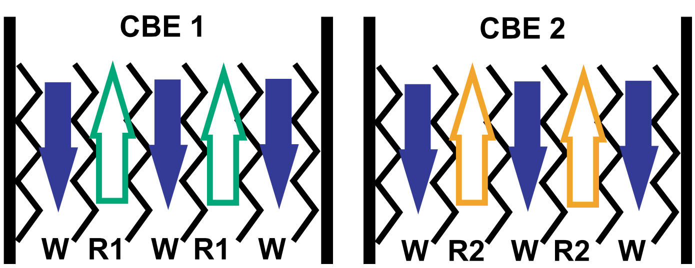


Figure 6.40 Two parallel CBEs with a variable compressor system.

System Design: Installing two CBEs in parallel (see Figure 6.41) to obtain a dual refrigerant circuit requires the most complicated piping of the methods discussed in this chapter. Although the refrigerant circuits can be assumed to work independently, the water circuit must be divided proportionately between the two units. A very high degree of symmetry in the piping is therefore required. Additionally, in most cases a flow control valve has to be fitted to adjust the water flow. Due to the relatively complex piping work involved, which leads to longer assembly times in the factory, and the extra component costs, the popularity of this arrangement has declined in favor of the True Dual technology. The system configuration for single circuits in parallel is also less compact compared with both Back-to-Back and True Dual systems.

Every CBE can be designed independently with refrigerant and water connections on the same or opposite sides. It is possible to have two differently sized CBEs connected in parallel for asymmetric heat loads, but

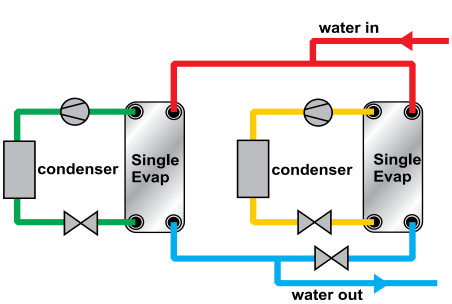


Figure 6.41 An example of a system solution with two single evaporators in a parallel system.

precautions must be taken to ensure that the water flow is proportionately divided between the units. Regulating valves are therefore often necessary to control the flow.

Single CBE with Two Compressors

A special arrangement is to use a single circuit CBE and to connect two or more compressors to the same refrigerant channel. This removes the safety aspect of having two separate refrigerant circuits. However, the advantages of such an installation are that the CBE always uses 100% of the heat transfer area during operation and that the piping for the secondary fluid is less complex than for two CBEs mounted in parallel.

Capacity Control: During full-load operation with both compressors (C1 and C2) running, the CBE will operate as a normal single circuit unit, assuming good symmetry between the two refrigerant streams.

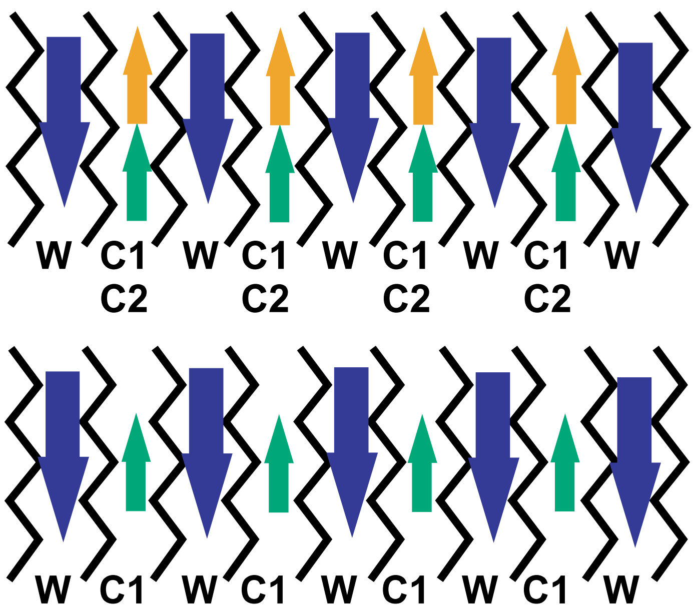


Figure 6.42 The upper flow channels show a single CBE with two compressors in use, the lower the same CBE but with only one refrigerant circuit activated.

Deactivating one refrigerant circuit still leaves all refrigerant channels active (see Figure 6.42). The water temperature is homogeneous, and no extra freezing risk is induced. The relatively large oversurface benefits the system performance due to the increased evaporation temperature. However, it also induces a potential risk of oil retention if the unit is too large to allow the single wactive refrigerant flow to carry the oil.

System Design: A typical application for the arrangement with one single CBE and two compressors is the use of tandem coupled scroll compressors, where the two refrigerant circuits are already combined before the evaporator (see Figure 6.43). Tandem compressors are currently available up to approximately 100 kW, but the capacity will grow with the increasing size of scroll compressors and the introduction of **triplet/trio compressors**.

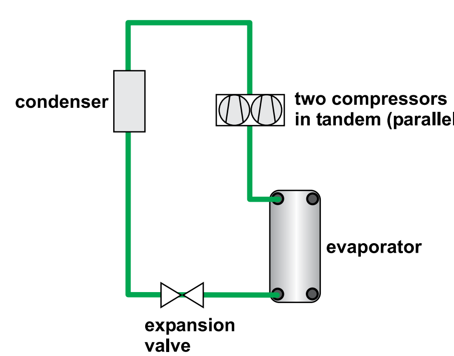


Figure 6.43 An example of a system solution using a single CBE with two compressors.

The piping for such a system is fairly uncomplicated, but precautions must be taken with the oil return after the evaporator when operating at half load. Tandem compressor sets are factory assembled. They can operate in parallel, and have a piping design that avoids problems with oil distribution between the two units.

Because there is only one expansion valve for the two refrigerant circuits, it must be able to operate over the full capacity span of both circuits. This in turn imposes very high demands on the expansion device, which has to be able to cope with changing mass flows and evaporation temperatures. It is often necessary to use nonstandard thermal expansion valves or electronic expansion valves. The high cost of these special expansion devices will partly or completely offset the capital saving of using only one evaporator. Taking into consideration the lack of two independently operating refrigerant circuits, such as for a dual system, it is understandable that this arrangement is uncommon.

6.9 Flooded Evaporators

An important difference between a flooded evaporator and a direct expansion (DX) evaporator is that the flooded evaporator operates in conjunction with a low-pressure receiver. The receiver acts as a separator of gaseous and liquid refrigerant after the expansion valve and ensures a feed of 100% liquid refrigerant to the evaporator. Unlike in a direct expansion (DX) evaporator, the refrigerant is not fully evaporated and superheated at the flooded evaporator outlet. The leaving refrigerant flow is a two-phase mixture with typically 50-80% gas.

Flooded evaporators, which are sometimes called wet evaporators, are divided into forced-flow evaporators and thermosiphon evaporators. Forced-flow evaporators use a pump or an ejector as the driving force, while the density difference between liquid and gaseous refrigerant drives thermosiphon systems.

The Thermosiphon Compression Cycle

In addition to the basic equipment in a direct expansion refrigeration circuit, i.e. evaporator, compressor, condenser and expansion valve, the flooded system needs a receiver (no. 1, Figure 6.44) to separate the two-phase mixture after the expansion valve (no. 5). The refrigerant leaving the bottom of the receiver is 100% liquid.

The refrigerant from the receiver enters the evaporator (no. 2, Figure 6.44) and evaporates due to the heat transferred from the secondary side. The refrigerant at the evaporator inlet is slightly sub-cooled due to the pressure increase from the receiver to the evaporator. After the evaporator, the two-phase refrigerant mixture again enters the receiver, where liquid and gas are separated. The gas then enters the compressor, while the remaining liquid is re-circulated through the evaporator. The gas is compressed in the compressor (no. 3) and condensed in the condenser (no. 4) in the same way as in the basic compression cycle. The force driving the refrigerant through the evaporator depends on the density difference

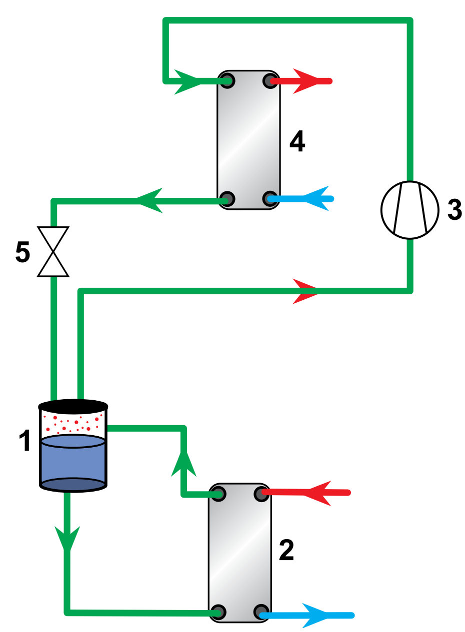


Figure 6.44 Sketch of the thermosiphon system principle.

between gaseous and liquid refrigerant. When refrigerant is evaporated inside the CBE, the lower density of the vapor allows more liquid refrigerant to flow inside the evaporator. Please note that the expansion valve needs no regulating action, because the flooded evaporator is self-regulating. The spontaneous vaporization in the receiver ensures that no liquid enters the compressor.

The Forced-Flow Compression Cycle

A forced-flow flooded system is identical to a thermosiphon system, except that a pump is installed before the evaporator to serve as a driving force for the refrigerant (see Figure 6.45).

If the installation site does not offer the minimum necessary height difference between the receiver and evaporator to allow density circulation, a forced-flow system may be preferable over a thermosiphon. The higher cost of a pump can still be more economical than elevating the roof of the installation room. Forced-flow systems often have a larger circulation number than thermosiphon systems due to the higher mass flow created by the pump.

Larger static head, i.e. a larger height difference between receiver and evaporator, increases the sub-cooling of the refrigerant. The preheating in the beginning of the evaporator is then increased, which may lead to the requirement of a larger evaporator, because much more heat transfer area is needed to preheat liquid instead of producing gas.

If the lubricating oil is insoluble in and heavier than the refrigerant, oil drainage can be installed before the pump. Oil droplets on the heat transfer surface may decrease the heat transfer dramatically.

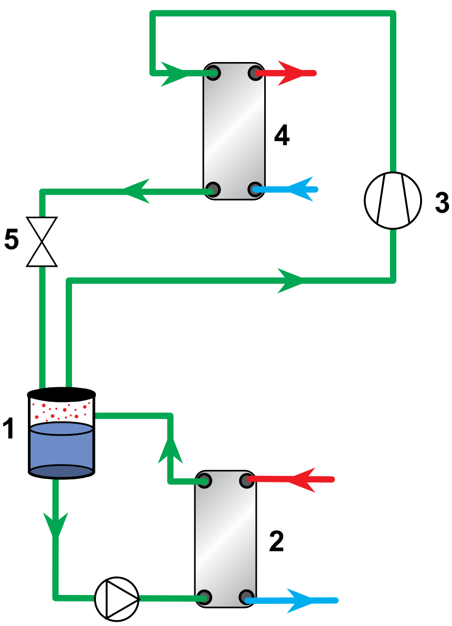


Figure 6.45 A forced-flow system.

Co-Current Versus Counter-Current Flow

For thermosiphons, it is often recommended that evaporation be operated in parallel flow, despite the mean temperature difference (MTD) being less favorable. Co-current flow ensures a large temperature difference in the beginning of the evaporator, enabling the boiling process to start as soon as possible. Because the thermosiphon evaporation process is driven by differences in density, the evaporator performance depends more on stable circulation than a large MTD. If the temperature difference between the refrigerant and secondary fluids is higher than 10K, counter-current flow may be the favored flow pattern.

Characteristics of Flooded Systems

An advantage of flooded evaporators is that the potential problem of poor refrigerant distribution in the evaporator is reduced. The refrigerant is 100% liquid, and a liquid stream is much better distributed between the channels compared with the two-phase mixture of DX systems. Thus, when selecting a flooded evaporator, a SWEP B-model should be used for flooded evaporators, in contrast to DX systems where V-models are the better choice.

The receiver separates the refrigerant vapor before feeding it to the compressor (see Figure 6.46), so there is no need for superheating the refrigerant in a flooded evaporator. A larger portion of the total heat surface area will thus be used for evaporation compared with a DX evaporator, where 10-30% of the total heat surface area may be dedicated to superheating.

Figure 6.46 compares a flooded system and a DX system. The green-line detour (b-c) to the liquid saturation line (bubble line) shows the phase separation in the flooded system receiver. Because of the pressure gain (c-d) in the pipe that connects the receiver and the flooded evaporator, the evaporator inlet liquid is sub-cooled. Please note that the (c-d) line in Figure 6.46 is exaggerated. The pressure gain in the receiver-evaporator connection is actually approximately 5-50 kPa (0.05-0.5 bar), while the pressure lift over the compressor is roughly 12-17 bar. Because there is no need for superheating in a flooded evaporator, the evaporation temperature can be a few degrees higher than in a DX evaporator (see Figure 6.47).

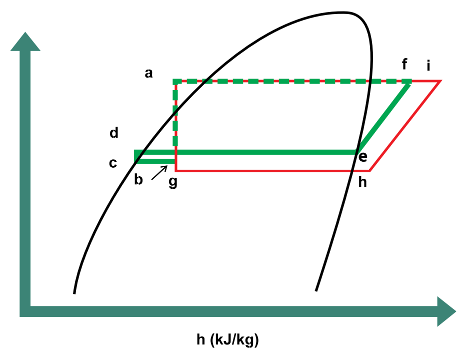


Figure 6.46 Flooded-flow refrigeration circuit shown in green. The red line symbolizes evaporation and compression in a DX evaporator designed for equal duty.

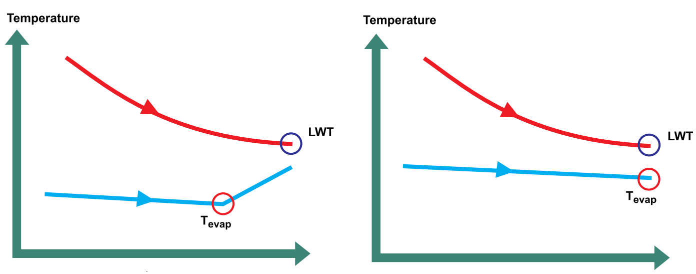


Figure 6.47 (a) Temperature profile through a superheated evaporator (DX). (b) Temperature profile through a non-superheated evaporator (Flooded). LWT=Leaving Water Temperature, i.e. the leaving secondary fluid temperature.

Energy transfer is much more efficient through a boiling turbulent liquid film than through dry superheated vapor. As a consequence, the temperature program is “closer” for a non-superheated evaporator than for a superheated evaporator (see Figure 6.47). A “closer” temperature program means a smaller difference between the leaving secondary fluid temperature and the evaporation temperature (LWT-T_{evap}).

Due to the higher evaporation temperature in a flooded evaporator, the pressure lift between the evaporator and condenser sides is smaller. The advantage is that less compressor work (W) is needed. Perhaps the largest advantage of flooded evaporators is that they use all the latent energy of the refrigerant in the phase transition between liquid and gas to cool a fluid. This is shown in Figure 6.46, where line (d-e) stretches over the whole transition length, while the red line (g-h) does not utilize the entire phase transition length. In other words, the COP (coefficient of performance) is higher for a flooded system.

[1] $$COP_{evap} = \frac{Q}{W} = \frac{\text{cooling capacity}}{\text{compressor work}}$$

When considering flooded systems, the **circulation number** is important. Because evaporation is incomplete, the two-phase mixture must flow through the receiver-evaporator circuit more than once to achieve 100% evaporation. The degree of evaporation at the evaporator outlet is stated as x_{out}, which is measured in kg vapor per kg total inlet mixture. The circulation number is defined as the reciprocal of the degree of evaporation:

$$\text{[2]} \qquad \text{CircNo} = \frac{1}{x_{out}}$$

The circulation number indicates how many times a certain liquid volume has to pass through the evaporator to be completely evaporated. A smaller circulation number indicates less pipework, a smaller receiver and a lower refrigerant charge in the flooded-flow circuit. The circulation number of a CBE is between approx. 1.1 (x_{out}=0.91) and 1.4 (x_{out}=0.71), and the circulation number of an S&T unit lies between 5 (x_{out}=0.2) and 10 (x_{out}=0.1).

Design

The driving force of the thermosiphon process is based on natural density differences. It is therefore of vital importance that the relationship between the **liquid static head** and the **two-phase static head** (see Figure 6.48) is correct, i.e. the driving force should be larger than the restraining forces. For the two-phase mixture to return to the separator, the static head pressure must be larger than the total pressure drop through the evaporator and reconnection pipe. The driving force can be calculated as:

$$p = \rho \cdot g \cdot h$$

Where:

p = Static Head Pressure [Pa]

ρ = Liquid Density [kg/m^3]

g = Gravitational Acceleration [9.81 m/s^2]

h = Height [m]

The restraining forces in the two-phase connection are more complicated to estimate, because the outlet flow from the evaporator is in two

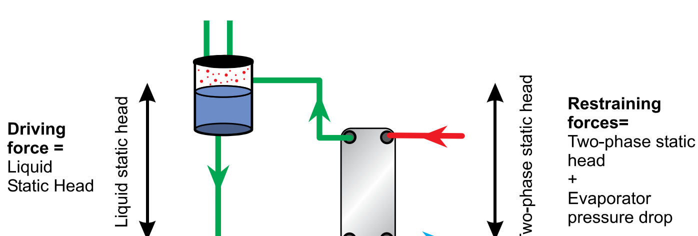


Figure 6.48 The balance of forces over a thermosiphon evaporator.

phases. The evaporator pressure drop is given in SSP. Table 6.2 is a receiver height converter for different refrigerants. The heights are calculated using the same formula as for the driving force.

Table 6.2 Examples of how the height between the receiver and evaporator varies with the static head pressure.

Static-head pressure (Pa)	Receiver height in meters		
	R22 at 0°C	**NH_3 at 0°C**	**R404a at 0°C**
5	0.40	0.80	0.44
10	0.79	1.60	0.88
15	1.19	2.39	1.33
20	1.59	3.19	1.77

Refrigerants with a large glide are not recommended for flooded evaporators. The refrigerant is not fully vaporized in a flooded evaporator, and the composition would consequently change for a refrigerant with glide. This would make it difficult to evaluate temperature changes throughout the system.

When are Flooded Systems Used? The advantages and disadvantages of flooded systems are shown in Table 6.3. Flooded systems are economic for large refrigerant systems, due to the lower requirements on power input to the compressor. For smaller systems, the pay-back time for the larger installation cost of a flooded-flow system is often considered to be too long despite the smaller power input.

Table 6.3 Advantages and disadvantages of flooded systems (compared with DX systems).

Advantages	Disadvantages
Higher COP in flooded systems.	Higher installation cost, because more equipment is needed (receiver, more pipes)
Better flow distribution into evaporator, because the inlet is	More refrigerant needed to avoid superheating in the evaporator. 100% liquid.
Better utilization of plate area, because there is no superheating zone.	More important with a low pressure drop in the evaporator, because it is driven by density differences.
The flooded evaporator is self-regulating.	More difficult to design, because the system is sensitive to deviations from the calculated refrigerant flows.

6.10 Heat Flux in Evaporators

A brazed plate heat exchanger is designed as an evaporator in SSP, the SWEP calculation program. The software calculates the film coefficients and the pressure drops from correlations describing the phenomena. An overall heat transfer coefficient is calculated, and the required heat surface is decided. The required area and the pressure drop limitations, together with the economic aspects, determine the heat exchanger model and the number of plates the duty requires. A simple way of evaluating different design calculations is to compare the heat flux of the CBEs. Heat flux can be regarded as the heat exchanger's density of heat transfer, and is defined as the heat flow per heat transfer surface:

$$Q = k \cdot A \cdot MTD \quad [3]$$

This equation is derived from the heat flow equation. The heat flow in a heat exchanger is:

[4] $$\frac{Q}{A} = k \cdot MTD \qquad [W/m^2]$$

Where:
Q = Capacity [W]
A = Heat Transfer Area [m^2]
k = Overall Heat Transfer Coefficient [$W/m^2°C$]
MTD = Mean Temperature Difference between the refrigerant and the heat transfer fluid [°C]

In chapter 1.5, the LMTD (logarithmic mean temperature difference) was introduced for single-phase calculations. The reason for using a logarithmic mean value is the logarithmic characteristics of the temperature profiles in a single-phase heat transfer process. In two-phase calculations, a so-called MTD must be used. The calculation of MTD is not shown in this handbook, because it is too difficult to calculate manually.

The k-value depends on the CBE characteristics and the flow profile inside the channels. Correlations integrated in SSP calculate the k-value for the temperature program used to design the CBE. For a given heat transfer surface, the k-value determines the temperature difference needed. A high k-value results in a closer temperature program, and a lower k-value means a higher temperature difference is needed.

A higher k-value means that a lower temperature difference is required between the refrigerant and secondary fluid. However, the mean temperature difference is often difficult to determine practically. An easier way of showing the difference in operating systems is to use the difference between the leaving secondary fluid temperature and the evaporating temperature ($LWT - T_{EVAP}$) (see Figure 6.49).

The characteristics of the CBE are predicted from the correlations in SSP. The k-value depends on the heat transfer media and the turbulence of the flow. It is easy to see from the heat flux equation that a larger temperature difference ($LWT - T_{EVAP}$) is required to achieve a higher heat flux in a pre-defined CBE. Figure 6.50 shows a typical curve of the heat flux as a function of ($LWT - T_{EVAP}$).

Figure 6.50 shows the typically sloped performance curve. The temperature difference between the evaporating medium and the secondary fluid increases with increased heat transfer per m^2 (heat flux). The slope is unique for each heat exchanger and refrigerant, as discussed below.

The Influence of Heat Exchanger Characteristics

The physical dimensions of the heat exchanger influence the film coefficient and thus the heat flux. The hydraulic diameter is twice the pressing depth, and will affect the flow because a smaller hydraulic diameter increases the k-value and also the pressure drop. The corrugation angle may be in a

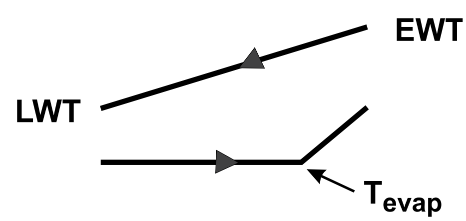


Figure 6.49 Definition of the evaporation temperature in an evaporator.

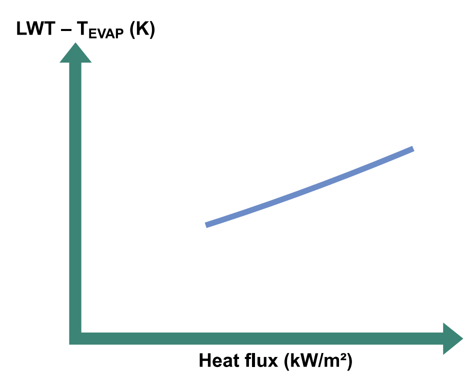


Figure 6.50 Heat flux diagram showing the correlations between the difference between the leaving water temperature and the evaporation temperature (which is the same as the saturation temperature, T_{EVAP}) and the heat flux. An increase in the temperature difference increases the heat flux transferred per m^2

specific range, approximately 30-80°. The larger the angle, the higher the film coefficient becomes, but a larger angle will also cause a higher pressure drop.

The Influence of Evaporating Fluid

The physical properties of refrigerant fluids differ between the liquid and vapor phases. Due to the differences in physical properties, the film coefficient and the heat flux will vary. Some important parameters are:

- Cp = Heat Capacity [J/Kg·°C]
- μ = Viscosity [Pa·s]
- λ = Thermal Conductivity [W/m·°C]

The easiest way of comparing the heat flux between different refrigerants is by using the heat flux diagram, similar to Figure 6.50. The different physical properties of R22, R404A and R134a require more or less heat surface area to achieve the same performance. Figure 6.51 shows SSP calculations to reach an evaporation temperature of 2°C (corresponding to LWT - T_{EVAP} = 5).

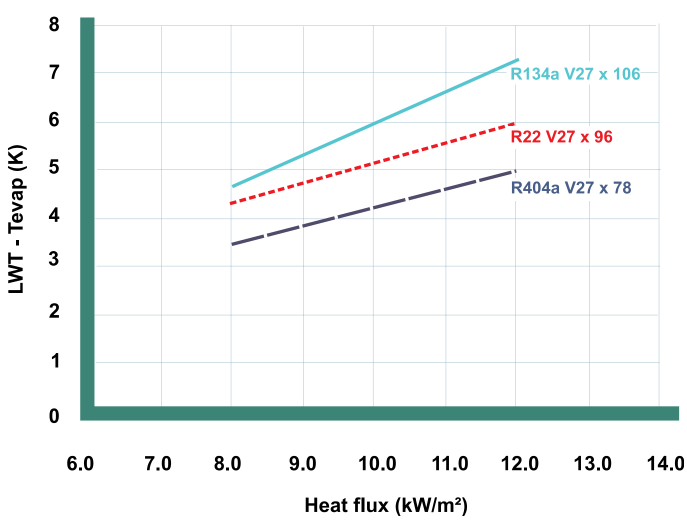


Figure 6.51 SSP calculation of heat flux with refrigerants R22, R404A and R134a.

As Figure 6.51 shows, R404A would need less area than R22 or R134a. For the same evaporation temperature (2°C), the heat flux is higher: 12 kW/m^2 compared with 9.5 kW/m^2 and 8.8 kW/m^2 for R22 and R134a, respectively.

6.11 Oil in Evaporators

The oil lubricating the compressor is always carried into the system to some extent, even if an oil separator is installed. To ensure lubrication of the compressor, the oil must be returned. Accumulation of oil in any part of the system could result in decreased functionality and/or efficiency. The solubility of oil in the refrigerant decreases with the temperature, whereas the viscosity and density increase. The evaporator is therefore critical, because it is the coldest point of the system. The flow in the evaporator is also generally upwards, which requires the oil to be lifted by the refrigerant flow. Because the evaporation temperature of the oil is much higher than that of the refrigerant, the oil must be carried out as droplets by the refrigerant vapor.

For normal concentrations (<3%) of oil inside the evaporator, the heat transfer coefficients will not vary significantly. A small concentration of oil may even affect the heat transfer positively. However, if oil accumulates, the thickening oil film will act as an insulating layer, and the evaporator efficiency will be noticeably reduced.

The most important parameters allowing the oil film to be carried are the refrigerant flow shear stress and the viscosity of the oil. It is difficult to give a precise minimum channel velocity to obtain a shear stress high enough to ensure oil return, because the geometries of the various models differ. The induced shear stress will be higher for a high-theta plate than for a low-theta plate at the same channel velocity. Consequently, a lower channel velocity is required for a high-theta plate than for a low-theta plate. The refrigerant vapor velocity in suction pipes to ensure oil return is often set to approximately 5-10 m/s, the higher value being for low temperature applications (<-20°C). The highly turbulent flow in CBE evaporators allows channel velocities of less than a tenth of these values with assured oil return. SSP warns of a risk to oil return for channel velocities less than 0.3 m/s. This warning should be a good indication of normal or high evaporation temperatures. However, for low evaporation temperatures (<-20°C), or for low-theta evaporators, the CBE should be designed with a higher channel velocity.

Some measures can be taken to improve oil return in potentially problematic systems, i.e. those with very low channel velocities and/or low evaporation temperatures. Increasing the channel velocity by decreasing the number of plates, or selecting a different CBE model, improves the shear stress. This may result in a larger temperature difference between the evaporating refrigerant and the secondary medium due to reduced heat transfer area. However, the increased pressure drop leads to a higher heat transfer coefficient, which may partly or completely compensate for the decreased heat transfer area.

The density of the oil is also important if the oil is insoluble in the refrigerant. If the oil is denser than the refrigerant, e.g. for NH_3 and hydrocarbon refrigerants, it will sink to the bottom of the evaporator. If the oil is less dense than the refrigerant, it will float on top of the refrigerant as soon as the flow settles, e.g. in receivers or similar. If the oil is soluble in the refrigerant, the density is not critical, because oil and refrigerant then form a single phase. It is important to remember that the solubility of oil decreases with decreasing temperature. An evaporator operating at low temperatures may have problems with insoluble oil even if the oil and refrigerant are considered soluble.

For systems with an increased risk of oil retention, a good oil separator is recommended to minimize oil accumulation. A more radical solution is to turn the evaporator upside down and evaporate downwards. The oil will thus flow out of the evaporator due to the force of gravity. It is then important to remember that for dedicated evaporator models, i.e. the V-line, the refrigerant must still enter through the F3 port because the distribution system is located there. The force of gravity also increases the risk of liquid carry-over. Arranging the piping to collect potential liquid before entering the compressor is recommended.

7 Condensers

The normal function of a condenser is to transform hot discharge gas from the compressor to a slightly sub-cooled liquid flow, by transferring heat from the refrigerant gas to the secondary cooling liquid. The basic operation of condensers is divided into three parts: desuperheating, condensation and sub-cooling. All three operations can be carried out inside the condenser. The total heat transfer is called the Total Heat of Rejection (THR). Alternatively, the desuperheating or sub-cooling operation can be carried out in a separate heat exchanger.

7.1 General Function and Theory

The refrigerant enters the condenser as superheated gas, i.e. at a temperature higher than the saturation temperature (point a in Figure 7.2). The heat rejection can be followed in a log P/h diagram. The first part of the condenser cools (desuperheats) the gas to the saturation temperature (a-b). This cooling represents 15-25% of the total heat of rejection. It is a one-phase heat transfer where the temperature of the refrigerant gas decreases typically by 20-50K, depending on the system and refrigerant. When the refrigerant reaches its saturation temperature, the latent heat is rejected and a liquid film forms on the heat transfer surface. The condensing process represents the majority (70-80%) of the total heat of rejection (b-c). Finally, the fully condensed refrigerant (c) is sub-cooled a few degrees (c-d) to ensure that pure liquid enters the expansion valve (d). This is also a one-phase heat transfer operation, representing approximately 2-5% of the total heat of rejection.

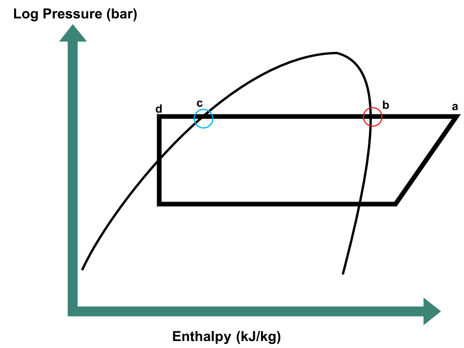


Figure 7.2 The refrigerant condensing operation explained in a log P/h diagram. The hot gas enters in (a) and is desuperheated until it reaches the saturation temperature (b). At the saturation temperature the temperature remains constant as refrigerant vapor condenses. At the bubble point (c), all refrigerant is condensed. By continuing to cool the condensate below the saturation temperature (subcooling), more energy is extracted. The condensate leaves the condenser at (d).

Temperature Profile Inside the Condenser

The temperature of the refrigerant decreases during the desuperheating and sub-cooling processes, but remains constant during the condensing process (see Figure 7.3). The energy rejected from the refrigerant heats the secondary medium, whose temperature thus increases.

The refrigerant pressure changes little from desuperheating to sub-cooling. In a similar way to evaporation, the only pressure difference between the entrance and the exit of the heat exchanger is the pressure drop. Because the flow velocity in a condenser decreases, the induced pressure drop is much lower than in an evaporator.

The temperature difference of the refrigerant between the condenser inlet and outlet is much larger than in an evaporator, due to the desuperheating. True counter-current flow in a plate heat exchanger makes it possible to utilize this temperature difference. The temperature on the secondary fluid side can be increased to approach or even exceed the condensing temperature. A temperature increase results in a smaller flow of the secondary fluid for the same heat load. This reduces the required pump capacity and the size of the **liquid cooler**. However, there is a "temperature pinch" that must be considered and avoided for stable operation, as discussed below.

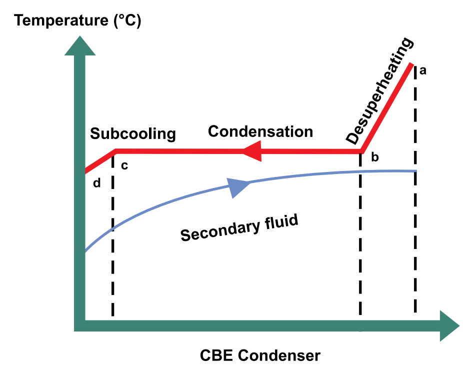


Figure 7.3 The refrigerant condensing operation in a temperature diagram. The hot gas enters (a) and is cooled during the desuperheating until the saturation point (b). During condensation (b-c), the temperature is constant at the saturation temperature. At the bubble point (c), all refrigerant gas has condensed and the temperature will again decrease during the subcooling (c-d).

In Figure 7.4, the inlet temperature of the secondary fluid is identical for the two cases, but the flow of curve (b) has been reduced to utilize the high gas temperature. The minimum temperature difference between the refrigerant and the secondary fluid in a counter-current condenser, the **pinch**, occurs at the beginning of the condensation process, as shown in Figure 7.4. The temperatures of the two media in a heat exchanger may converge but never equalize. The temperature of the leaving secondary fluid cannot therefore become more than a few degrees higher than the saturation temperature without "hitting the roof" in the tight section at the condensing temperature.

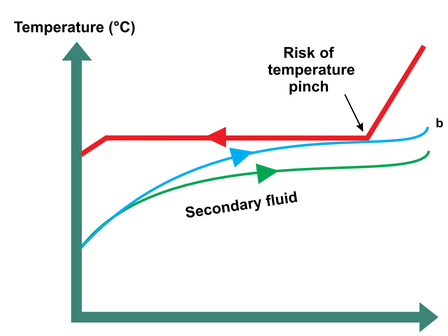


Figure 7.4 Condensing temperature profile for a normal temperature program (a) and a low water-flow temperature program (b).

Reducing the flow of secondary fluid exaggeratedly in an attempt to approach the temperature lines results in a heat exchange approaching zero. This greatly reduces the efficiency of the heat exchanger, and may result in only partial condensation and unpredictable performance.

The Effect of Pressure Drop

Pressure drop is created by the friction of the fluid, and is highly dependent on the fluid velocity. In a condenser, the refrigerant flow velocity is reduced as the refrigerant condenses, because the liquid phase has a much smaller specific volume than the gas phase. Hence, the major part of the pressure drop of the condenser is induced in the desuperheating operation, when the refrigerant is still a gas. Reducing the pressure results in a decrease of the saturation temperature, i.e. the level of superheating is increased. Normally, these effects are very limited. However, they are discussed further in chapter 7.6.

Counter-Current vs. Co-Current Flow

True counter-current flow is always preferred in a condenser for optimal utilization of the high desuperheating temperature. The mean temperature difference between the refrigerant and secondary fluid sides also becomes larger for counter-current flow, because there is no risk of the entering and leaving temperatures converging. However, the “temperature pinch” should still be avoided, as discussed above.

Operating in co-current flow achieves the minimum temperature difference between the leaving sub-cooled refrigerant and the leaving water, as shown in Figure 7.5. Not only do the approaching temperatures reduce the heat transfer. The already low heat transfer coefficient of the one-phase subcooling operation also results in very low heat transfer efficiency. A large extra surface is therefore required for a co-current plate heat exchanger compared with a heat exchanger operating with counter-current flow.

The sub-cooling must also be kept low for co-current condensers, because the converging temperatures greatly reduce the already low heat transfer coefficient.

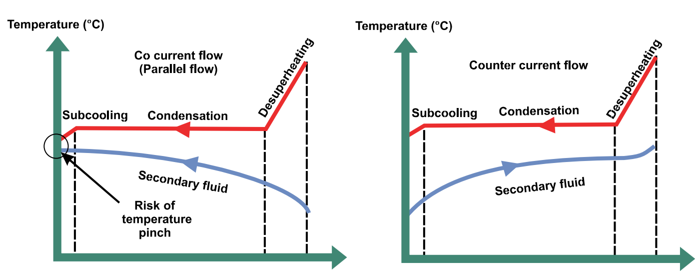


Figure 7.5 Temperature profiles for co-current and counter-current condensers.

Influence of Inert Gases

Incondensable, inert gases are normally not present in the system. However, they could be found in a system that has been unsatisfactorily evacuated before startup, from decomposed refrigerant or oil, etc. If inert gases are present in the system, they may accumulate in the condenser, resulting in phenomena that reduce the general performance. The incondensable gas may accumulate in a layer close to the heat transfer wall. This blocks direct contact of the refrigerant gas with the heat transfer surface. Instead, the refrigerant gas must diffuse through the inert gas layer. Furthermore, the partial pressure of the refrigerant gas decreases and the saturation temperature has to be lowered to compensate, resulting in a smaller temperature difference over the heat exchanger, as shown in Figure 7.6.

Venting

The condenser may act as an accumulator for inert gases because the refrigerant enters the condenser as a gas but leaves it as a liquid. Although all refrigerants except ammonia are heavier than the most common inert gases in refrigerant systems, i.e. air and carbon dioxide, the inert gases will be carried down towards the exit by the pressure drop in the channels. However, it is difficult for the gases to exit the condenser. If there is a condensate level inside the condenser, the incondensable gases accumulate in the gas/liquid interface and thus reduce the heat transfer efficiency. A venting purge should therefore be placed in the lower part of the condenser on the refrigerant side. To minimize the refrigerant loss, the purge may be cooled by cold low-pressure refrigerant gas to re-condense and collect the refrigerant.

If the pressure drop over the channels is too small, pockets of air or other incondensable gases may also form inside the condenser on the secondary fluid side. If the static pressure recovery due to the gas column inside the channels is larger than the dynamic pressure drop over the channels, there is a risk that a stable air pocket will form, as shown in Figure 7.7.

If the pressure drop is too small, all liquid will pass through the first channels, leaving the last channels without liquid. The air pocket can be emptied by means of a purge. An effective way of providing purge is by utilizing an extra connection outlet on the secondary fluid side, as can be seen in Figure 7.7. A simple mechanical float valve or equivalent ensures that only gas leaves the system.

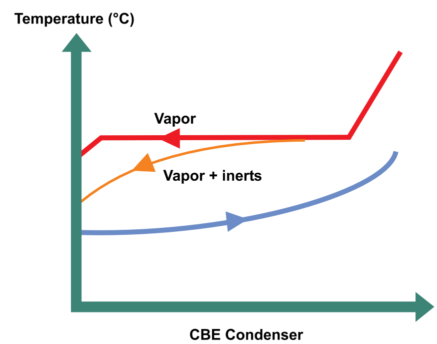


Figure 7.6 Condenser temperature profile in the case of inert gases accumulating in the condenser.

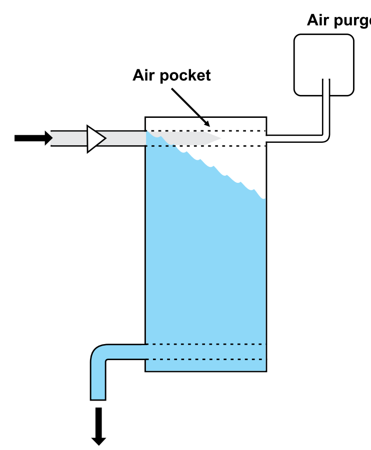


Figure 7.7 Stable air pocket in a CBE. The air pocket can be vented by the air purge. When the air has left and the liquid enters the purge, the float rises and closes the vent.

7.2 Sub-Cooling

Sub-cooling is carried out in the lower part of the condenser after all refrigerant gas has been transformed to liquid, as shown in Figure 7.8. The amount of energy transferred in the sub-cooling is small compared with desuperheating and condensation. The amount of energy released when sub-cooling one degree corresponds to approximately 1% of the total heat of rejection. Some sub-cooling is necessary to ensure that no gas is formed in the liquid line between the condenser and the subsequent component. A pressure drop in the liquid line reduces the pressure of the warm condensate liquid, with a risk of spontaneous boiling (flash gas) if there is little or no sub-cooling. Expansion valves are especially sensitive to flash gas, because it seriously disturbs their operation.

Applying additional sub-cooling has further advantages for the system. By extracting more energy from the condensing side, i.e. increasing the

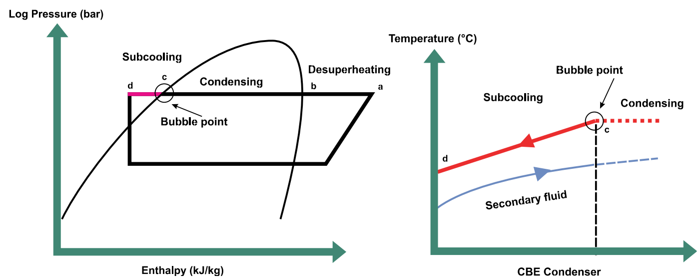


Figure 7.8 Detailed temperature program of subcooling and a log P/h diagram.

sub-cooling, the amount of flash gas formed after the expansion valve will decrease. Because the flash gas formed after the expansion valve does not contribute to the refrigeration effect, a larger amount of liquid refrigerant entering the evaporator increases the system performance. The improvement in the system performance, **COP**, is 0.5-2% per degree sub-cooling, depending on the type of refrigerant and the operating conditions.

In evaporators, just like condensers, the largest amount of energy is transferred by the **latent energy** when liquid is transformed to gas or vice versa. Increased liquid content therefore increases the total heat of absorption for the same mass flow and hence also the COP.

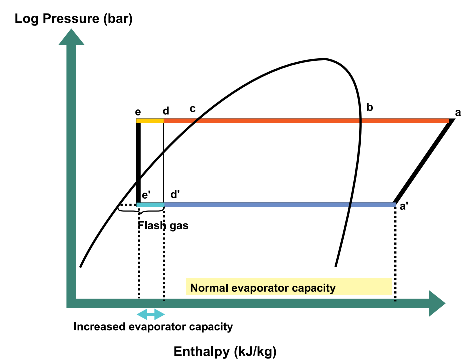


Figure 7.9 The effects of subcooling shown in a log P/h diagram. Increasing the subcooling from (c-d) to (c-e) increases the corresponding available cooling capacity inside the evaporator from (a'-d') to (a'-e'). The amount of flash gas formed after the expansion valve is also reduced, as shown in the figure.

Sub-Cooling in the Condenser

A plate heat exchanger is capable of sub-cooling the condensate a few degrees without requiring a considerable condensate level in the heat exchanger. This sub-cooling is sufficient to ensure that no flash gas is formed in the liquid line. However, if more sub-cooling is required, it is recommended that a separate heat exchanger, a sub-cooler, be fitted after the condenser. This is discussed further in chapter 7.3.

To achieve substantial sub-cooling with a plate heat exchanger condenser, it must operate with a variable condensate level. This may cause practical and stability problems as discussed below. An alternative is to vary the flow of the secondary fluid, but this solution is not commonly used because it is more complex and affects the condenser capacity.

Condensate Level Control

A solenoid valve placed in the condensate line normally controls the condensate level. When the valve opens, the condensate level is purged for a few seconds by the pressure in the condenser. It may take several minutes subsequently to return to the same condensate level, because the volume of the liquid condensate flow is very small compared with that of the gas. The valve should be placed as close as possible to the condenser outlet to minimize the pipe volume that must be filled before the condensate levels increase inside the condenser. Figure 7.10 shows a suggested arrangement for installing a control valve.

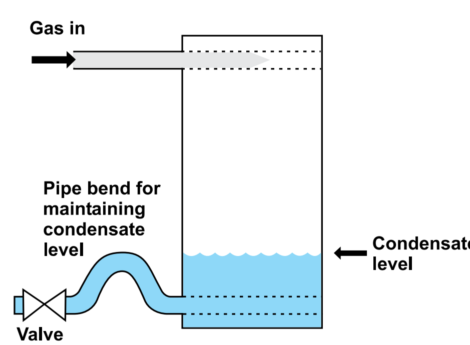


Figure 7.10 Example of arrangement for condensate subcooling. Inert and insoluble decomposition products accumulate above the condensate surface and are easily trapped there, with a resulting decrease in the heat transfer coefficient (k-value).

A sub-cooling operation is a one-phase heat exchange with low turbulence, due to the small volume of the liquid refrigerant flow. The area needed for sub-cooling is therefore relatively large. When the condensate level is purged, a large heat surface area that was previously used to sub-cool liquid refrigerant becomes available for condensing operation until the liquid level rises again. By expanding into the extra available heat transfer area, the condensation pressure and temperature decrease, which may affect the total heat of rejection. The rapid fluctuations in heat transfer area may also give control problems and fluctuating condensing performance.

Incondensable gases, and oils less dense than the refrigerant, can accumulate in the interface between the gas and liquid level in the condenser. These media reduce the heat transfer and have to be purged periodically with condensate.

Seasonal variations can also influence the condensate level. The low hold-up volume of a plate heat exchanger may be too small to compensate for this while maintaining the desired conditions. A surge liquid receiver must then be installed after the condenser to compensate for the varying refrigerant volume. It is important not to allow the leaving refrigerant liquid to reach equilibrium in a through liquid receiver, because this will remove all sub-cooling from the system. In this case, a separate sub-cooler should be installed after the receiver. Figure 7.11 shows the system outlines for surge liquid receivers and 7.12 for through liquid receivers.

7.3 Sub-Cooling the Condensate

In the condenser, the saturated refrigerant liquid is sub-cooled to a temperature below the **bubble point** to ensure pure liquid enters the expansion valve. If additional sub-cooling is desired, to increase the available evaporating energy, this is best carried out in a dedicated sub-cooler situated after the condenser. Three alternatives for dedicated sub-coolers are discussed below.

Water Sub-Cooler

The simplest CBE sub-cooler would be a heat exchanger where the liquid refrigerant is cooled with a water flow, which can be the same as for the condenser. If the water stream continues from the sub-cooler to the condenser, the total pressure drop equals the total pressure drop of the two CBEs, and should not exceed the maximum allowed pressure drop of the system. Figure 7.13 shows a system with a water sub-cooler.

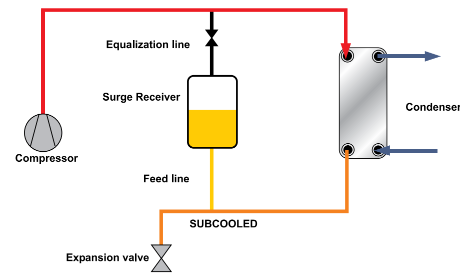


Figure 7.11 A surge receiver system. The surge type of receiver is normally bypassed by the flow and is useful for handling normal variations of refrigerant volume in the plant. With this type of receiver, it is possible to sustain subcooling after the condenser/subcooler.

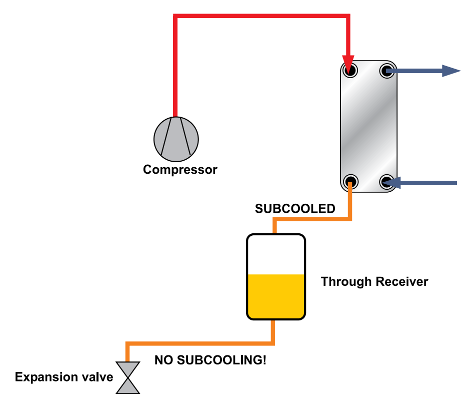
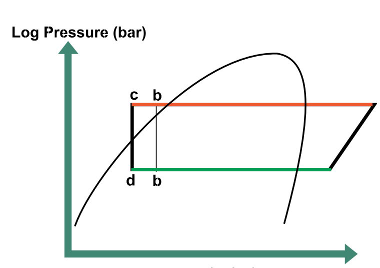


Figure 7.12 A through receiver system. The through type of receiver is placed in the flow direction of the refrigerant condensate. This type is useful for handling larger refrigerant fluctuations or for dumping refrigerant from other parts of the system. Because the vapor and liquid refrigerant are in equilibrium inside the through receiver, the leaving condensate will be saturated and NOT subcooled.



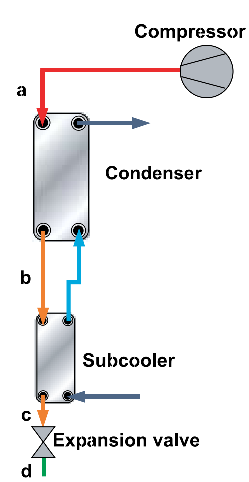


Figure 7.13. The right picture shows a subcooler system with combined water circuit, The left picture shows the system drawn in a log P/h diagram. The increased subcooling from (b) to (c) reduces the flash gas after the expansion valve and increases the available cooling capacity inside the evaporator.

Economizers

Another solution is to let part of the refrigerant flow evaporate and cool the remaining condensate liquid. The evaporated refrigerant then enters the compressor at an intermediate pressure level. This intermediate gas flow can also be used to provide extra cooling for the compressor, but condensate sub-cooling is the main application. The economizer system design requires extra components, piping and a compressor with an "economizer" entrance. The increased capital expense makes this system solution viable only for large refrigeration systems. Figure 7.14 shows a system with an economizer.

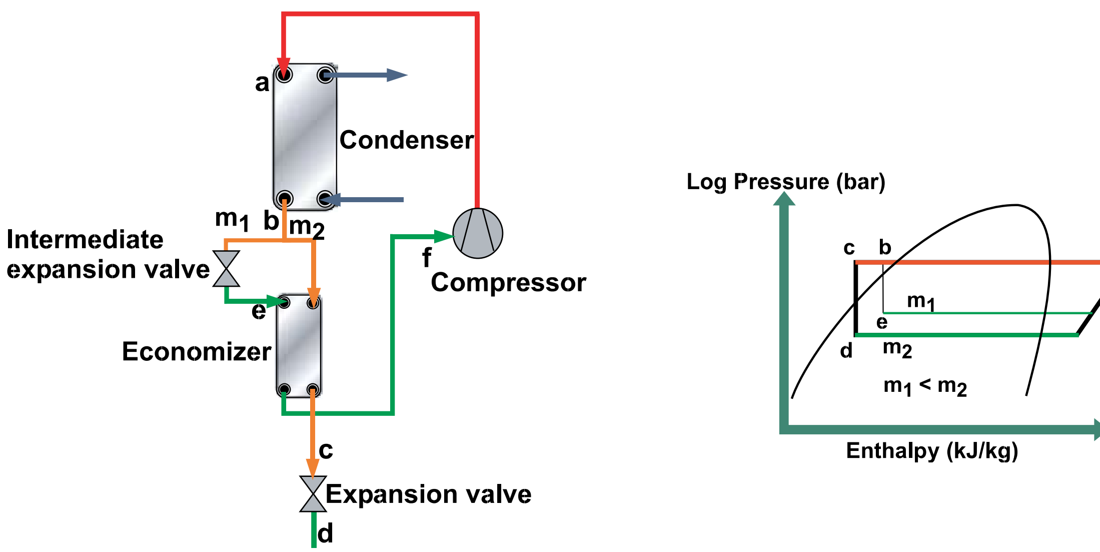


Figure 7.14 Economizer system with a log P/h diagram. The refrigerant flow is divided after point (b). The smaller flow (m_1) is evaporated at an intermediate pressure to cool the main flow (m_2). The cold intermediate gas (f) enters the compressor in a special economizer port where it also provides additional cooling for the electric motor.

Liquid Suction Heat Exchanger

A third method is to sub-cool the refrigerant liquid by utilizing the superheating of the gas leaving the evaporator. In a liquid suction heat exchanger, the cold gas that leaves the evaporator is used to reduce the temperature of the warm condensate liquid before the expansion valve, as shown in Figure 7.15. Because the refrigerant mass flow is the same on both sides of the CBE, the decrease in enthalpy of the condensate exactly corresponds to the enthalpy increase of the gas. If the level of gas superheating is too high, this may cause problems with elevated discharge gas temperatures, which may limit the level of sub-cooling.

However, there are advantages in "lifting out" the superheating from the evaporator to a liquid suction heat exchanger. The increased temperature

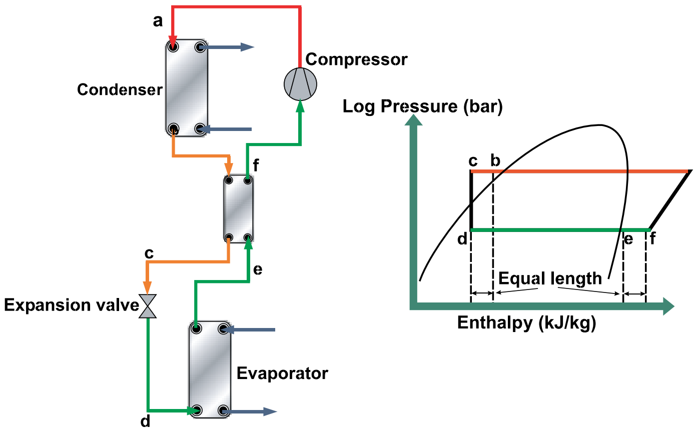


Figure 7.15 Liquid suction heat exchanger with a log P/h diagram. Because the mass flow though both sides of the suction heat exchanger is identical, the obtained subcooling (b-c) is exactly the same as the superheating obtained (e-f) in a log P/h diagram.

of the gas limits, or even eliminates, the negative influence of heat exchange between the refrigerant and the surroundings. The problem of water condensing on the suction pipes is also largely reduced. A similarly high level of superheating inside the main evaporator would require a much larger surface area than in the case of a liquid suction heat exchanger, due to the smaller temperature difference.

The disadvantages of liquid suction heat exchanger systems include the need for careful regulation of the expansion valve. Increased sub-cooling gives a lower pressure drop over the expansion valve and thus increases the mass flow. On the cold side, a higher gas temperature causes the expansion valve to open, because it is interpreted as if the mass flow passing through the evaporator were too low. The expansion valve must be regulated to avoid these two effects amplifying each other, which would lead to **hunting**.

7.4 Glide Refrigerants

Non-azeotropic, or glide, refrigerants are mixtures of two or more refrigerants where the components have different saturation temperatures at the same pressure level. When a glide refrigerant enters a condenser, the least volatile component condenses first. As the concentration of this low-volatile refrigerant decreases, the temperature of the remaining refrigerant mixture will also decrease, approaching the saturation temperature of the second least volatile refrigerant, and so on. Hence, the condensing temperature is higher in the beginning of the condenser than in the end, even if the condensing pressure remains constant. The temperature glide in a condenser is shown in Figure 7.16. This process is the opposite of a glide refrigerant in an evaporator, where the most volatile refrigerant evaporates first, at the lowest temperature.

In a condenser operating with glide refrigerants, there are three temperatures of special interest: the **dew point** (the highest condensing temperature), the mean condensing temperature and the **bubble point** (lowest condensing temperature), which is achieved just before all refrigerant has

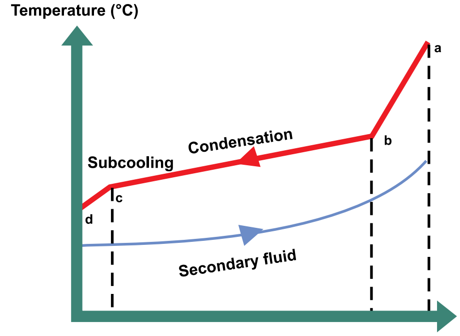


Figure 7.16 Temperature profile in counter-current condenser with glide refrigerant. Because the condensing temperature decreases, the risk of temperature pinching is avoided.

been transformed to liquid. The three temperatures are shown in Figure 7.17.

The available desuperheating should be considered as the difference between the inlet discharge gas temperature and the dew point. The subcooling achieved is calculated as the difference between the leaving temperature and the bubble point. The middle point is a purely theoretical temperature and is calculated as:

$$\frac{T_{DEW} + T_{BUBBLE}}{2} \quad (1)$$

To avoid misunderstandings in design discussions, it is important to be clear about which temperature is given. The initial condensing temperature, i.e. the dew point, is the one most commonly stated.

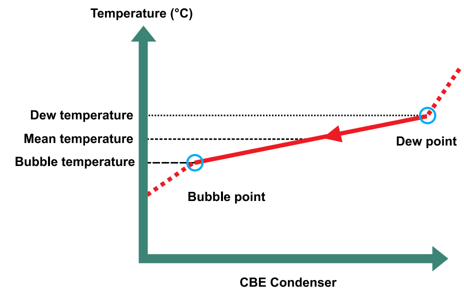


Figure 7.17 Dew, mean and bubble point condensing temperatures for refrigerants with glide.

Counter-Current vs. Co-Current Flow for Refrigerants with Glide

The decreasing condensing temperature of a glide refrigerant improves the already favorable temperature program over a plate heat exchanger operating in counter-current mode. The initial condensing temperature of the least volatile refrigerant is higher than the final condensing temperature. The increasing temperature of the secondary fluid will therefore not approach the temperature pinch as described above (see Figure 7.18). The mean temperature difference will increase, resulting in a more efficient heat transfer process. This additional heat transfer efficiency may be utilized by (1) increasing the outlet temperature of the secondary fluid, (2) decreasing the heat transfer area, i.e. reducing the number of plates, or (3) lowering the condensation pressure.

A co-current condenser with a glide refrigerant will instead have an impaired temperature profile, as described in Figure 7.18. The decreasing refrigerant and increasing water temperatures will converge at the condenser outlet, resulting in very poor heat transfer. To avoid the need for an "infinite" heat exchanger with a very high number of plates, the temperature difference between the refrigerant and secondary fluid sides must be increased.

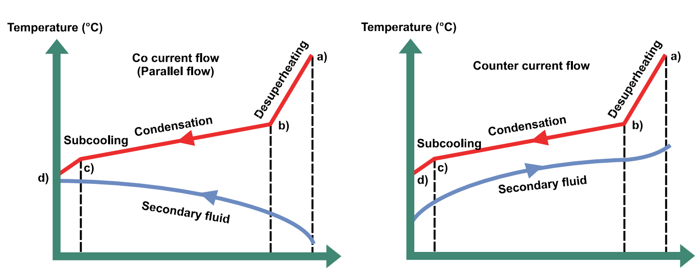


Figure 7.18 Temperature profiles in a counter-current and a co-current condenser with a "glide" refrigerant. For counter-current flow, the temperature convergence is avoided thanks to the decreasing condensation temperature. However, for co-current flow, the temperature difference between condensate and secondary fluid decreases rapidly. A higher saturation temperature of condensing may be necessary to maintain condensation.

The sub-cooling must also be kept low for a co-current condenser, because the converging temperatures greatly reduce the already low heat transfer coefficient. More sub-cooling requires a very large heat transfer area, leaving a smaller surface available for the condensation of the gas. Thus, the condensation pressure and temperature increase, and the system performance (COP) decreases.

Potential Problems with Glide Refrigerants

Non-azeotropic (glide) refrigerants are reported to have lower heat transfer coefficients than azeotropic refrigerants, because of the two-way mass transfer of refrigerant molecules from the heat transfer area to the bulk gas. The refrigerant gas condenses at the heat transfer area, which creates a constant underpressure that attracts more refrigerant. However, the more volatile refrigerants do not condense if the temperature is too high. There will therefore be an increased concentration in the gas of non-condensing refrigerant. This must diffuse from the heat surface into the bulk gas to allow more condensable refrigerant to access the heat transfer area, as shown in Figure 7.19. The mass transfer resistance results in a lower heat transfer coefficient. This can be overcome by a higher temperature difference between the refrigerant and the secondary side, similarly to the problem of inert gases discussed above.

The favorable temperature program of a true counter-current condenser counteracts this negative effect in a way that is impossible for cross-flow heat exchangers such as Shell and Tubes (S&T). In an S&T heat exchanger with the refrigerant on the outside of the tubes, the most volatile refrigerant may even accumulate and cause problems similar to inert gas, with low general heat transfer and high system pressure. This is generally not a problem in plate heat exchangers, because all refrigerant is forced through by the channel geometry.

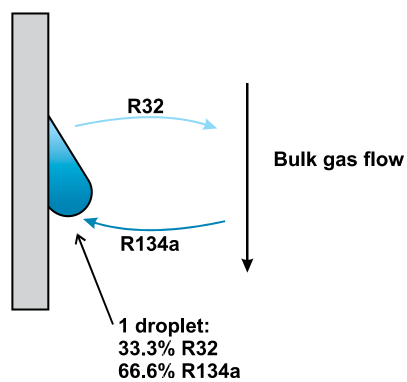


Figure 7.19 Accumulation of non-condensing volatile refrigerant. The formation of the first condensate droplet in a mixture of 50% R32 and 50% R134a (by mass). The mass transfer is perpendicular to the bulk gas flow in the figure.

7.5 Reversible Systems

Reversible refrigerant systems are designed to provide cooling and heating from the same heat exchangers, depending on the requirements of the recipient. The same CBE thus acts both as an evaporator and as a condenser. The condenser-evaporator duty is switched by a four-way reverse valve, as shown in Figure 7.20.

Because the refrigerant flow is reversed when changing from evapora-

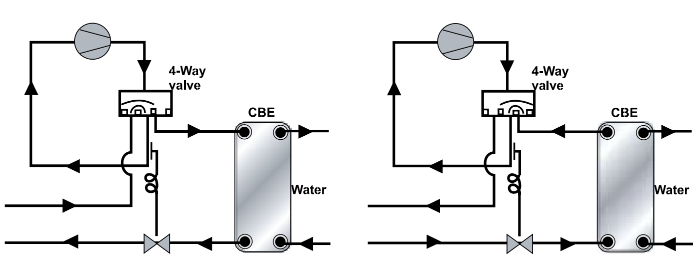


Figure 7.20 The figure to the left shows a counter-current condenser operation. The figure to the right shows a co-current evaporator operation.

tor to condenser, while the water direction remains constant, the CBE cannot operate in counter-current flow for both duties. When deciding whether the condenser or the evaporator should work with a co-current flow pattern, some design parameters must be taken into account:

- The main operating mode: If the reversible system is to be installed in a warm climate, it is more important that the evaporator operates in the most efficient way, and it should thus be designed for counter-current flow. For a cold climate, the operating efficiency of the condenser as a heat pump becomes more important, and the condenser should be designed for counter-current flow.
- If one of the heat exchangers is of the air coil type, the selection of flow pattern through the CBE is affected. Air coils operate with cross-flow, so they are less sensitive to reversed flow even if their efficiency is reduced when operating in reversed mode. The most efficient operation for the CBE should therefore determine the normal flow pattern.

In Figure 7.20, the evaporator works in co-current mode and the condenser in counter-current mode. Normal performance can thus be expected for the condenser. Operating the evaporator with co-current flow would usually give a lower level of performance than with counter-current flow, due to a lower LMTD**.** In this specific case, however, the evaporator performs almost as in a normal counter-current application, for the following reason: part of the superheating is created by the relatively hot reverse valve, which acts as a small heat exchanger in the suction line. Because the expansion valve bulb must always be connected after the reverse valve, for protection against high gas temperatures in condenser duty, the evaporator works with a very low level of superheating. This gives nearly the same performance as a counter-current case with "normal" superheating. A disadvantage of controlling the refrigerant flow this way is that the expansion valve reacts very slowly to changes.

Evaporator Models as Condensers

CBEs mounted in reversible systems must operate both as evaporators and as condensers, as mentioned above. The CBE performance in evaporator duty is often vital for the system economy. It is therefore common to install a dedicated CBE evaporator with a distribution device in the lower left refrigerant inlet port (F3). When the system is reversed, the evaporator operates as a condenser instead, with the hot gas entering in the top left port (F1) and leaving as liquid through the lower left port (F3), where the distribution device is situated.

One may wrongly suspect that the distribution device, which induces a high pressure-drop when operating as an evaporator, disrupts the operation of the condenser. However, because the leaving refrigerant flow is 100% liquid, which has a much smaller specific volume than gas, the pressure drop through the distribution device is negligible when the heat exchanger operates as a condenser.

7.6 Desuperheaters

Desuperheaters, or recuperaters, operate by utilizing the high-pressure, superheated discharge gas temperature to heat water. The cooling of the discharge gas normally represents 15-25% of the total heat of rejection. The temperature difference between inlet and outlet gas is typically 20-50K, depending on the refrigerant and system condensation temperature (see Figure 7.21). The refrigerant gas is not meant to condense to any significant extent inside the desuperheater, so the heat exchanger should be designed to have a leaving gas temperature up to a few degrees above the condensing temperature.

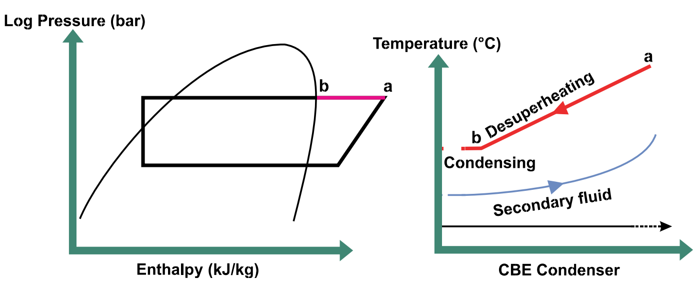


Figure 7.21 Desuperheating temperature profile and a log P/h diagram.

The advantage of using a separate desuperheater is shown in Figure 7.22. Even if the available energy of gas desuperheating is lower than for condensing, it is possible to obtain hot water with a small heat exchanger because the "temperature pinch" is avoided. By utilizing only the **sensible energy** of the gas, the leaving water temperature can approach the inlet discharge gas temperature. In contrast, in a condenser the constant condensation energy makes it impossible to obtain water temperatures more than a few degrees above the condensation temperature.

Desuperheaters can be used in dedicated heat pumps to provide hot tap water, while the condensation energy is used for room heating. In a refrigeration or air conditioning system, where the condensation energy is often discharged to the ambient air or to a low-temperature water sink, the usefulness of the system can be increased by producing hot water for cleaning, sanitary purposes or for heating other processes.

The desuperheater unit can be used as a "once through" heat exchanger, where cold tap water is heated directly when it is needed (see Figure 7.23). Thus, when no hot water is needed the desuperheater will be bypassed and the condenser will reject the desuperheating energy as well. The drawback with this design is that the output of hot water is limited by the momentary capacity of the system.

If instead the desuperheater is installed with a tank for hot water, this reservoir allows a momentary use of hot water that is much faster than the direct capacity of the desuperheater (see Figure 7.24). The hot water is stored in a tank and circulated by a small pump. If the temperature in the water tank becomes too high, a valve closes and the desuperheater is bypassed.

Drainage of Desuperheaters

Ideally, the desuperheater should be fitted above the condenser to allow any condensed refrigerant liquid to be drained away. This is not always

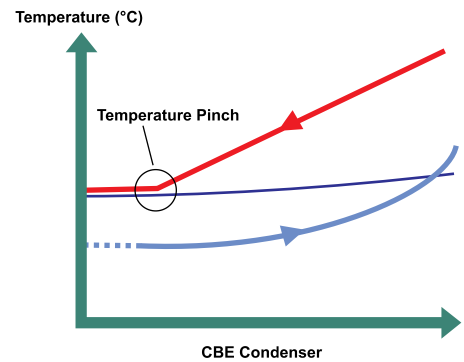


Figure 7.22 A temperature program inside a CBE condenser with a low secondary flow is compared with a CBE desuperheater. The similar temperatures in the condenser reduce the heat transfer. In contrast, the pinch is avoided in a desuperheater and the heat transfer remains efficient.

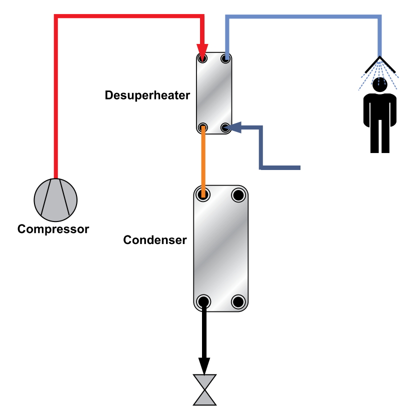


Figure 7.23 A "once through desuperheater system" without a receiver tank for heated water.

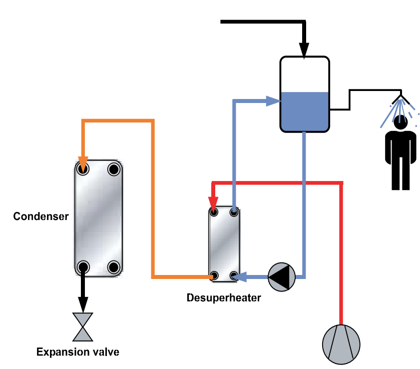


Figure 7.24 A desuperheater system with a receiver tank to build up a buffer with hot water.

possible, especially in refrigeration or air conditioning systems where the main condenser may be fitted on the roof, while the desuperheater is placed close to the compressor in the basement. This problem is even more evident with the use of glide refrigerants, such as R407C. If partial condensation takes place, the condensate has a higher concentration of the least volatile refrigerant. It is therefore even more important to ensure drainage and condensate transport to the condenser. Otherwise, there is a risk of the refrigerant components separating inside the system, resulting in unpredictable and impaired performance.

The construction of a CBE works in favor of keeping the liquid and gas phases together. In a brazed heat exchanger, the leaving droplets of refrigerant are dispersed and easily carried away by the dominant gas phase. Designing the connection pipe from desuperheater to condenser for a gas velocity of 5-10 m/s provides sufficient turbulence to avoid liquid condensate accumulation.

Scaling Problems

The solubilities of calcium carbonate (limestone, $CaCO_3$) and calcium sulfate (gypsum, $CaSO_4$) decrease as the water temperature increases. As a result, there may be deposits of these two minerals on heated surfaces, e.g. in a desuperheater. This phenomenon is called scaling. If scaling occurs, the extra crust of mineral will have the same effect as fouling, i.e. heat transfer is impaired and the capacity of the desuperheater is decreased. Unless the water flow is adequately turbulent, there is also a risk that pockets of stagnant water will form, encapsulated by the crust. In unfavorable conditions, these pockets could act as reaction points for pitting corrosion of the stainless steel. The problem is small with soft water, but if the deposits are allowed to accumulate or hard water is used, precautions should be taken to avoid or minimize the problem.

Avoiding Scaling

To avoid or minimize the risk of problems induced by scaling, such as decreased capacity, increased water pressure drop or even leakage, three parameters can be manipulated: temperature, turbulence and chemical properties.

Problems with deposits of limestone and gypsum occur only at water temperatures exceeding 65-70°C. Even if the inlet refrigerant gas temperature is much higher than 70°C, only the water temperature influences the risk of scaling. The design temperature of the water should therefore never exceed 65°C at the warmest point. Because the maximum water temperature is achieved at the heat transfer surface inside the heat exchanger, this value should be monitored in the calculation program.

If the CBE design becomes substantially over-dimensioned, due to considerations of pressure drop or other factors, it may be difficult to avoid a dangerously high leaving water temperature. In that case, it may be possible to design a desuperheater with parallel flow instead. The converging temperatures of refrigerant gas and water will efficiently limit the maximum water temperature.

Turbulent flow reduces the probability of scale, because the induced shear stresses can dislodge the newly formed deposits from the heat surface walls as harmless particles that are easily transported out of the CBE. A higher shear stress increases the efficiency of this self-cleaning effect. Hence, it is important always to operate the CBE at turbulent flow. The transition point between laminar and turbulent flow in a CBE is difficult to define, because the flow constantly changes direction inside the winding channels. An absolute minimum value of the Reynolds number to achieve turbulent flow in CBEs is Re = 150. A much higher Reynolds number should be used if a high outlet temperature or the quality of the water increases the risk of scaling.

If scale has formed, it is still possible to remove the minerals by the **cleaning in place (CIP)** method. A tank with a weak acid is connected to the circuit, and the acid is circulated with the water flow in the reverse direction (back-flush). Five-percent phosphoric acid is sufficient. If the heat exchanger is cleaned frequently, the slightly weaker 5% oxalic acid can be used. The acid dissolves the alkaline minerals, allowing them to be removed by the flow.

8 Practical Advice

When using a CBE in refrigerant systems, some considerations have to be made. Some are related to the refrigerant application itself, others are related to the CBE as a specific component. This chapter offers practical advice on utilizing the CBE optimally and on avoiding potential problems related to refrigerant applications.

8.1 CBE Construction and Denomination

There are several different types of Compact Brazed Heat Exchangers (CBEs), depending on their material combinations, pressure ratings and functions. The standard material is stainless steel, vacuum-brazed with pure copper or nickel-based filler. The basic construction materials indicate the types of fluids that SWEP's CBEs can be used with. Typical examples are synthetic or mineral oil, organic solvents, water (not seawater), glycol/water mixtures (e.g. water/ethylene glycol and water/propylene glycol) and refrigerants (e.g. HCFC). Please note that if ammonia is employed, CBEs with nickel-based brazing material must be used.

The front plates of SWEP's CBEs are marked with an arrow, either on an adhesive sticker or embossed in the cover plate. The purpose of this marker is to indicate the front of the CBE and the location of the inner and outer circuits/channels. With the arrow pointing up, the left side (ports F1, F3) is the inner channel and the right side (ports F2, F4) is the outer channel. The outer circuit has a lower pressure drop because it contains one more channel. The inner circuit consequently has a slightly higher pressure drop. Ports F1/F2/F3/F4 are situated on the front of the heat exchanger (see Figure 8.1). Ports P1/P2/P3/P4 are situated on the back. Note the order in which they appear.

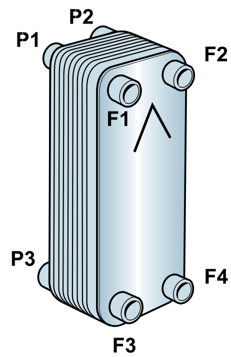


Figure 8.1 A SWEP CBE with ports marked.

Construction

In principle, the CBE is constructed as a plate package (of corrugated channel plates) between the front and rear cover-plate packages. The cover plate packages consist of sealing plates, blind rings and cover plates (see Figure 8.2). The type of connection can be customized to meet specific market and application requirements. During the vacuum-brazing process, a brazed joint is formed at every contact point between the base and the filler material. The design creates a heat exchanger consisting of two separate channels or circuits

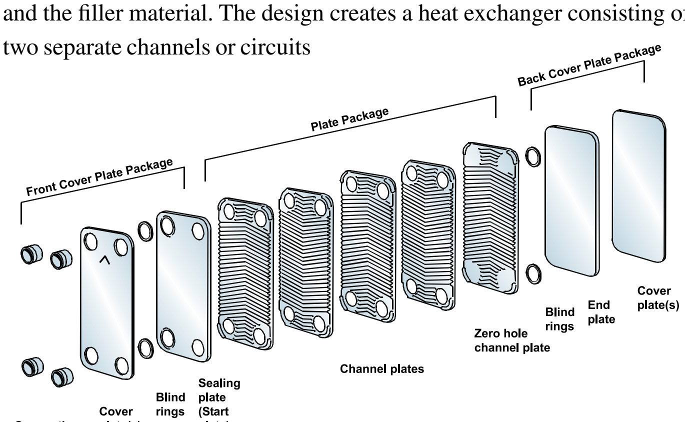


Figure 8.2 A SWEP CBE plate package.

Sealing plates are used to seal off the space between the cover plate and the first and last channel plates. The number of cover plates varies, e.g. with the CBE's type, size and pressure rating. Some CBEs have a blind ring to seal off the space between the channel plate and the cover plate. In some CBEs, the blind rings are integrated in the cover plate and first/last channel plates.

Material Combinations

There are different types of CBE product categories depending on their material combinations and design pressures (see Figure 8.3). They are defined as standard CBEs, all-stainless CBEs, Mo-steel CBEs and high-pressure CBEs. The standard plate materials are stainless steel, **S**, of AISI 316 type, vacuum-brazed with a pure copper filler, **C**, or a nickel-based filler, **N**. Carbon steel can be used to some extent, e.g. for certain types of connections. For demanding applications, the plates can be made of SMO 254, a stainless steel with a higher content of molybdenum, **M**. CBEs are available in standard pressure rating, **S**, or high pressure rating, **H**. The material and pressure denominations are shown below and in Table 8.1.

B35Hx40/P-(X)(Y)-(Z)

Where:
(X) is the plate material (S=stainless steel, M=Mo-steel)
(Y) is the braze material (C=copper, N=nickel alloy)
(Z) is the pressure rating (S=standard pressure, H=high pressure)

Table 8.1 Examples of CBEs with various materials and design pressures.

CBE Category	Denomination	Explanation
Standard	B25H/1P-**SC-S**	B25 with stainless steel plates brazed with copper. Standard pressure rating.
High-pressure	B25H/1P-**SC-H**	B25 with stainless steel plates brazed with copper. High pressure rating.
All-stainless	B25H/1P-**SN-S**	B25 with stainless steel plates brazed with nickel alloy. Standard pressure.
Mo-steel	B25H/1P-**MC-S**	B25 with Mo-steel plates brazed with copper. Standard pressure rating.

CBE Plates and Channel Types

Some CBEs are available with different types of channel plates, i.e. with different herringbone patterns. The chevrons, i.e. the V-shaped pattern on the plates, can be obtuse (creating a high-theta plate) or acute (creating a low-theta plate) (see Figure 8.3). The thermal characteristics of the CBE can be modified by mixing high- and low-theta plates. For example, it is possible to construct a CBE with the same pressure drop on both sides despite different flow rates.

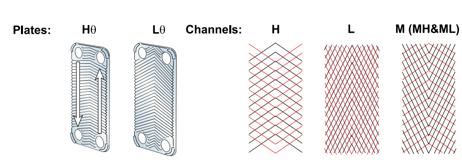


Figure 8.3 Plates with a low theta pattern (Lθ) and high theta pattern (Hθ). By putting two H -plates together (turning one 180°) gives the H-channel type. Doing the same with two L-plates gives the L-channel type. Taking one of each gives the M-channel type.

Flow Configurations

The fluids can pass through the heat exchanger in different ways. For parallel-flow CBEs, there are two different flow configurations: co-current or counter-current (see Figure 8.4).

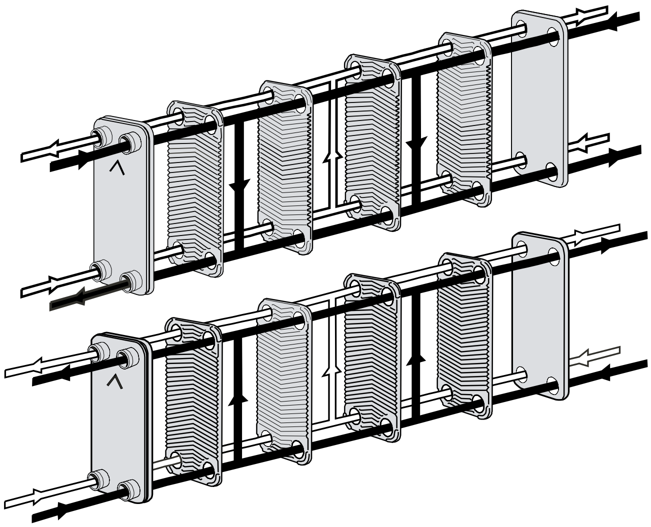


Figure 8.4 Counter-current and co-current flow in a CBE.

There are several different versions of the channel plate packages. Some examples are shown in Figure 8.5. These examples are further explained in the chapter 1.7 for Two-pass and 10.1 for the Duals.

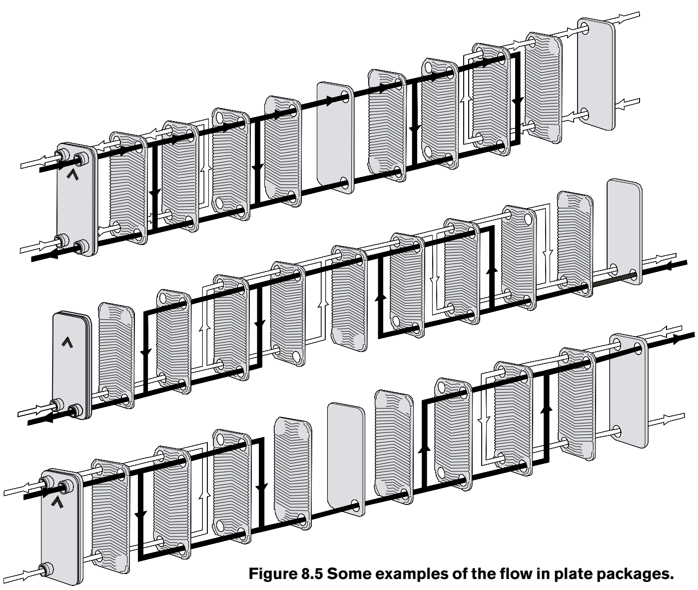


Figure 8.5 Some examples of the flow in plate packages.

Design Conditions and Approvals

The standard pressure rating for SWEP CBEs, i.e. maximum operating pressure, is 31 bar. SWEP's standard maximum operating temperature is 185°C for copper-brazed CBEs, and 350°C for all-stainless CBEs (nickel-based filler). However, because temperature and pressure are closely coupled, it may be possible to increase the pressure if the temperature is reduced. For details, please check the label (cf. Figure 8.6) and other technical documentation.

SWEP's CBEs are approved by a number of independent bodies, e.g.:

- Canadian Standard Association (CSA)
- The High Pressure Gas Safety Institute of Japan (KHK)
- Underwriters Laboratories (UL) – USA
- Pressure Equipment Directive (PED) – Europe

SWEP also has design approvals, from e.g.: Lloyds Register (LR), Great Britain; Det Norske Veritas (DNV), Norway; American Bureau of Shipping (ABS), USA; Korean Register of Shipping (KR), Korea; Registro Italiano Navale (RINA), Italy.

Labeling System and Operating Conditions

All CBEs carry an adhesive label (see Figure 8.6) with vital information about the unit, e.g. the type of heat exchanger, and SWEP's serial number. This indicates the basic CBE model. The operating conditions state the maximum operating temperature and pressure as determined by the various approving organizations. The label also includes the serial number (see Figure 8.7). The engraved serial number provides information about where and when the CBE was produced, etc.

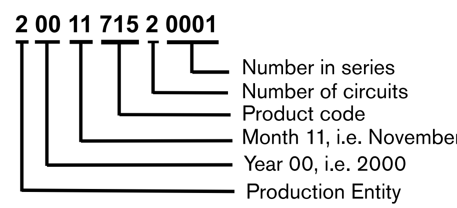


Figure 8.7 Explanation of the CBE serial number.

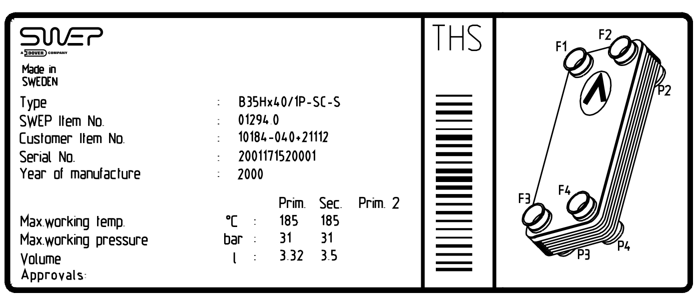


Figure 8.6 A CBE label.

8.2 Installation Guide

General mounting advice

Never expose the unit to pulsations or excessive cyclic pressure or temperature changes. It is also important that no vibrations are transferred to the heat exchanger. If there is a risk of this, install vibration absorbers. For large connection diameters, the use of an expanding device in the pipeline is recommended. It is also suggested that a rubber mounting strip, for example, should be used as a buffer between the CBE and the mounting clamp.

In single-phase applications, e.g. water-to-water or water-to-oil, the mounting orientation has little or no effect on the performance of the heat exchanger. In two-phase applications, however, the orientation of the heat exchanger becomes very important. In two-phase applications, SWEP's CBEs should be mounted vertically, with the arrow on the front plate pointing upwards.

Several mounting suggestions for SWEP CBEs are shown in Figure 8.8. Mounting stud bolts (see Figure 8.9) in various versions and locations are available as an option on CBEs. These stud bolts are welded to the unit. For smaller

CBEs, it is also possible to mount the unit by simply suspending it from the pipes/connections.

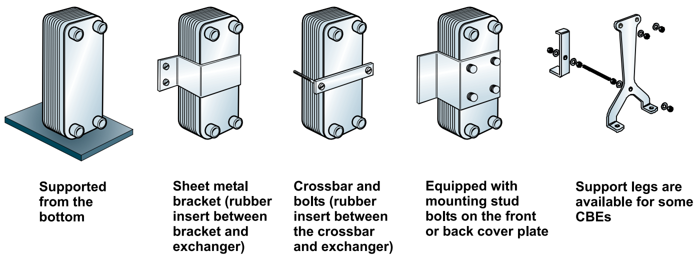


Figure 8.8 Different mounting suggestions for SWEP CBEs.

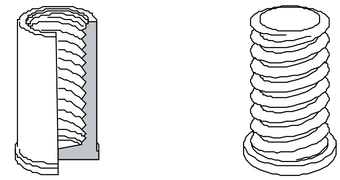


Figure 8.9 Stud bolts.

Connections in general

All connections are brazed to the heat exchanger in the general vacuum brazing cycle. This process gives a very strong seal between the connection and the cover plate. However, take care not to join the counterpart with such force that the connection is damaged.

Depending on the application, there are many different versions and locations available for the connections, e.g. Compac flanges, SAE flanges, Rotalock, Victaulic, threaded connections and welding connections (see Figure 8.10). It is important to have the correct international or local standard of connection, because they are not always compatible.

Some connections have an external heel (see Figure 8.12) to simplify the pressure and leakage testing of the CBE in production.

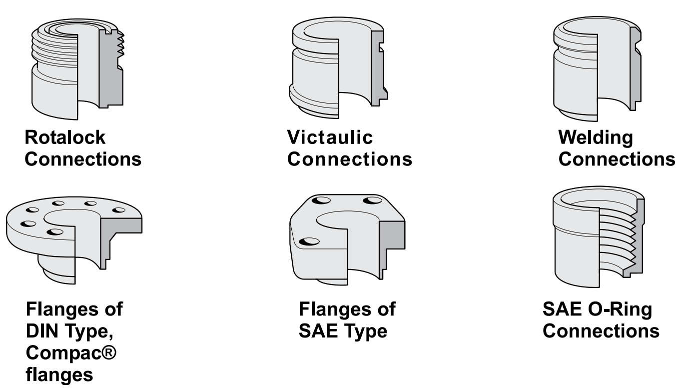


Figure 8.10 Examples of available connections.

Some connections are fitted with a special plastic cap to protect the threads and sealing surface (see Figure 8.11) of the connection, and to prevent dirt and dust from entering the CBE. This plastic cap should be removed with care, to prevent damage to the thread or any other part of the connection. Use a screwdriver, pliers or knife.

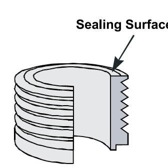


Figure 8.11 The sealing surface of a connection.

Threaded Connections

Threaded connections can be female or male (see Figure 8.12), in well-known standards such as ISO-G, NPT and ISO 7/1.

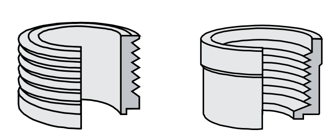


Figure 8.12 An externally threaded connection (male) to the left, and an internally threaded connection (female) to the right. Both connections are of standard type.

Soldering Connections

The soldering connections (sweat connections) (see Figure 8.13) are in principle designed for pipes with dimensions in mm or inches. The measurements correspond to the internal diameter of the connections. Some of SWEP's soldering connections are universal, i.e. fit both mm and inch pipes. These are denominated xxU, such as the 28U, which fits both 1 1/8" and 28.75 mm.

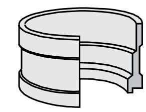


Figure 8.13 Example of a soldering connection.

All CBEs are vacuum-brazed with either pure copper filler or nickel-based filler. Under normal soldering conditions (no vacuum), the temperature should not exceed 800°C. The material's structure can be altered if the temperature is too high, resulting in internal or external leakage at the connection. It is therefore recommended that all soldering uses silver solder containing at least 45% silver. This type of solder has a relatively low soldering temperature and high moistening and fluidity properties.

When soldering flux is used to remove oxides from the metal surface, this property makes the flux potentially very aggressive. Consequently, it is very important to use the correct amount of soldering flux, because too much may lead to severe corrosion. No flux should be allowed to enter the CBE.

It is important to degrease and polish the surfaces when soldering. Apply chloride flux with a brush. Insert the copper tube into the connection and braze with minimum 45% silver solder. Point the flame towards the piping and braze at max. 650°C. Avoid internal oxidation, e.g. by protecting the inside of the refrigerant side with nitrogen gas (N_2).

Welding Connections

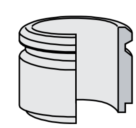


Figure 8.14 Example of a welding connection.

Welding is recommended only on specially designed welding connections (see Figure 8.14). All SWEP's welding connections are made with a 30° chamfer on top of the connection. Do not weld pipes on other types of connections. The measurement in mm corresponds to the external diameter of the connection.

During the welding procedure, protect the unit from excessive heating by:

a) using a wet cloth around the connection.

b) making a chamfer on the joining tube and connection edges.

Use **TIG** or **MIG/MAG** welding. When using electric welding circuits, connect the ground terminal to the joining tube, not to the back of the plate package (see Figure 8.15). Internal oxidation can be reduced by using a small nitrogen flow.

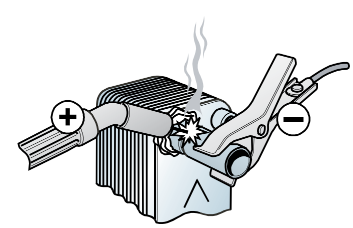


Figure 8.15 When welding a connection, connect the ground terminal to the joining tube.

Strainers

If any of the media contain particles larger than 1 mm, a strainer (see Figure 8.16) with a size of 16-20 mesh (number of openings per inch) should be installed before the exchanger. The particles could otherwise block the channels, causing low performance, increased pressure drop and risk of freezing. Some strainers can be ordered as CBE accessories.

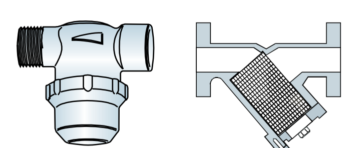


Figure 8.16 Pictures of strainers.

Insulation

CBE insulation (see Figure 8.17) is recommended for evaporators, con-

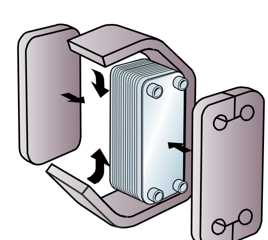


Figure 8.17 Insulation of a CBE.

densers and district heating applications, etc. For refrigeration, use extruded insulation sheets, e.g. Armaflex or equivalent, which can also be supplied by SWEP.

Single-Phase Applications

Normally, the circuit with the higher temperature and/or pressure should be connected on the left side of the heat exchanger when the arrow is pointing upwards. For example, in a typical water-to-water application, the two fluids are connected in a counter-current flow, i.e. the hot water inlet in connection F1 and its outlet in F3, and the cold water inlet in connection F4 and its outlet in F2 (see Figure 8.18). This is because the right-hand side of the heat exchanger contains one more channel than the left-hand side, and the hot medium is thus surrounded by the cold medium to prevent heat loss.

In single-phase applications, e.g. water-to-water or water-to-oil, the mounting orientation has little or no effect on the performance of the heat exchanger. This means that it is possible to mount the unit horizontally, on its side or back, without affecting performance. However, make sure that no air is trapped inside the heat exchanger when it is on its side.

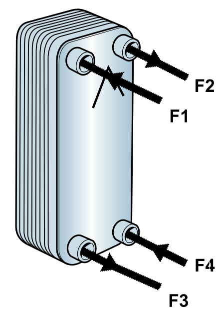


Figure 8.18 Inlets and outlets of a CBE with counter-current flow.

Evaporators; V-Type CBEs

In all refrigerant applications, it is very important that every refrigerant channel is surrounded by a secondary fluid channel on both sides. Normally, the refrigerant side must be connected to the left-hand side and the secondary fluid circuit to the right-hand side of the CBE (see Figure 8.19).

If the refrigerant and secondary fluid connections are transposed, the evaporation temperature will fall, with the risk of freezing and very low performance. SWEP CBEs used as condensers or evaporators should always be fitted with adequate connections on the refrigerant side.

V-type CBEs are equipped with a special distribution device at the refrigerant inlet, i.e. normally port F3. The purpose of the distribution device is to distribute the refrigerant evenly in the channels.

The refrigerant liquid should be connected to the lower left connection (F3) and the refrigerant gas outlet to the upper left connection (F1). The secondary fluid circuit inlet should be connected to the upper right connection (F2) and the outlet to the lower right connection (F4).

In two-phase applications such as evaporators, the orientation of the heat exchanger becomes very important. For evaporators, SWEP's CBEs should be mounted vertically, with the arrow on the front plate pointing upwards.

Although evaporators normally have the refrigerant inlet in the bottom, it is possible to evaporate downwards. This means that refrigerant will enter the evaporator in connection F1 and gas will leave the evaporator from connection F3. In these circumstances, performance will be reduced. Ensure that no refrigerant liquid enters the compressor. This installation may also lead to control problems.

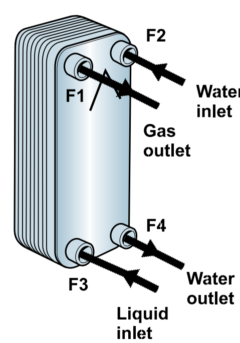


Figure 8.19 Connections for an evaporator.

For **DX-evaporators**, counter-current flow is the normal flow arrangement because it results in the highest MTD. For **flooded evaporators**, co-current flow is a common flow arrangement because a high inlet temperature difference is needed to initiate the evaporation process.

In cases of very **high outlet port velocity**, double outlet connections may be needed. This will decrease the velocity and pressure drop in the outlet port and thus increase the performance of the evaporators.

In the case of **parallel** installed **evaporators** with one compressor, it is important to make sure that the pressure drops in the suction lines are the same. This is to minimize maldistribution and prevent poor performance.

For evaporators, it is practical to measure the water temperature inside the heat exchanger. This can be achieved by equipping the evaporator with sensor connections on the back of the evaporator (P2/P4). The sensor connections are internally threaded where a temperature sensor can be attached. It is important here to ensure that the sensor is long enough to reach at least the middle of the port.

Condensers

As with evaporators, the orientation of the heat exchanger is also very important for condensers. For condensers, SWEP's CBEs should be mounted vertically, with the arrow on the front plate pointing upwards.

In a SWEP condenser, the refrigerant gas should be connected to the upper left connection, F1, and the condensate to the lower left connection, F3 (see Figure 8.20). The secondary fluid circuit inlet should be connected to the lower right connection, F4, and the outlet to the upper right connection, F2.

Counter-current flow is the normal flow arrangement, resulting in the highest MTD. Condensers can be tilted with some performance loss. The condenser is normally less sensitive to tilting compared with evaporators, for which the performance losses are significant.

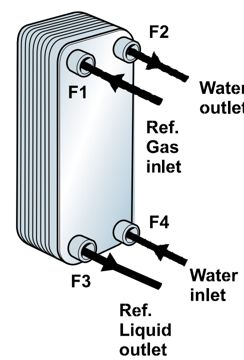


Figure 8.20 Connections for a condenser.

8.3 Recommended Port Connections

Field experience and extensive laboratory tests carried out by SWEP International's R&D department have shown that the size of the connections and pipes on the refrigerant side of the CBE can have an important effect on the performance of the CBE and the system in total. Selecting the correct connections and corresponding pipes can help to improve the performance and stabilize the evaporation process.

Evaporation is a complicated two-phase process where the two-phase fluid must be evenly distributed over the channels as well as over the plate itself. Great efforts have been made to optimize efficiency with special plate patterns and distribution devices. Figure 8.21 shows the CBE and the immediate components, i.e. connections and pipes, mentioned in the text.

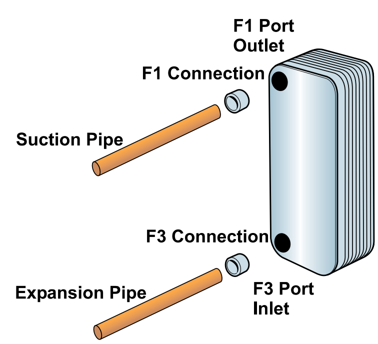


Figure 8.21 Basic outline of a SWEP CBE evaporator.

Evaporator Inlet Pipe and Connection (F3)

SWEP's distribution device is most effective when liquid and vapor are a homogeneous mixture at the entrance of the CBE. Directly after the expansion valve, vapor and liquid are completely mixed. They will stay in this homogeneous form for a reasonable time provided the flow velocity is high enough to create the necessary turbulence. If the velocity is too low, i.e. the pipe dimension is too large, phase separation will occur more quickly. The refrigerant flow into the evaporator then becomes divided into one fast vapor stream and one slower liquid stream, giving less predictable performance. Using a pipe that is too small induces unnecessarily high pressure-drops and results in energy losses.

The inlet connection selected should never be larger than the inlet port diameter of the F3 port, because this increases the risk of phase separation. Due to the distribution device, the inlet port size (F3) is smaller in a V-type evaporator than in a B-model.

The refrigerant vapor leaving the evaporator should have sufficient velocity to carry the small volume of compressor oil circulating in the system. Otherwise, the oil will accumulate and adhere to the channel walls and dramatically reduce the heat transfer coefficient. This results in a lower evaporation temperature and reduced system capacity.

If the velocity on the suction side of the CBE becomes high, the induced pressure drop can cause problems. In the suction pipe, velocities above 25 m/s will lead to energy losses, thus lowering the total COP for the system. If the port velocity becomes too high, the induced port pressure drop will cause maldistribution of refrigerant inside the CBE. A large pressure drop will also amplify the pressure difference over the plate pack, increasing the risk of boiling instability.

Conclusion: The recommended velocity in the expansion pipe (F3) is 10-25 m/s and in the suction pipe (F1) maximum 25 m/s. If the suction pipe velocity exceeds 25 m/s, the resulting pressure drop will start to affect the distribution measurably, and a change to a larger pipe dimension is highly recommended. The expansion valve should normally be placed 200-300 mm from the inlet. However, with correctly dimensioned piping, the distance can be increased without large losses of performance.

Recommended dimensions for the inlet pipes and F3 connections for standard chiller specifications (to give an inlet vapor/liquid velocity of 10-25 m/s) are given in the Appendix in the CD-ROM version of this handbook. Suggestions for positioning the thermal expansion valve (TEV) are shown in Figure 8.22.

Figure 8.23 shows the recommended position of the expansion valve relative to the inlet port of a SWEP CBE evaporator. The best position is at the same level as the evaporator inlet or higher. If this is not possible, the selection of the correct pipe size and distance from the TEV to the inlet port becomes more critical.

Different solutions for the soldered evaporator inlet connection have different consequences:

- A small inner diameter will lead to increased vapor and liquid velocities.
- A large pipe connected to a smaller inner diameter inlet will help to return liquid and vapor to a homogeneous mixture.

The inner connection diameter should never be larger than the inlet port diameter (see Figure 8.23).

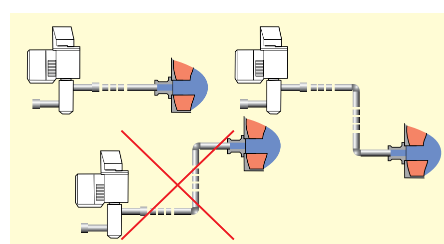


Figure 8.22 Recommended positioning of the expansion valve.

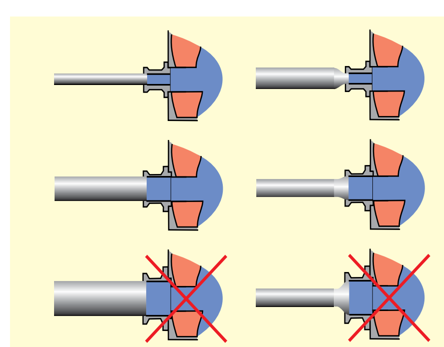


Figure 8.23 Suggestions for different inlet (F3) connections. The inlet connection can be smaller than the inlet port (top) or the same size (middle), but never larger (bottom).

Evaporator Outlet Pipe and Connection (F1)

The vapor velocity in the suction line should be sufficiently high to allow oil return to the compressor, but not so high that the induced pressure drop disturbs the evaporation inside the CBE. A guideline is that the vapor velocity in the suction line should be maximum 25 m/s. Recommendations for connections and pipe sizes for R22, R407C and R134a are given in the Appendix in the CD-ROM version of this handbook.

If the vapor velocity exceeds 25 m/s, the large pressure drop in the port may lead to a measurable decrease in efficiency. A further increase in the connection and suction pipe dimensions will reduce the pressure drop in the suction pipe. However, the risk of maldistribution and instability problems in the evaporator remains. SWEP recommends changing to a larger model, using double vapor exits or oversurfacing the CBE to compensate for the decreased efficiency.

Figure 8.24 shows outlet connection and piping variations. Versions (d) and (f) are considered immediately unsuitable. In version (d), an unnecessary constriction is induced, which could cause pulsations in the evaporator if the port velocity is high. Versions (b) or (e) are suggested instead. Version (f) has a large connection but a reduced pipe that will increase the flow velocity in the suction pipe. Versions (b) or (c) are recommended instead. Versions (a) and (e) may be unsuitable, depending on the system design. If calculations show that connection and pipe velocities are above 25 m/s, an increased pipe size will decrease energy losses. However, there is still a risk of maldistribution. Designing a larger CBE or two vapor exits will decrease the port velocity. In version (e), the pipe is also reduced, which will increase the velocity, and the fitting will induce an extra pressure drop.

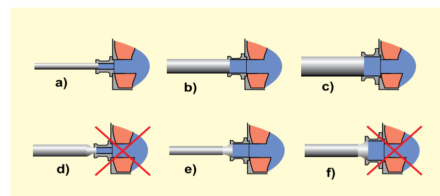


Figure 8.24 Outlet connection variations: (a) Smaller connection than port, (b) Connection size the same as port size, (c) Connection size larger than port, (d) Connection size smaller than port with larger pipe, (e) Connection size the same as port with smaller pipe, (f) Connection size larger than port with smaller pipe.

Conclusion: The inlet and outlet velocities affect the evaporation performance and the stability of the evaporator. Maldistribution, which will reduce performance and cause instability, is magnified by:

- Separated vapor and liquid flows in the inlet pipe, caused by low velocity and/or unsuitable piping design.
- High pressure-drop in the outlet port, caused by high vapor velocity.

By choosing the correct refrigeration side connections, the performance and stability of evaporation can be improved.

8.4 Freeze Protection

Freeze protection is a very important subject when designing a refrigerant system. The problem of freezing concerns only the evaporator, because it normally cools water with a refrigerant evaporating at low temperatures. It becomes a problem when the system is poorly designed and/or when external factors have a negative influence.

The protection against freezing can be provided on the secondary fluid side and/or the refrigerant side of the heat exchanger. The actual cause of freezing is a low evaporation temperature. If this temperature can be kept above the freezing temperature of the secondary fluid in all operating conditions, there will be no risk of freezing. In this section, situations where there is a potential risk of freezing are described, and methods to protect a system against freezing are discussed.

Situations with a Potential Freezing Risk

In some evaporator applications, there is a risk of freezing of the secondary fluid channels. A low evaporation temperature with low secondary fluid temperatures and a low total or local water flow rate increases the risk of freezing. In normal, steady operation, freezing is very seldom a problem. However, there are other situations (see below) where there is an increased risk of freezing.

Start-up at low ambient temperatures (systems with an air-cooled condenser)

At low ambient temperatures, the condensing pressure is low during the start-up of the compressor. This also results in a low evaporation temperature. Figure 8.25 shows how the suction temperature varies in such a system during the start-up.

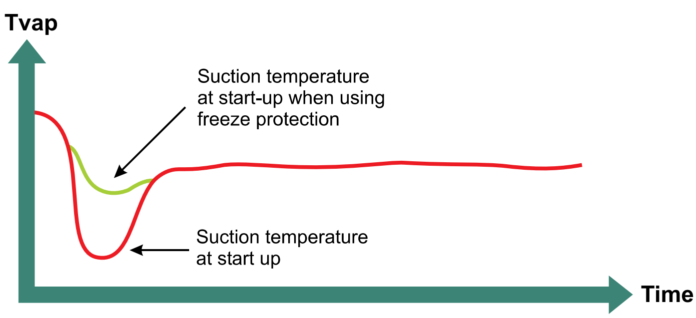


Figure 8.25 Suction temperature at start up, with and without protection against freezing.

The various methods for increasing the evaporation temperature during the start-up include:

- Hot gas by-pass valve
- Suction pressure control valve
- Condensing pressure control valve
- Compressors with variable capacity

More details are given below.

Reversed cycle defrosting (heat pumps)

For heat pumps with outdoor air-to-refrigerant evaporators, it is sometimes necessary to defrost the coil. This can be achieved by reversing the refrigerant flow. The evaporator is then run as a condenser for a short time, allowing the ice on the outside of the coil to melt. Because the coil is very cold when defrosting starts, the condensing pressure will be very low. The situation during defrosting is therefore the same as when starting up an air-cooled system at low ambient temperatures, i.e. the evaporation will be relatively low. During the defrosting cycle, the heat exchanger that normally operates as a condenser will be operating as an evaporator. Due to the low evaporation temperature, there may be a risk of freezing in this heat exchanger.

Pump down stop

In a traditional pump down stop of a system, the liquid line solenoid valve is first closed. The compressor continues to run, and the refrigerant is pumped from the evaporator to the high-pressure side of the system. The evaporation temperature will fall to relatively low temperatures. Eventually, when the evaporation temperature reaches a pre-set level, the low-pressure control will stop the compressor. Due to the low evaporation temperature during the pump down, there is a risk of freezing the evaporator. With CBE evaporators, the hold-up volume of the evaporator is so small that no pump down is necessary.

Simultaneous compressor and water pump stop at low suction temperatures

If the compressor is stopped at evaporation temperatures below 0°C, and the water pump is stopped at the same time for some reason, there is an obvious risk of freezing in the secondary fluid channels.

Outdoor units during off periods at ambient temperatures below 0°C

If the heat exchanger (evaporator or condenser) is installed outdoors, there is a risk of freezing when the ambient temperature is below 0°C.

Disrupted water flow rate

The water flow can be disrupted locally inside the evaporator, or the total flow can be disturbed. A local disturbance can be caused by the blocking of a channel, or part of a channel, by fouling or particles. The flow is then reduced or stopped, and the risk of freezing increases dramatically. The total flow can be disrupted by a closed valve, pump failure or initial cavitation when starting the pump, which also increases the risk of freezing.

Protection Methods Against Freezing on the Secondary Fluid (Water) Side

Freezing protection on the secondary fluid (water) side uses indirect methods to avoid or minimize situations with a risk of freezing. Figure 8.26 shows a refrigerant system that can be protected from freezing by controlling the water side of the system. Some of the protection methods are described below.

- **Flow switch in the water circuit of each evaporator**. The flow switch stops the compressor if the water flow falls below a certain level. This level should be as high as possible to secure a maximal water flow rate.

  As an alternative to a flow switch, a differential pressure sensor in the water circuit can be used. When there is a water flow, the sensor detects a pressure difference between the evaporator inlet and outlet. If there is no flow, there will be no pressure difference and the com-

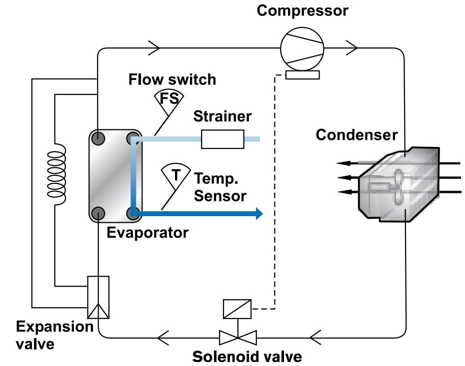


Figure 8.26 Refrigerant systems can be prevented from freezing by controlling the water side.

pressor will be stopped. A second alternative to a traditional flow switch is a pressure sensor mounted in the water inlet pipe. When the water pump is running, there is increased water pressure, which is detected by the sensor. If the pump (and thus the flow) is shut off, the pressure will decrease, and the sensor will tell the compressor to stop.

- **Strainer to prevent locally low flow inside the evaporator**. The flow switch protects only against a low total water flow. A strainer is required to prevent particles from entering the evaporator and disturbing the flow locally inside the heat exchanger. The strainer should stop particles larger than 1 mm. This corresponds to a mesh size of 16-20 mesh, depending on the wire diameter.
- **Temperature sensor in the water flow leaving the evaporator**. The temperature sensor protects against low water temperatures. When the leaving water temperature falls below a certain level, the compressor is stopped. This level should be as high as possible without disturbing the operation of the system.
- **Delayed water pump stop when stopping the compressor**. The pump can be allowed to run for some minutes after the compressor is stopped. This gives the evaporation temperature time to increase to a level high enough to avoid freezing.
- **Electric heater and insulation applied to the outside of the heat exchanger**. An electric heater and insulation can prevent the freezing of outdoor units during off periods.
- **Avoid reversed cycle defrosting when the condenser entering water temperature is low.** To minimize the risk for freezing, defrosting should not be allowed when the temperature of the secondary fluid entering the condenser is below 20°C.
- **Start the water pump before starting the compressor**. The pump should be started before the compressor to allow the water flow to stabilize. This avoids possible disturbance of the water flow due to initial pump cavitation.

Protection Methods Against Freezing on the Refrigerant Side

Protection on the refrigerant side aims to keep the evaporation temperature above the freezing temperature of the secondary fluid. Some methods of freezing protection on the refrigerant side are described below.

- **Low-pressure (LP) control.** LP control stops the compressor when the evaporation temperature falls below a certain level. This could therefore provide ideal freeze protection. However, it should be noted that for practical reasons, this device is often bypassed and not working during the most critical situation, i.e. during the start-up. In practice therefore, LP control works only during regular operation when the risk of freezing is in any case low.
- **Hot gas bypass valve** (see Figure 8.27). When the evaporation temperature falls below a set level, this valve leads gas from the high-

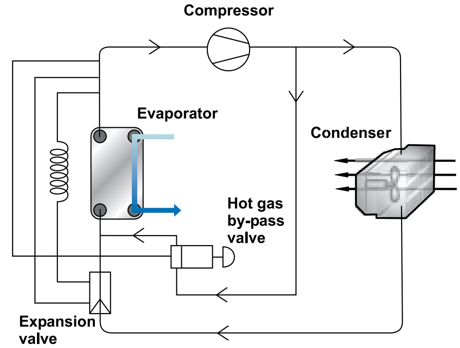


Figure 8.27 Refrigerant system with freezing control using a hot gas bypass valve on the refrigerant side.

pressure side of the system to the evaporator inlet. This will prevent the evaporation temperature from decreasing, and thereby protect from freezing. This function can be realized using either a modulating valve (continuous regulating) or a controller combined with a solenoid valve.

- **Suction pressure control valve** (see Figure 8.28). This valve is positioned in the suction line. When the evaporation temperature falls below the set level, the valve starts to throttle. The evaporation temperature is thereby prevented from decreasing any further. This valve provides good freeze protection. The disadvantage is that there is also a pressure drop through the valve when it is fully opened. The pressure drop will have a negative impact on the system efficiency (COP).
- **Condensing pressure control valve** (see Figure 8.29). This valve is positioned in the liquid line between the air-cooled condenser and the receiver. During the winter, when the low ambient temperature results in a low condensing pressure, the valve starts to throttle. Liquid is then backed up into the condenser, which decreases the heat transfer surface available for condensing. The condensing pressure hence increases, and the fall in evaporation temperature will be prevented.
- **Temperature sensor in the suction line or refrigerant liquid inlet line.** Temperature sensors in the suction line or refrigerant liquid line can be used to control the temperature of the refrigerant at the evaporator outlet or inlet. When the temperature falls below a certain level, the compressor is stopped. When measuring in the suction line, the superheating of the gas should be considered. When measuring in the liquid inlet line, the pressure drop through the refrigerant side of the evaporator should be considered. The temperature at the inlet will differ from the actual evaporation temperature inside the evaporator, particularly if the evaporator is equipped with a distributor.
- **Avoiding low condensing pressure at start-up.** Delayed condenser fan start can be used to minimize the problem of a low condensing pressure. The compressor is allowed to start before the fan is started. The condensing pressure increases more quickly, and the fall in evaporation temperature will not be as great. There are indications that a delayed condenser fan start could increase the evaporation temperature by approximately 5°C. When the pressure increases, the fan is started. If possible, the fan speed should be increased gradually to avoid a steep fall in condensing pressure.
- **Variable compressor capacity.** If a compressor with variable capacity is installed, it can be used to avoid low evaporation temperatures. The compressor is started at as low a capacity as possible. This will minimize the fall in evaporation temperature during the start-up.
- **No pump down.** In systems using CBE evaporators, the refrigerant volume is very small and there is no need to use a pump down system.

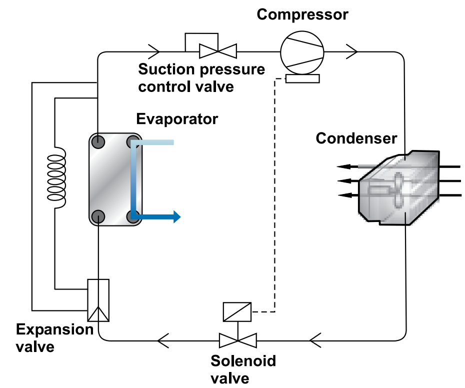


Figure 8.28 Refrigerant system with freezing control using a suction pressure control valve on the refrigerant side.

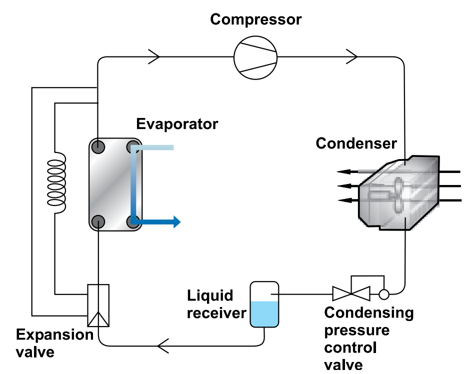


Figure 8.29 Refrigerant system with freezing control using a condensing compressor control valve on the refrigerant side.

Instead, the liquid line solenoid valve is closed at the same time as the compressor switches off. Switching the system off like this avoids the low evaporation temperature occurring during a traditional pump down.

8.5 Corrosion

Corrosion is the term for a chemical or electrochemical reaction between a material, usually a metal, and its environment, which produces a deterioration of the material and its properties, e.g. rust.

Types of Corrosion

A metal can be exposed to many different kinds of corrosion. The types of corrosion considered below are those most common when compact brazed heat exchangers are exposed to corrosive environments.

Pitting and Crevice Corrosion

In principle, pitting and crevice corrosion are the same phenomenon. Pitting may appear on exposed surfaces, for example attacking stainless steel if the passive layer is damaged. The passive layer is a protective surface film that is formed spontaneously when stainless steel 316 is exposed to air. The attack can be sudden, and may quickly cause leakage. Crevices may occur in welds that fail to penetrate, in flange joints and under deposits on the steel surface.

General Corrosion

General corrosion is a deterioration distributed more or less uniformly over a surface. This type of corrosion is more predictable than pitting; if a device has corroded 0.1 mm in one year, it will very probably corrode 0.2 mm in two years.

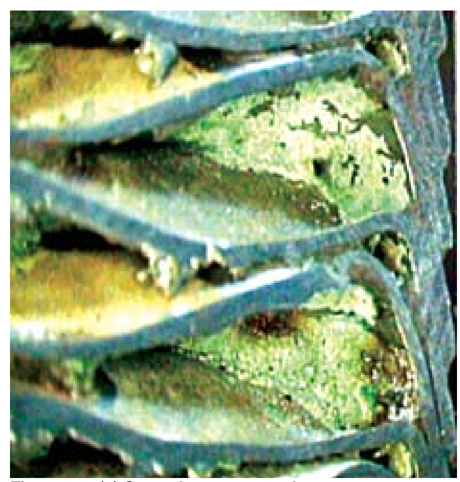


Figure 8.30 (a) General copper corrosion.

Corrosion in CBEs

When the CBE is exposed to the recommended environment, there should be no corrosion problems. Neither stainless steel 316 nor copper corrodes easily. However, if CBEs from the standard range are exposed to an unfavorable environment, corrosion can attack either the stainless steel 316 or the copper brazing, as shown in Figure 8.30.

CBEs do not resist high concentrations of chloride ions (Cl^-) in an oxidizing environment, because chlorides form a galvanic cell with oxygen and the metals of the CBE. Stainless steel, in particular, is sensitive to this kind of attack, and the result may be pitting and/or crevice corrosion. It is also important to mention that higher temperatures make chlorides more aggressive towards stainless steel. When chlorides or other halogen ions (bromides, iodides) are present in high concentrations, SWEP recommends a CBE with channel plates made of molybdenum steel (SMO 254) or a Minex with channel plates in titanium.

Figure 8.30 (b) Pitting corrosion in stainless steel 316.

When copper corrodes, it is more often degraded by general corrosion than by pitting. General corrosion will most probably attack copper exposed to ammonia (NH_3) or fluids with high sulfur contents. SWEP's all-stainless CBEs have a nickel alloy as the brazing material (instead of copper), which is resistant to high sulfur and ammonia contents. Another threat to copper is the presence of dissolved salts in the fluid that affect the CBE. Maintaining electrical conductivity within the recommended range will minimize this source of corrosion. General corrosion may attack both copper and stainless steel in strongly acidic solutions. However, copper is the more sensitive metal in an alkaline environment.

Water Quality

The corrosive effect of natural water can vary considerably with its chemical composition. Water quality is of great importance in avoiding corrosion in CBEs.

City water is normally of good quality, and is used as make-up water in cooling towers.

Well water is usually fairly cold and clean, which implies that it has a low biological content (see chapter 8.6). However, the concentration of scale-forming salts (calcium and magnesium sulfates and carbonates) can sometimes be very high. Pitting corrosion may be initiated under these salt deposits.

Cooling tower water (see Figure 8.31) is circulated in an open circuit between the CBE unit and the cooling tower. The salt content can be ten times higher than in the make-up water, which is usually city water (very clean) or well water (fairly clean). In heavily polluted areas, the water may pick up dust and/or corrosive gases, such as sulfur and nitrogen compounds, during its circulation. The net effect could be a corrosive brew that requires treatment on a regular basis. As it is an open loop, treatment is fortunately quite easy to carry out.

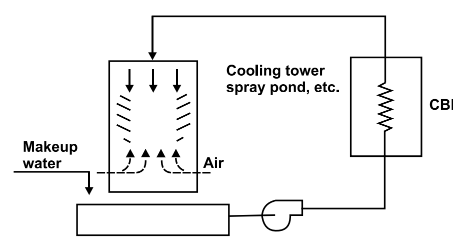


Figure 8.31 Cooling tower water.

In **river and lake surface water,** the concentration of scale-forming salts is usually fairly low. However, there may be appreciable amounts of solids ranging from salts and soil particles to leaves and algae. Some type of pre-treatment is usually necessary, particularly to control biological activity.

Brackish water and seawater are not recommended in standard CBEs (stainless steel 316/copper), all-stainless CBEs (stainless steel 316/nickel) or Mo-steel CBEs (SMO 254/copper) because of the corrosive action of the very high chloride concentrations. However, Minex is available in titanium, which is compatible with seawater. Because titanium forms a very stable, continuous and protective oxide film, it has excellent resistance to corrosion. A damaged oxide film can generally heal itself immediately, provided traces of oxygen or water are present in the environment.

Avoiding Corrosion

In an environment containing halogen ions (e.g. chlorides or bromides) a Mo-steel CBE in SMO 254 or a Minex with titanium channel plates is recommended. Stable and turbulent water flow does not give corrosive substances the time needed to start the corrosion process. It is therefore important to maintain a stable water flow to avoid stagnant zones inside the CBE. For the same reason as mentioned above, it is worth rinsing and drying the CBE carefully before a long standby.

Bleed-offs and make-up water input to cooling towers must be carried out regularly, because the circulating water in an open cooling tower will be more or less contaminated with corrosive substances. When using cooling tower water, a strainer should always be installed at the inlet of the CBE. The recommended strainer stops particles larger than 1 mm, which correspond to a mesh size of 16-20 mesh (depending on the wire diameter). Smaller particles will pass through the heat exchanger due to the high turbulence.

Alongside the advice above, salts, chlorides, pH and temperature, etc., must be kept within the recommended ranges to avoid CBE corrosion. For more specific information see Appendix A.

8.6 Fouling

Fouling is a very undesirable phenomenon in the world of heat transfer and heat exchangers. In most heat exchangers, the fluid flowing is not completely free from dirt, oil, grease and chemical or organic deposits. In all cases, an unwanted coating can collect on the heat transfer surface, decreasing the heat transfer coefficient. The thermal efficiency of the heat exchanger will be reduced and the pressure drop characteristics may change.

This section discusses several types of fouling, the reasons for their occurrence and the preventive measures that can be taken avoid them. The different types of fouling discussed are:

- Scaling
- Particulate fouling
- Biological growths
- Corrosion

Scaling

Scaling is a type of fouling caused by inorganic salts in the water circuit of the heat exchanger. It increases the pressure drop and insulates the heat transfer surface, thus preventing efficient heat transfer. It occurs at high temperatures, or when there is low fluid velocity (laminar flow) and uneven distribution of the liquid along the passages and the heat transfer surface.

The likelihood of scaling increases with increased temperature, concentration and pH. Studies have shown that high turbulence and a small hydraulic diameter, such as with SWEP CBEs, have beneficial effects on this type of fouling. Proper maintenance and treatment of the cooling water, e.g. pH treatment, greatly reduce the risk of scaling, especially in cooling towers.

Most scaling is due to either calcium carbonate (lime) or calcium sulfate (gypsum) precipitation. These salts have inverted solubility curves (see

Figure 8.32), i.e. the solubility in water decreases with increasing temperature. The salts are therefore deposited on the warm surface when the cold water makes contact with it. Pure calcium sulfate is very difficult to dissolve, which makes cleaning more complicated. In general, other types of scale are more easily removed.

Calcium sulfate:	$Ca^{2}+[aq] + SO_4^{2-}[aq] \Leftrightarrow CaSO_4[s]$
Calcium carbonate:	$Ca^{2}+[aq] + CO_3^{2-}[aq] \Leftrightarrow CaCO_3[s]$
[aq] = in solution	
[s] = as solid material	

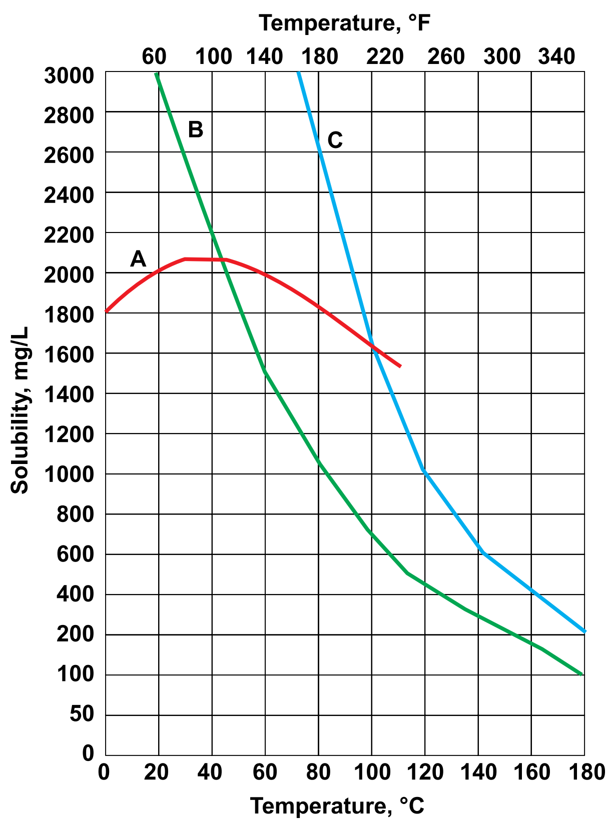


Figure 8.32. Solubility of three forms of calcium sulfate ($CaSO_4$) versus water temperature: A) $CaSO_4 \cdot 2\,H_2O$, B) Anhydrous $CaSO_4$, C) Hemihydrate $CaSO_4 \cdot H_2O$.

The most important factors that influence scaling are:

- Temperature
- Turbulence
- Velocity
- Flow distribution
- Surface finish
- Composition and concentration of the salts in the water
- Water hardness
- pH

Scaling is more likely at a high pH, so a general approach to this problem is to keep the pH between 7 and 9. The risk of scaling generally increases with increasing water temperature. Experience shows that scale is seldom found where wall temperatures are below 65°C. This implies that the temperatures are usually not high enough to lead to scaling in refrigerant condensers. This is illustrated in the two examples below.

Example 1 – Condenser

A typical R22 condenser works at T_{in}=85°C and the condensing temperature is T_{cond}=40°C on the refrigerant side. The water inlet temperature is 29°C and the leaving water temperature (LWT) is 36°C.

As shown in the temperature program in Figure 8.33, the maximum leaving water temperature (LWT) is only slightly above the condensing temperature (T_{cond}). This is due to the water temperature at the pinch point (circled in Figure 8.33) always being lower than the refrigerant temperature (T_{cond}) at the pinch point. No crossing of temperatures is possible, because the temperature difference is the driving force of the heat transfer (see chapter 1.1). Furthermore, the amount of heat transferred from the refrigerant gas to the water is relatively small. The discussion above shows that the temperatures in the bulk are not high enough to lead to scaling.

Figure 8.34 shows an example of the temperatures inside a condenser. Although the bulk gas temperature is as high as 85°C on the refrigerant side, the wall temperatures are determined by the bulk water temperature (36°C). This is because the film coefficient is much higher on the water side than on the refrigerant side. The maximum wall temperature is therefore 38°C on the water side and 38.6°C on the gas side, still below temperatures where scaling is a problem.

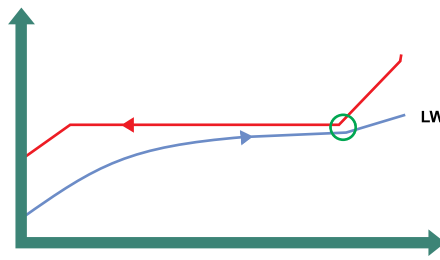


Figure 8.33 A temperature program for example 1.

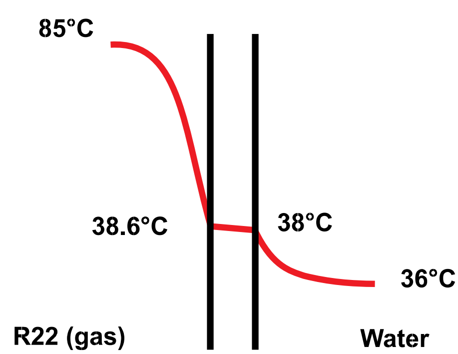


Figure 8.34 An example of the temperatures inside a condenser.

Example 2 – Desuperheater/Heat Recovery

A typical R22 desuperheater works at T_{in}=85°C and T_{out} =45°C on the refrigerant side. No condensing occurs. The water inlet temperature is 10°C and the leaving water temperature is 50°C.

As can be seen in Figure 8.35, in this case there is no temperature pinch to prevent the leaving water temperature from rising. Nevertheless, there is no risk of scaling in this case because the LWT is designed to be only 50°C.

Figure 8.36 shows that although the gas temperature is high at the inlet (85°C), the maximum wall temperature will not be higher than 51°C, i.e. normally no risk of scaling. The reason for the relatively low wall temperature is the low heat transfer coefficient (α_{gas}) on the gas side compared with the water side (α_{water}).

However, it is important in desuperheating/heat recovery applications to have a high constant flow on the water side. If the water flow is reduced or turned off, the temperature will rise and there will be a risk of scaling.

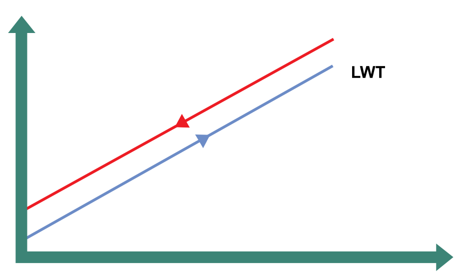


Figure 8.35 Temperature profile in a desuperheater.

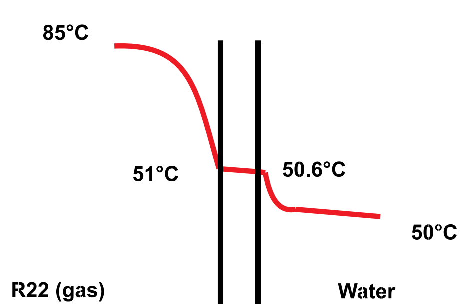


Figure 8.36 An example of the temperatures inside a desuperheater.

Types of scale

Calcium carbonate ($CaCO_3$) can be formed when calcium or bicarbonate alkalinity (HCO_3^-, CO_3^{2-} and OH^- ions in the water) is present. An increase in heat and/or an increase in pH will cause precipitation of calcium carbonate according to the equation:

$$Ca(HCO_3)_2 \rightarrow CaCO_3\,[s] + CO_2\,[g] + H_2O$$

Calcium sulfate ($CaSO_4$) is 50 times more soluble than calcium carbonate and will therefore precipitate only after calcium carbonate has been formed. This type of scale can exist in various forms, and its formation depends strongly on the temperature. An increase in temperature decreases the solubility of this salt and increases the risk of scaling.

Water Scaling Tendency

In order to estimate the scaling tendency of natural water, several parameters must be analyzed and determined:

- pH
- Calcium content
- Alkalinity
- Ionic strength of the water

The first three parameters are relatively straightforward to determine. However, the ionic strength depends on the total amount of dissolved, dissociated compounds, i.e. salts and acids, as well as the relative concentrations of the various salts and acids.

The Langlier saturation index, I_s, is calculated from the amount of total dissolved solids (TDS), calcium concentration, total alkalinity, pH and solution temperature. It shows the tendency of a water solution to precipitate or dissolve calcium carbonate. In this method, the pH_s (the pH at the equilibrium state) is calculated from the total salt content (pS), the alkalinity (pAlk) and the calcium content (pCa). The pH_s is then compared with the actual pH for the water, giving the Langlier Index, I_s:

$$\mathbf{I_s = pH - pH_s}$$

where

$$\mathbf{pH_s = pS + pAlk + pCa}$$

The **pH** measurement is straightforward and is performed routinely. Because the pH may vary with the season and the climatic conditions, it should be measured on several different occasions. The calcium content**, pCa,** is normally expressed as the concentration of calcium either as calcium carbonate ($CaCO_3$) or as calcium ion (Ca^{2+}). The bicarbonate **alkalinity, pAlk,** can be determined by titrating the water with an acid and a suitable indicator (e.g. methyl orange). The result is expressed in various ways, e.g. as the equivalent $CaCO_3$. The corresponding pAlk is obtained from the Langlier diagram (see Figure 8.37). The relative proportions of the various salts are fairly constant in naturally occurring water. Langlier uses the total salt content (mg/l), i.e. TDS (Total Dissolved Solids), as a measurement of the ionic strength. The corresponding amount of total solids**, pS** is obtained from the Langlier diagram (see Figure 8.37). All these measurements can be obtained from a general water analysis.

Please note:

- For water analysis, mg/l is equal to ppm
- The relationship between calcium and calcium carbonate is:

$$40\text{g } Ca^{2+} \Leftrightarrow 100\text{g } CaCO_3$$

- TDS = Salt content (mg/l), or possibly the conductivity x 0.63 (µS/cm).

If I_s is negative, the water has a tendency to be corrosive. This corrosivity is valid for carbon steel and, to a lesser extent, copper, but not for 316 stainless steel. If I_s is positive, the water has a tendency to cause scaling.

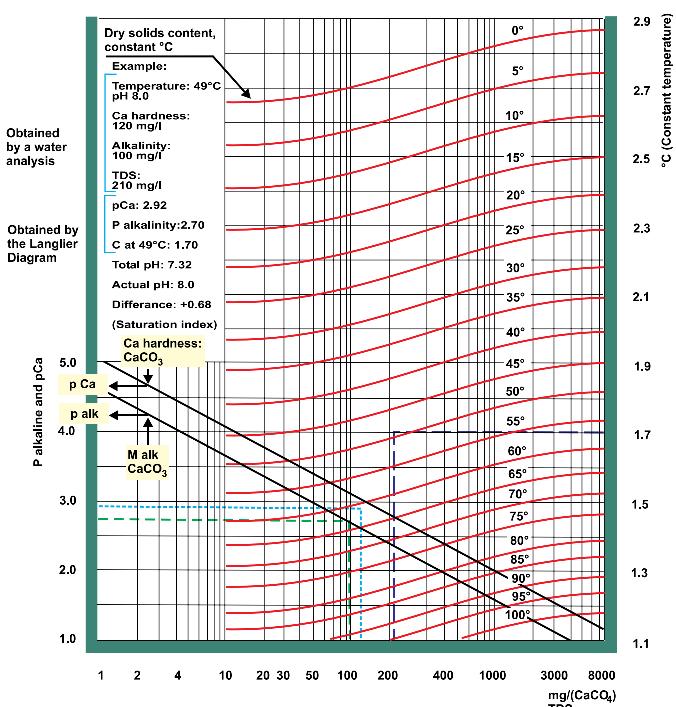


Figure 8.37 Langlier diagram.

Example of the Use of the Langlier Diagram

When analyzing the water sample, the following values were obtained:

Temperature	49°C
pH	8.0
Ca	120 mg/l $CaCO_3$
Alkalinity	100 mg/l $CaCO_3$
TDS	210 mg/l

Using the results from the water analysis above, the pCa, pAlk and pS can be found in the Langlier diagram as follows:

pCa

On the Langlier diagram (Figures 8.37 and 8.38), note the measured Ca concentration of 120 mg/I $CaCO_3$ (or 120 ppm). Read off the pCa value at the point where the calcium concentration meets the diagonal line for Ca/pCa. This gives **pCa=2.92**.

pAlk

On the Langlier diagram (Figures 8.37 and 8.38), note the measured alkalinity value of 100 mg/l $CaCO_3$ (or 100 ppm). Read off the pAlk value at the point where the alkalinity value meets the diagonal line for $CaCO_3$/pAlk. This gives **pAlk=2.70**.

pS

On the Langlier diagram (Figures 8.37 and 8.39), note the measured TDS concentration of 210 mg/I. Read off the pS value at the point where the TDS concentration meets the temperature line (in this case 49°C). This gives **pS=1.70**.

Using the results extracted from the Langlier diagram, pH_s can be calculated, and then the saturation index I_s:

$$pH_s = pS + pAlc + pCa = 1.70 + 2.70 + 2.92 = 7.32$$
$$I_s = pH - pH_s = 8.0 - 7.32 = (+)0.68$$

Because $I_s > 0$, the water in this example has a tendency to cause scaling.

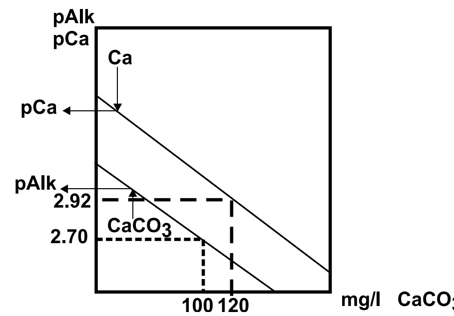


Figure 8.38 How to obtain the pCa and the pAlk from the Langlier Diagram in Figure 8.37.

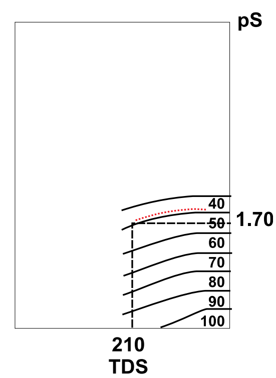


Figure 8.39 How to obtain the pS from the Langlier Diagram in Figure 8.37.

Determine Whether Scaling Has Occurred

To be able to clean the heat exchanger unit easily, it is important to note the signs of scaling before the unit is completely clogged. This can be done by measuring the inlet and outlet temperatures of the heat exchanger, which indicate whether fouling has occurred. Fouling of the heat transfer surface decreases the heat transfer, resulting in a temperature difference smaller than specified. Another way to detect fouling is by measuring the pressure drop over the heat exchanger. Because fouling restricts the passages, and thus increases the velocity, the pressure drop will increase. When using this method, make sure that the water flow rate is as specified, because changes in the flow rate will of course also affect the temperature change and the pressure drop.

Prevention of Scaling

The formation of *calcium carbonate* can be controlled by adding acids or specific chemicals (phosphate compounds, e.g. **AMP**, or organic polymers, e.g. polyacrylates) tailored to inhibit the precipitation of the compound. However, water treatment is not an easy task, and a water specialist should be consulted in order to determine the correct treatment. Improper use of acids can cause severe corrosion of the CBE in a very short time.

Calcium sulfate scaling can be controlled most effectively with chemicals such as polyacrylates or AMP.

Removing the scale that has been formed restores the operating efficiencies of the equipment and the heat transfer surfaces. Other benefits from removing the scale are that it lowers the pressure drops, reduces the power consumption and extends the lifetime of the equipment.

Particulate fouling

Particulate fouling is caused by suspended solids (foulants) such as mud, silt, sand or other particles in the heat transfer medium. Important factors that affect particulate fouling are:

- velocity
- distribution of the flow
- roughness of the heat transfer surface
- size of the particles

Velocity

The velocity is an important factor in the sense that it controls whether the flow is turbulent or laminar. Turbulent flow is desirable for several reasons. Turbulent flow will keep particles in the fluid in suspension, i.e. no particles are allowed to collect on the surface, which will avoid surface fouling. Another very important reason, of course, is that turbulent flow improves the heat transfer.

CBEs have a high degree of turbulence, and the fluid has a scouring action that keeps the heat transfer surface clean. This is due to the unique design of CBEs. As the fluid passes through the channels, it constantly changes its direction and velocity. This ensures turbulent flow even at very low flow rates and pressure drops.

For shell and tube (S&T) heat exchangers, a much higher velocity is required to reach turbulent flow.

In an S&T, the water can flow either inside the tubes or outside the tubes. When the water passes through a tube, the maximum velocity is at the center of the tube. The turbulence at the walls is too low to keep particles in the fluid in suspension. These particles are allowed to precipitate and collect on the tube wall, which causes fouling of the heat transfer surface. When the water flows outside the tubes, the flow rate is lower and low-flow areas are created, which increases the risk of fouling. This means that S&T heat exchangers are much more sensitive to fouling than plate heat exchangers. When designing S&T heat exchangers, the use of so-called **fouling factors** is recommended to account for the risk of fouling and the consequent decrease in performance.

Distribution of the Flow

It is of great importance for the flow over the heat transfer surface to be well distributed to maintain uniform velocity. The flow distribution depends very much on the plate pattern. A special distribution pattern in the port areas of SWEP CBEs ensures a well-distributed flow. In other heat exchangers (S&T, coaxial and other brazed heat exchanger brands), there can be areas of low velocity (resulting in laminar flow) due to uneven distribution of the fluid through the exchanger. These sections are sensitive to fouling. The fouling starts at the low velocity areas and spreads over the heat transfer surface.

Roughness of the Heat Transfer Surface

Rough surfaces are known to encourage fouling by collecting particulate matter. The material used in every SWEP CBE is AISI-316 stainless steel, and the smooth surface of this material minimizes fouling. The round shape of the brazing points ensures that no pockets of stagnant water can be formed.

In applications where a cooling tower or other open system is used, the cooling water will be rich in oxygen. This can cause the corrosion of materials such as the carbon steel used in conventional heat exchangers. This corrosion is usually in the form of iron oxide scale on the carbon steel surface, but loose iron oxide can be deposited elsewhere as well. The stainless steel used in the SWEP CBE is not subject to the uniform corrosion that causes fouling problems. However, SWEP CBEs are not completely immune to corrosion under certain conditions.

The Size of the Particles

Particulate fouling can influence the performance of the heat exchanger in two ways, depending on the particle size. First, if the particles are large (>1 mm) or have a fibrous structure, they may lodge in the inlet of the heat exchanger or clog the channels. The result is an increased pressure drop in the water circuit of the heat exchanger. Clogged channels also mean low water velocities, which can result in freezing when using the CBE as an evaporator. Second, particles may adhere to the heat transfer surface and build up a layer of low thermal conductivity material. Initially, this leads to reduced heat transfer, and a higher pressure drop in severe cases of fouling.

Prevention of Particulate Fouling

Clean Cooling Water

The best way to avoid particulate fouling is to *keep the cooling water clean* and thereby prevent particles from entering the heat exchanger. However, in all cooling systems, and especially when using open cooling systems (with cooling towers), there will always be particles present in the cooling water. The correct maintenance of cooling towers will dramatically reduce the risk of fouling, including particulate fouling, scaling and corrosion. The evaporation of water in cooling towers is unavoidable, and they must be re-filled with make-up water. However, it is very important to bleed (discharge) water from the tower, otherwise impurities will accumulate and soon reach dangerous concentrations. This bleed water is called blowdown.

Strainer

A *strainer* is recommended before the inlet of the heat exchanger. A strainer will prevent large particles (>1 mm) from entering the heat exchanger. The recommended size of strainer for this purpose is 16-20 mesh or a mesh size of 0.5 to 1 mm. If a smaller mesh size is used, this will of course result in better filtration of the water, but the system will also need more frequent cleaning. It also creates an unwanted pressure drop.

Side-Stream Filtration

When the make-up water for the cooling tower contains significant suspended matter, it is advantageous to use side-stream filtration. A filtration unit (several types are available) is connected to the cooling tower basin. Water from the basin is then pumped through the filtration unit and back again. In general, passing a few percent of the re-circulating water through the side-stream filter will reduce the suspended solids by 80-90%.

Adequate Flow Rates

High flow rates will keep particles in suspension and prevent them from depositing on the heat transfer surface.

Chemical Water Treatment

Chemical treatment of water can also be an effective method of controlling suspended particulates. As in the prevention of scaling, polyacrylates disperse foulants (suspended solids) very efficiently. Concentrations of a few milligrams per liter are required in open re-circulating systems.

Biological Growths

Fouling through biological growths (also called biofouling) occurs when living matter grows on the heat transfer surface. In many cases, re-circulating cooling systems are ideal for promoting the life of microorganisms. Three types of living organisms are considered here: algae, fungi and bacteria.

Algae are easily detected by their green color. They need oxygen and sunlight to grow, and they can therefore exist in cooling towers. In addition to reducing the thermal performance, algae can also have a severe impact on metal corrosion by providing conditions that increase the risk of corrosion.

Fungi are similar to algae but do not require sunlight. They require moisture and air and exist on nutrients found in water or on the material they are attached to, for example bacteria, algae or wood.

Bacteria can live with or without oxygen. Water and other wet environments with organic content are suitable for the growth of bacteria. Heat exchangers can therefore provide an excellent environment for this type of micro-organism, which will reduce the heat transfer. Bacteria can also initiate pitting corrosion.

Prevention of Biological Growths

Biocides are the most practical and efficient method of controlling the growth of micro-organisms in cooling water systems. Biocides kill or inhibit the growth of the organisms. Although these chemicals inhibit biofouling, they will not remove material already adhering to surfaces. This emphasizes the importance of a clean system from the start. A number of methods and chemicals are available.

Chlorination

Chlorine (Cl_2) is an excellent bactericide and algaecide. Chlorination can be either continuous or of the shock type. When adding chlorine continuously, levels of 0.1 to 0.2 mg/l (ppm) are recommended. For the best biocide effect, the pH should be between 6.5 and 7.5. A lower pH will accelerate corrosion, while a higher pH will have less impact on bio-organisms. The shock chlorination method uses a chlorine concentration approximately 10 times higher, but only for a few brief periods every day. The advantage of this method is lower chlorine consumption.

Because chlorine decomposes into chloride (Cl^-) ions, there is a risk of pitting corrosion of the 316 stainless steel used in CBEs. For this reason, it is very important to bleed off cooling water from the cooling tower to avoid the accumulation of chloride ions and dangerous chloride concentrations. Due to the risk of corrosion, chlorine should always be added as far from the heat exchanger as possible.

Fouling Due to Corrosion

In some cases, fouling can be due to corrosion. The added layer of corrosion products on the heat transfer surface will reduce the heat transfer efficiency. The degree of corrosion depends very much on the water quality.

Prevention of corrosion

The main fouling risk is due to corrosion products from other parts of the system. These particles will be carried by the water and may adhere to the heat transfer surface or lodge inside the heat exchanger. This type of fouling should be considered as particulate fouling, and prevented as for particulate fouling.

Fouling Resistance due to SWEP's CBE Design

SWEP's CBE heat exchangers provide good resistance against surface fouling for several reasons. The unique design of a SWEP CBE allows the heat exchanger to operate at extremely low velocities while maintaining a turbulent flow. As the fluid passes through the channel, its direction constantly changes, which disturbs the boundary layer and ensures turbulent flow even at extremely low velocities. A SWEP CBE is actually much less prone to fouling than other heat exchangers. This is because of its internal geometry (which ensures evenly distributed fluid), the higher turbulence and the hardness and smoothness of the stainless steel in the channel plates. With laminar flow, the velocity of the fluid close to the plate surface is very low, which means that the suspended particles are allowed to settle (cf. Figure 8.40).

Figure 8.40 Laminar (left) and turbulent flow (right) through a channel.

- The smooth surface of the channel plate material has a positive effect on minimizing fouling. Rough surfaces are known to encourage fouling, because they collect particulate matter by giving it a chance to adhere to the surface.
- The design of SWEP CBEs ensures that no dead zones (where fouling compounds can settle) are created. In a dead zone, the liquid is stagnant and the suspended material has the chance to settle and accumulate.
- The particles are kept in suspension by the very high turbulence, even at low flow rates, caused by the corrugations in the plates. Turbulent flow and a small hydraulic diameter, such as with SWEP CBEs, are important to prevent the suspended particles from settling. With laminar flow, the particles have a much higher tendency to settle.

Optimization of Factors that Affect Surfaces under Various Conditions

Some factors that affect the surfaces of a heat exchanger are discussed below:

- Use the highest possible water pressure drop. A high pressure-drop implies higher shear stresses, and large shear stresses are always beneficial if there is any scale. The shear stresses work as descalers by constantly imposing forces on the adhered material that pull the particulate material away from the surface. The shear stresses also help keep the particles in suspension (see Figure 8.41).
- For a heat exchanger with a temperature above 70°C on the hot side and/or very hard water (and hence a danger of scaling), the pressure drop should be increased as much as possible on the cold water side and reduced on the hot side. This reduces the wall temperature on the cooling water side and increases the shear stresses, thus making it more difficult for the scaling compounds to adhere.
- Consider the use of co-current instead of counter-current flow. The warmest part of the hot side, the inlet, will then face the coldest part of the cold side. This normally decreases the maximum wall temperature on the cooling water side, which automatically limits the outlet water temperature.
- The normal practice is to let the cold water enter the lower port. This arrangement should be used whenever possible, because if the cold water enters through the upper port, it could encourage debris to enter the channels.

Figure 8.41 Shear stress keeps the particles in suspension.

8.7 Cleaning in Place

Provided the heat exchanger is not completely clogged, it is possible to clean the exchanger by circulating a cleaning liquid (Cleaning in Place, CIP). Heat exchangers should therefore be cleaned at regular intervals.

If the installation operates under difficult conditions, for example with hard water, installation of a heat exchanger with extra connections on the back for CIP piping is recommended to facilitate maintenance (see Figure 8.42). This makes it possible to connect and circulate the CIP solution through the system without having to disassemble the ordinary installation.

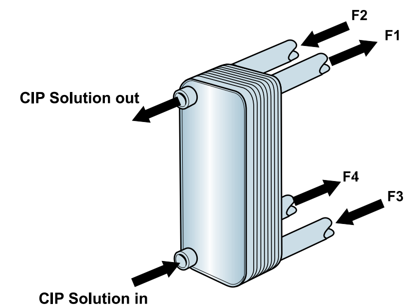


Figure 8.42 CBE with extra CIP connections.

The choice of cleaning solution depends on the problem, but a weak acid is a good start. This could be 5% phosphoric acid or, if the exchanger is cleaned frequently, 5% oxalic acid. The cleaning liquid should be pumped through the exchanger. For optimal cleaning, the flow rate of the cleaning solution should be at least 1.5 times the normal flow rate. Preferably, the flow should be in a back flush mode, which has a better chance of dissolving the scale because it attacks the deposits from the opposite direction.

After cleaning, the heat exchanger should be rinsed carefully with clean water. A solution of 1-2% sodium hydroxide (NaOH) or sodium bicarbonate ($NaHCO_3$) before the last rinse ensures that all acid is neutralized. One way to get an indication of the appropriate rinse time is to test the pH of the liquid at the outlet from the heat exchanger. A quick and easy method is to use litmus paper. The pH should be 6-9.

Circulation Systems

The circulation system could be a vertical peristaltic pump. In this type of pump, the liquid is forced forwards by an eccentrically rotating wheel connected to an engine (see Figure 8.43).

Important features of a CIP pump:

- The reservoir for the CIP solution should be manufactured in acid- and alkali-resistant material.
- Hoses should be made in PVC.
- It is an advantage if the pump is provided with a reverse flow device. Using a model with a reverse flow device makes it possible to attack scale from both directions.
- It is an advantage if the pump is equipped with a heating device. Heating the CIP solution usually increases the cleaning effect.
- The required flow rate capacity depends on the size of the heat exchanger.

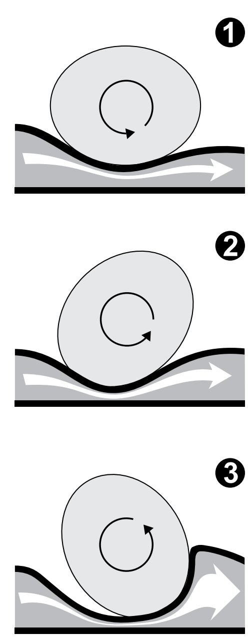


Figure 8.43 A peristaltic pump works by forcing the liquid forwards with an eccentric rotating wheel.

Eliminating the Scaling Problem: Principles

There are several ways to eliminate the scaling problem. Usually, a commercial product containing additives to enhance the effect and/or prevent corrosion can be employed. Do not use any product containing ammonia if the filler material of the CBE is copper. Take great care when using strong inorganic acids such as hydrochloric, nitric or sulfuric acids, because they are extremely hazardous. Under certain conditions, hydrochloric acid can corrode stainless steel in minutes, and nitric acid corrodes copper.

Chemical cleaning is the use of chemicals to dissolve or loosen deposits from process equipment and piping. Removal is uniform and generally at a lower overall cost. In principle, there are two steps in this process. The last step can sometimes be excluded.

Step 1: Chemical Cleaning Solutions

Mineral acids such as hydrochloric acid (HCl), sulfamic acid (NH_2SO_3H), nitric acid (HNO_3), phosphoric acid (H_3PO_4) and sulfuric acid (H_2SO_4) have a good ability to dissolve scale. However, they can also corrode the stainless steel or copper if used improperly. Organic acids are much weaker than mineral acids, in terms both of their dissolving ability and their ability to corrode the base material of the CBE. This makes these acids more useful when attempting to remove scale from the CBE because they are potentially less dangerous. They are often used in combination with other chemicals to bind the scale into complexes. Another advantage of organic acids is that they can be disposed of by incineration. Organic acids include formic acid (HCOOH), acetic acid (CH_3COOH) and citric acid ($C_3H_4(OH)(COOH)_3$).

Phosphoric acid is sometimes used at 2% concentration and 50°C for 4-6 h to pickle and passivate steel piping. It is not as effective as HCl in removing iron oxide scale, but is preferred for cleaning stainless steels. Formic acid is generally used as a mixture with citric acid or HCl, because alone it is unable to remove iron oxide deposits. Formic acid can be used on stainless steel. It is relatively inexpensive and can be disposed of by incineration. Acetic acid is used to clean calcium carbonate scale, but it is ineffective in removing iron oxide deposits. Because it is weaker than formic acid, it may be preferred where extremely long contact times are necessary. **Inhibitors** are specific compounds that are added to cleaning chemicals to diminish their corrosive effect on metals. Finally, **surfactants**, or **detergents**, are added to chemical cleaning solutions to improve their wetting characteristics. They are also used to improve the performance of inhibitors, and act as detergents in alkaline and acidic solutions.

Step 2: Passivation

A passive surface is one where the corrosion rate is reduced due to the precipitation of corrosion products on the metal surface. These corrosion products usually consist of oxides that inhibit further corrosion in water or in air. The term passivation is applied to procedures that are used to remove surface iron contamination from stainless steel equipment. To passivate stainless steels, mild iron contamination may be removed using a mixture containing 1% each of citric and nitric acids. For more persistent contamination, strong nitric acid solutions must be used.

8.8 Water Hammer

Water hammer occurs when the installation pipelines carry incompressible fluids such as water, ethylene glycol, etc., and the fluid flow suddenly changes its velocity. A common cause of water hammer is the rapid closing of a solenoid valve in the liquid line. Abruptly stopping the fluid flow produces a substantial pressure rise. High-intensity pressure waves will travel back and forth in the pipes between the point of closure and a point of relief, such as a larger diameter header, at extremely high speed. As it moves, the shock wave alternately expands and contracts the pipes.

Water hammer is the cause of many problems such as ruptured pipes and damage to valves, CBEs and other equipment. In a CBE, the water hammer will cause a bulge in the front or back plate, resulting in internal/external leakage (see Figure 8.44).

To avoid or eliminate these problems, the designer can install an air chamber or a **water hammer arrester**. Another way to control water hammer is to use valves with controlled closing times or controlled closing characteristics. The graphs in Figure 8.45 illustrate the difference between using standard quick-closing water valves and slow-closing time-controlled water valves.

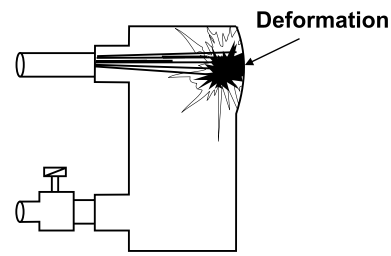


Figure 8.44 Water hammer causing deformation in a CBE.

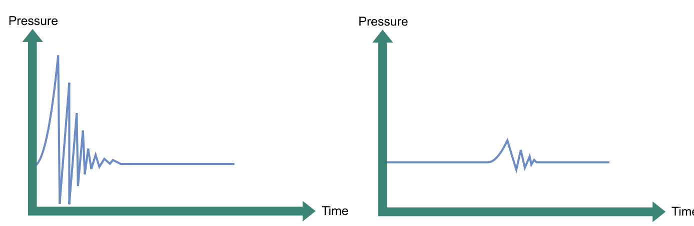


Figure 8.45 The pressure in the system can be seen to the right, for a "standard" quick closing valve, and to the left, for a slow closing time-controlled valve.

9 Troubleshooting

Dealing with refrigerant systems in reality differs from dealing with refrigerant system in theory. It is a matter of matching components with an adequate system layout. Although it works in theory, various problems may occur when running the actual system.

This chapter deals with troubleshooting and discusses potential problems with condensers and evaporators. When troubleshooting a system, it is important to have a systematic approach to collecting and analyzing data.

9.1 General

The first task when a system-related problem is reported should always be to gather information and evaluate the accuracy of the received information. Expending effort on a problem without accurate information can cause a lot of extra costs and lost time.

The data received in connection with problems are often third-hand information. This information is often presented as being more accurate than it actually is, due to the desire of all the parties involved for immediate action to correct the problem. To avoid being misled, it is important to have a systematic approach to dealing with system-related problems, i.e. collecting the correct data and evaluating them systematically.

Making a qualified evaluation requires accurate information. In a refrigerant application, this is often quite difficult, because every component and condition will affect the whole cycle. For example, if there is a problem in the discharge line it will affect the evaporator, and vice versa. However, the fact that the system data are linked to each other is an advantage, because it is possible to check the reasonableness of the received data.

9.2 Necessary Input

A very good method of troubleshooting is to enter the received data into a pressure/enthalpy diagram (see Figure 9.1). When the compression cycle has been plotted, it can be analyzed and compared with a “normal” cycle. It is also easy to compare the plotted data with data from the suppliers’ software.

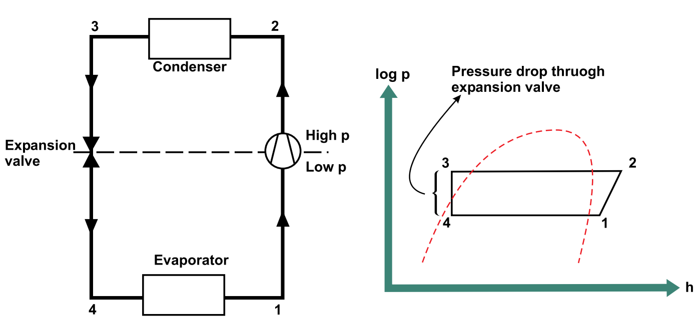


Figure 9.1 A log P/h diagram.

Refrigerant Side Parameters

To plot the compression cycle in a diagram, five measurements are required:

- Suction pressure (at evaporator outlet/compressor inlet)
- Suction temperature
- Discharge pressure (at compressor outlet/condenser inlet)
- Discharge temperature
- Liquid temperature (at condenser outlet/expansion valve inlet)

Liquid Side Parameters

To evaluate the CBE heat exchanger performance, two additional temperatures are required for each CBE:

- Liquid temperature into the evaporator

- Liquid temperature out of the evaporator
- Liquid temperature into the condenser
- Liquid temperature out of the condenser

System Capacity

To evaluate the heat exchanger performance, the system capacity must be known. This can be established in several ways:

1 The easiest, and normally most accurate, way is to process compressor data through the manufacturer's software with measured data as input. Nominal data and tables are normally based on other conditions.

2 Estimate the flow of liquid from:
   - Flow meters
   - Pressure drop over valves
   - Pump characteristics
   - Differential pressure over pumps The capacity can be calculated from the liquid flow and the temperatures (see chapter 1.3).

3 Measure the electrical input to the compressor (pumps and fans excluded). Use compressor software to calculate the capacity.

The first method is by far the easiest, and it is very accurate provided the pressures and temperatures are measured correctly and the compressor is not damaged. This can be verified by checking the compressor efficiency in the diagram, provided the normal compressor efficiency is known.

9.3 Methods for Collecting Input Data

Provided data are documented during the handling of a problem, and accurate measurements are obtained, in most cases it is possible for an experienced engineer to evaluate the data without visiting the installation.

Service Pressure Gauges and Thermometer

It can often be hard to collect the desired data. The normal measuring equipment is a service pressure gauge and preferably some type of thermometer. Data are commonly collected using fixed gauges or by connecting a service pressure gauge to the system. After the expansion valve or other component has been adjusted, the consequences of the change are monitored. However, a much better method is to gather more information systematically. Again, it is important to collect as much information as possible on-site. Data should be collected over as short a time span as possible, and measurements should be carried out several times over a period of time to verify the stability of the system.

Service pressure gauges should be calibrated regularly and be of good quality. The data obtained from the service pressure gauges will have a modest accuracy, and will depend greatly on the "human factor". They should therefore be used with caution. The data can be compared with the calculated performance of a system to find discrepancies.

A thermometer with a sensor attached with heat transfer paste, aluminum tape and insulation should be used (see Figure 9.2) to increase accuracy. If possible, fixed thermometers should be compared with a known reference, e.g. the

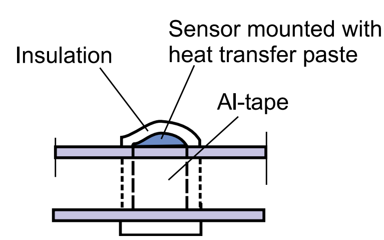


Figure 9.2 Thermometer with a sensor attached with heat transfer paste, aluminum tape and insulation.

service thermometer should be mounted at the same place as the fixed one.

A good way to check thermometers is to check the temperature difference over a heat exchanger and then stop the system for a few minutes, allowing the pump to circulate. The thermometers on each side of the heat exchanger should stabilize at the same temperature.

General Data Loggers

Loggers make it possible to collect accurate data simultaneously from several points and then display them over a period of time. Many loggers are only able to collect temperatures. However, they still give important information because evaporation and condensing pressures can be measured indirectly. The evaporation pressure/temperature is close to the temperature after the expansion device (the pressure drop in the distributor and evaporator should be compensated). The condensing pressure/temperature is close to the liquid temperature, when compensated for sub-cooling. Provided conditions and capacities do not change very much, fixed gauges or service pressure gauges can be used to “calibrate” these temperatures. Whenever possible, loggers that can measure both pressure and temperature should be used. This method will increase the accuracy of the data compared with data collected with regular service equipment.

Specialized Refrigeration Analyzers

Loggers are available with integrated software, specialized for refrigeration applications, that contains all the physical data for most refrigerants. These loggers collect all the relevant parameters, and intervals can be selected as desired. These analyzers are easy to use, because they are pre-programmed and can calculate and present all interesting data immediately on-site.

9.4 Evaporator Underperformance

The evaporation process is sensitive and potentially unstable. Minor changes in performance have major effects on the system performance. A one-degree change in the evaporation temperature changes the COP by approximately 3%. An unstable process could also cause the evaporation temperature to fluctuate, with a potential risk of freezing in the evaporator. The critical parameter for the system performance is the saturated pressure at the compressor inlet. Some checkpoints for determining the reason for underperformance are discussed below.

- **Check the installation**
  The exchanger should be mounted upright, with the refrigerant circuit connected to the left side when the arrow sticker is pointing upwards. The refrigerant and water circuits should be connected in counter-current mode. For further information, see chapter 8.2.
- **Check the evaporation temperature at the compressor inlet**
  Use compressor data to compare the cooling capacity with the design data.
- **Check the evaporation pressure at the evaporator outlet if possible**

Compare the evaporation pressure at the outlet with the evaporating pressure at the compressor inlet. This difference should be kept to a minimum to avoid unnecessary losses in cooling capacity. There is often only one place to measure the pressure on the suction side of the compressor. Measuring the temperature into the evaporator and knowing the calculated pressure drop over the CBE can give a good indication of the actual pressure drop.

- **Check the inlet/outlet water temperatures**
  Use the cooling capacity (above) to calculate the water flow rate, and compare it with the design data.
- **Check the amount of superheat**
  Adjust the superheat (normally 5-6 K) at a large pressure difference and check the stability. A high level of superheating could indicate that the expansion valve is too small.
- **Check the expansion valve**
  Make sure that the valve is able to regulate. If not, a larger valve should be installed. An expansion valve with external pressure equalization should be used. For further information, see chapter 4.2.
- **Check the positioning of the expansion valve sensor bulb**
  If the bulb is positioned too close to the evaporator, the evaporation temperature may fluctuate. The bulb should never be placed on the bottom of the suction line, because refrigerant droplets may evaporate at this point, affecting the regulation and causing locally low temperatures.
- **Check the external equalization line**
  The external equalization line should be close to the surrounding temperature. If this line is at evaporation temperature, it indicates a leak in the valve, i.e. refrigerant is flowing from the expansion valve to the suction line without passing the evaporator. This will cause the valve to close if the bulb is positioned after the connection for the equalization line, decreasing the evaporation temperature. If the bulb is positioned before the connection, the valve will not close and liquid refrigerant will be injected into the compressor.
- **Check the sight glass in the liquid line**
  There should be no bubbles in the liquid line. If there are, adjust the refrigerant charge or the operation of the condenser to establish sufficient sub-cooling. If there are bubbles, measure the liquid temperature. If bubbles occur even when there is sub-cooling, check whether there is a temperature change over any components in the liquid line, indicating a pressure drop.
- **Check the pressure drop over the secondary side of the evaporator**
  If the pressure drop is higher than the design conditions, fouling could be the reason. Make sure that the pressure drop is read at the design flow rate, because the pressure drop is proportional to the flow rate squared.

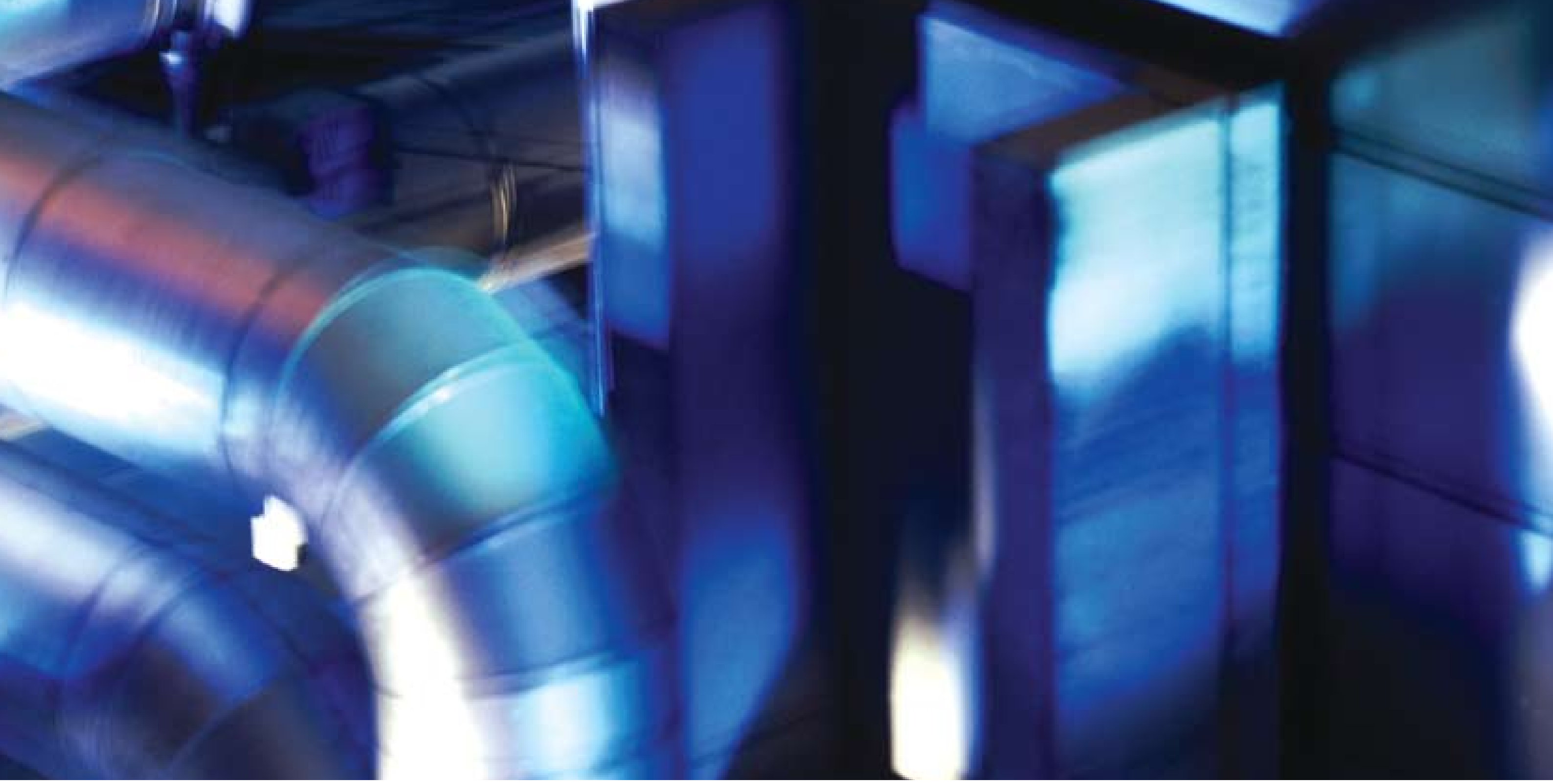
9.5 Condenser Underperformance

The condensing process is normally stable, and will not cause any major disturbances of the system performance. However, situations that reduce the capacity are possible. A major reduction in capacity is normally related to the evaporator performance. Problems with the condenser are usually manifested as an increased pressure head. Some checkpoints for determining the reason for underperformance are described below.

- **Check the installation**
  The exchanger should be mounted upright, with the refrigerant circuit connected to the left side when the arrow sticker is pointing upwards. The refrigerant and water circuits should be connected in counter-current mode with the refrigerant inlet at the top. For further information, see chapter 8.2.
- **Check the condenser for temperature variations on the outside**
  If inert gas is trapped, or water channels are obstructed, inside a CBE, there will be large surface temperature differences. Fouling can be identified as particles in the port or as general particle fouling on the plates.
- **Check cooling fluid if pure water is not used**
  A glycol solution with the wrong concentration will affect the heat transfer.
- **Check temperatures on the water side**
  Make sure that the entering and leaving water temperatures correspond to the design conditions. If the temperature difference is higher than the design conditions, the water flow rate has been reduced. Try to increase the water flow rate.
- **Check the amount of sub-cooling**
  Too much sub-cooling will block the heat transfer surface from the condensing process and thus increase the condensing pressure. This indicates that the system has been overcharged. Release the refrigerant charge until a reasonable level of sub-cooling (2-5K) has been achieved.
- **Check the compressor**
  The condenser capacity is always set by the compressor. Thus, if the compressor is working improperly, the condenser capacity will be affected. See chapter 3.4.
- **Check the suction pressure at the compressor inlet**
  Use the compressor data to compare the capacity with the design data.
- **Check the pressure drop over the secondary side of the condenser**
  If the pressure drop is higher than the design conditions, fouling could be the reason. Make sure that the pressure drop is read at the design flow rate, because the pressure drop is proportional to the flow rate squared.
- **Check the pressure drop in the discharge line**
  Make sure that none of the components in the discharge line, i.e.

between the condenser and compressor, has a pressure drop that is not taken into account. Oil separators, check-valves and mufflers can cause significant pressure drops if they are not correctly sized.

- **Check the discharge temperature**
  If the discharge temperature is significantly higher than in the design conditions, a larger surface is required for the desuperheating before condensing can start. This will reduce the condensing surface and increase the condensing pressure.



10 Systems

This chapter discusses various refrigerant systems that have been mentioned only briefly, or not at all, earlier in the handbook. These systems are specific, and have different application areas for heat exchangers. Some areas covered in this chapter are mentioned below.

True dual systems meet the need to be able to change the cooling capacity in large systems. Indirect systems are suitable if the aim is to keep the refrigerant charge at a minimum or avoid leakage in the system. The chapter also deals with reusing waste energy, heating or cooling buildings, cooling at very low temperatures and drying air.

10.1 True Dual Systems

With larger systems, two independent refrigerant circuits are commonly used to follow the changes in cooling demand better and to increase system reliability. If one circuit malfunctions, the side remaining in operation may still provide sufficient cooling until help arrives.

The dual circuit system could be implemented by arranging two CBEs in parallel, or mounting a "back-to-back" or "false dual" circuit CBE. However, this would be at the expense of additional piping or a higher risk of freezing. Part-load efficiency also decreases with these arrangements, because the flow arrangement means that only 50% of the secondary fluid undergoes heat exchange. The evaporation temperature at part load may therefore decrease, reducing system efficiency and increasing the risk of freezing.

Instead, SWEP True Dual technology CBEs have two independent refrigerant circuits combined with a common secondary fluid circuit. The patented plate technology ensures full counter-current flow and full symmetry between the refrigerant circuits. The True Dual models are available with or without the SWEP distribution device for evaporator or condenser duty. A True Dual heat exchanger is shown in Figure 10.1.

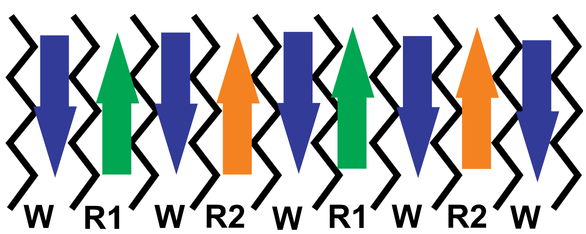


Figure 10.1 Cross-section of the channels inside a True Dual with both refrigerant sides operating.

A True Dual CBE running with both circuits active operates no differently from a high-efficiency single circuit evaporator with full contact between refrigerant and secondary fluid.

Even if one refrigerant circuit is closed, i.e. half-load operation, all secondary fluid channels remain in contact with the active refrigerant channel (see Figure 10.2). All the secondary fluid will still receive heat exchange, and the leaving water temperature will therefore be the same as for full-load operation provided the water flow is also halved. This allows the part-load evaporation temperature to remain at a high level, resulting in increased efficiency at part load. Because secondary fluid channels will surround the active refrigerant circuit, the evaporating process will also remain fully stable. True Dual technology therefore results in higher operational efficiency than

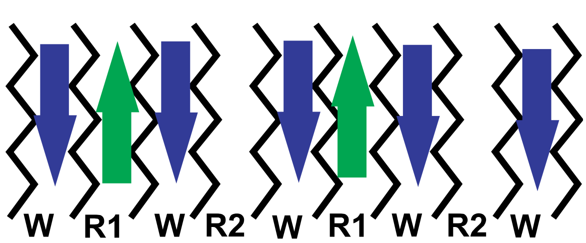


Figure 10.2 Cross-section of the channels inside a True Dual with only one refrigerant circuit operating.

"back-to-back" or "false-dual" CBEs, without additional risk of freezing at part load, and it requires less piping. For a schematic system sketch of a True Dual system, see Figure 10.3.

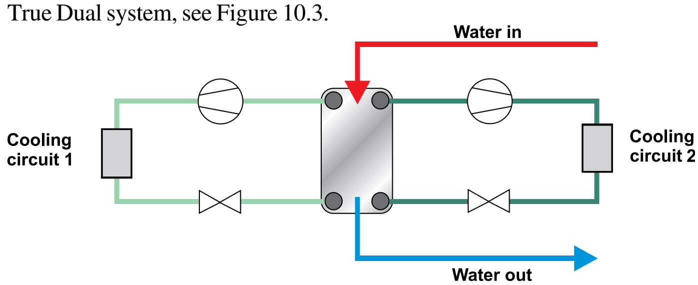


Figure 10.3 Schematic system sketch of a True Dual system.

10.2 Indirect Refrigerant Systems with CBEs

There are two reasons for using an indirect refrigerant system. First, the amount of refrigerant can be kept at a minimum. Second, the risk of leakage of primary refrigerant is decreased. This means that refrigerants that are not wanted in systems in public buildings (e.g. ammonia) can still be used in the primary system, which can be kept in a safe sealed room.

Using a CBE as both evaporator and condenser in a refrigerant system will result in the most efficient and compact refrigeration system available. Utilizing secondary fluids (water or brines) as carriers for the cold and heat will bring the system size and refrigerant charge to an absolute minimum.

No bulky air coil condenser is needed adjacent to the refrigerant system, which reduces the problem of noise. Instead, the secondary fluid can easily be carried away from the system and cooled at a convenient distance in a dry cooler. Additionally, the potential problem of transporting refrigerant over a long distance is avoided.

A good example of indirect refrigeration systems is in a supermarket (see Figure 10.4). All the cooling effect required is produced in the machine room, far from the display cases, resulting in high flexibility and a lower risk of leakage. A heat recovery system, also situated in the machine room, provides heating for the supermarket when required.

As discussed in chapter 10.1, the use of dual circuit heat exchangers, such as the SWEP True Dual models, is advantageous because the independent refrigerant circuits increase the safety and availability of cooling. This can be important in supermarkets, for example, where the total value

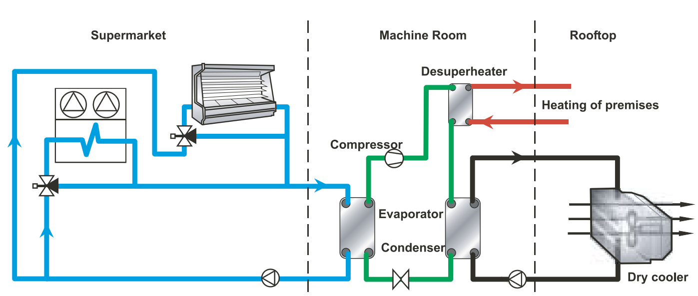


Figure 10.4 An indirect refrigeration system with a central cooling machine, separated from the supermarket area by the harmless heat transfer fluid.

of refrigerated or frozen groceries could reach significant levels. In the case of an indirect refrigeration system, the evaporator (see Figure 10.4) can be changed to a True Dual CBE.

10.3 Desuperheating/Heat Recovery

In general, the practical use of waste energy is called heat recovery. Figure 10.5 shows an example of such a system. Refrigeration plants with air-cooled condensers produce a lot of waste energy by dumping the condensation energy to the ambient air. By installing a CBE desuperheater, a large proportion of this waste energy can be turned into hot water that may be used for many purposes such as:

- sanitary hot water
- room heating
- hot water for processes
- cleaning water

Desuperheater units are located between the compressor and condenser to utilize the high-temperature energy of the superheated refrigerant gas. By using a separate heat exchanger to utilize the high temperature of the discharge gas, it is possible to heat water to a higher temperature than would be possible in a condenser (see Figure 10.6).

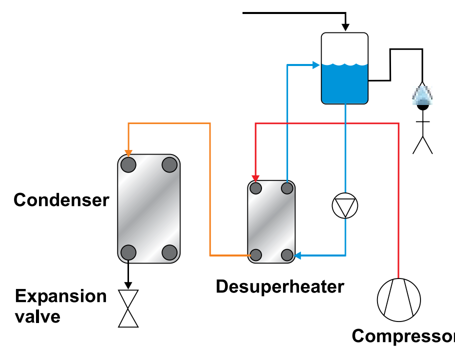


Figure 10.5 Schematic layout of a heat recovery system to provide hot tap water.

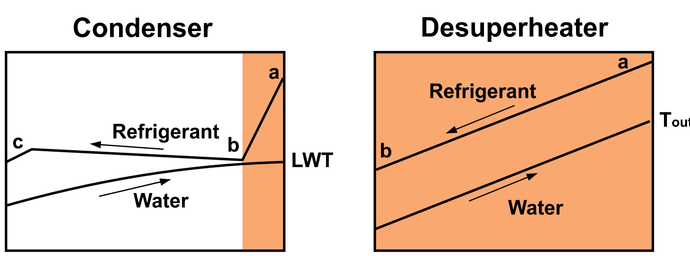


Figure 10.6 Temperature profile comparison between a condenser and a desuperheater. In the left picture, the desuperheating (a-b) is performed in the same heat exchanger as condensing and subcooling (b-c). If the desuperheating takes place in a desuperheater situated before the condenser, the absence of a temperature pinch will increase the heat transfer even for high water temperatures. Thus, a T_{out} higher than LWT can be obtained.

The total amount of available superheating depends on the difference between the discharge temperature from the compressor (point a in Figure 10.7) and the condensing temperature of the refrigerant gas (point b). The system can be arranged to bypass the desuperheater CBE if no hot water is required. The condenser should then be designed to be able to handle the full condensing capacity.

The desuperheater is normally designed not to condense any refrigerant. However, some liquid refrigerant may form, depending on operating conditions. This liquid must be carried to the condenser, which ideally should be situated below the desuperheater. However, for practical reasons it is often placed above the desuperheater. In a brazed heat exchanger, the droplets of refrigerant leaving will be dispersed and easily carried over by the dominant vapor phase. Designing the connection pipe from desuperheater to condenser for a gas velocity of 5-10 m/s will provide sufficient turbulence to avoid the accumulation of liquid condensate.

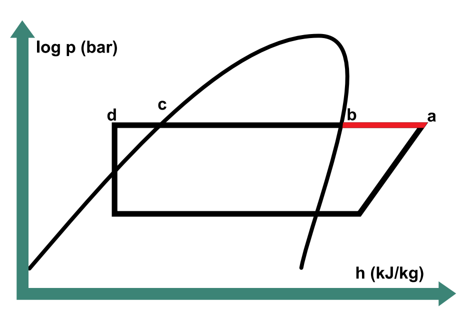


Figure 10.7 A log P/h diagram showing the available desuperheating energy in a refrigerant system.
(a) indicates the compressor discharge temperature.
(b) indicates the saturation temperature, i.e. where the refrigerant starts to condense.

Scaling is a potential problem in desuperheaters when the secondary fluid is water, because the solubility of limestone ($CaCO_3$) decreases with increasing temperature. The maximum water temperature should ideally not exceed 65-70°C to avoid scaling problems. If the risk of scaling is increased by the use of hard water, etc., the use of co-current flow should be considered to reduce the risk of excessively high water temperatures.

10.4 Heat Pumps

Dedicated Heat Pumps

A refrigerant system can be used to heat residential or commercial areas by utilizing the condensation energy. Heat required by the evaporation comes from the surroundings through a loop of brine in the ground (see Figure 10.8) or from a lake, the ambient air or the bedrock. Dedicated heat pumps are common in cold countries where the low requirement for air conditioning during the summer makes reversible heat pumps less attractive. By optimizing the heat pump for heating only, the operating cost will be lower than for reversible heat pumps. Dedicated heat pumps can also be used to produce hot tap water by increasing the condensing temperature when hot water is required or by combining the system with a desuperheater.

Dedicated heat pumps operate with relatively low temperature differences in both evaporator and condenser to maximize the coefficient of performance (COP). The use of a SWEP distribution device is highly recommended to maximize efficiency. The system performance can also be improved by using a separate condensate sub-cooler or liquid suction heat exchanger.

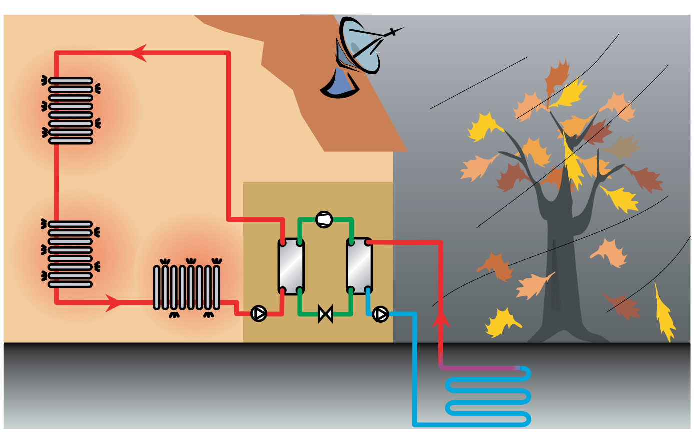


Figure 10.8 A dedicated heat pump system with a secondary fluid loop buried in the garden. The temperature increase is utilized to evaporate the refrigerant in the CBE evaporator. After compression, the "free" energy from outside is converted to comfortable heat.

Residential heating by heat pumps is supported by the governments of some countries as a step towards higher energy efficiency and a reduction in "dirty" heating, e.g. with local oil/coal burners.

Reversible Heat Pumps

Reversible heat pumps can be utilized to provide both air conditioning and heating, depending on the requirements. The operation of the compressor and secondary fluid is constant, but the refrigerant flow is reversed by a four-way valve. Figure 10.9 shows reversible heat pump systems with a CBE working as a condenser and an evaporator, respectively.

Because the direction of flow of the secondary fluid is constant, the heat exchange for the evaporator/condenser will be parallel for one operation. Parallel, or co-current, flow gives a lower mean temperature difference and thus lower performance. The decision to have the evaporation or condensation operate in parallel therefore depends on the climate and thus the system requirements. Normally, the evaporator is operated in counter-current and the condenser in parallel.

SWEP evaporator models with distribution systems have a higher efficiency than those without, and they operate without problems when reversed, i.e. in condenser duty. The pressure drop induced by the distribution device is negligible when only liquid passes through, as is the case when operating as a condenser.

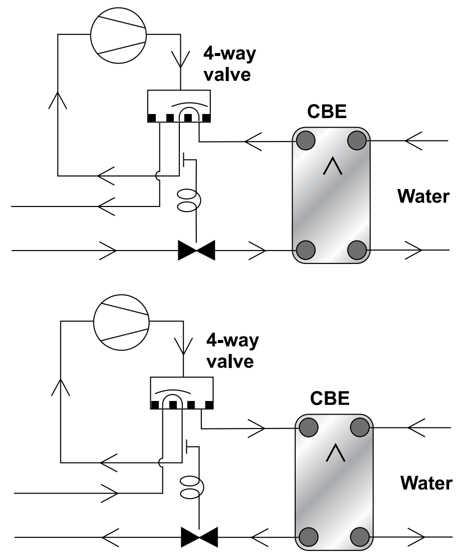


Figure 10.9 Schematic view of a CBE reversible heat pump operating in parallel flow as a condenser (bottom), and in counter-current flow as an evaporator (top). The four-way valve directs the refrigerant flow after the compressor and controls the operation.

Exhaust Air Heat Pump

In houses with ventilation systems in cold climates, the warm exhaust air can be used to operate a heat pump. The use of highly efficient CBEs as condensers makes it viable to utilize the warm air to evaporate the refrigerant in an air coil before releasing it. This minimizes the waste energy (see Figure 10.10).

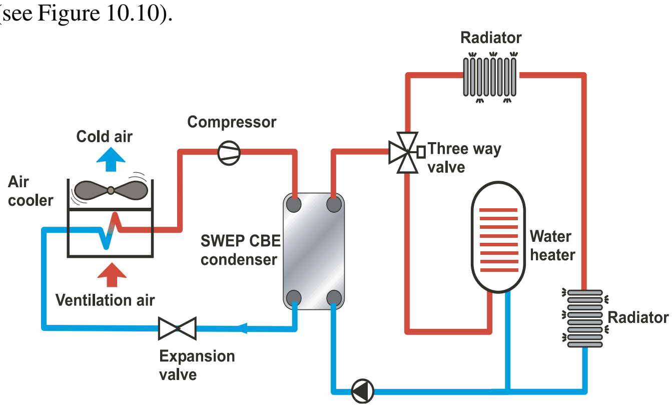


Figure 10.10 Exhaust air heat pump with a CBE condenser.

10.5 Subcoolers

In the condenser, the liquid refrigerant is cooled slightly below the saturation temperature to ensure that no flash gas is formed before the expansion valve. The level of sub-cooling achieved in the CBE condenser is normally 0.5-4K. More sub-cooling further increases the available evaporator capacity by decreasing the amount of flash gas formed after the expansion valve (see Figure 10.11). The decreased vapor content also decreases the risk of maldistribution in the evaporator. This additional sub-cooling is best carried out in a separate sub-cooler, because a high level of condensate inside the condenser may disturb the stability and thus the performance of operation. The simplest sub-cooler uses a liquid stream, which can be the same as for the condenser (see Figure 10.12).

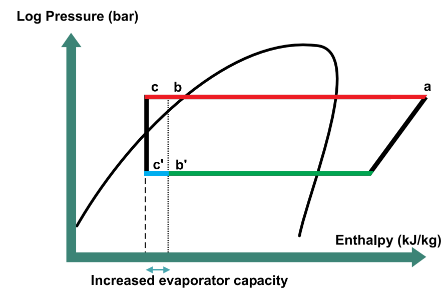


Figure 10.11. A log P/h diagram showing the effect of increased sub-cooling of the condensate. The increased sub-cooling from (b) to (c) reduces the flash gas after the expansion valve and increases the available cooling capacity inside the evaporator.

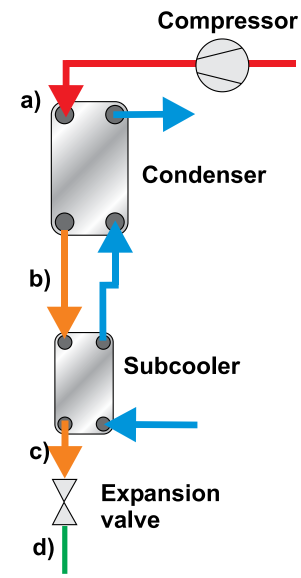


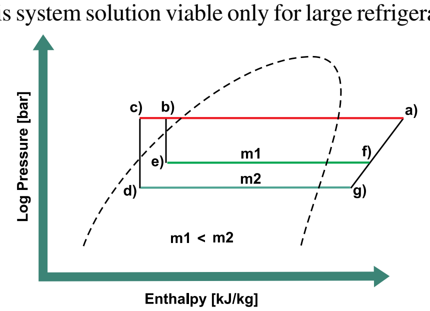
Figure 10.12 Schematic design of a condenser and a separate sub-cooler with a shared secondary fluid stream

10.6 Economizer

An economizer is a type of sub-cooler that uses part of the total refrigerant flow from the condenser to cool the rest of the refrigerant flow (see Figure 10.13). The evaporated refrigerant then enters the compressor at an intermediate pressure level. The cold gas from the economizer can also be used to provide extra cooling for the compressor.

The sub-cooling of the main refrigerant flow (m2) reduces the quality of the inlet vapor to the evaporator, which increases the cooling capacity. The high efficiency of the SWEP CBE economizer minimizes the required temperature difference between the sub-cooling and the evaporating streams, which in turn increases the overall efficiency of the system.

Economizer systems require extra components, such as piping and a compressor with an "economizer" entrance. The additional capital expense makes this system solution viable only for large refrigeration systems.



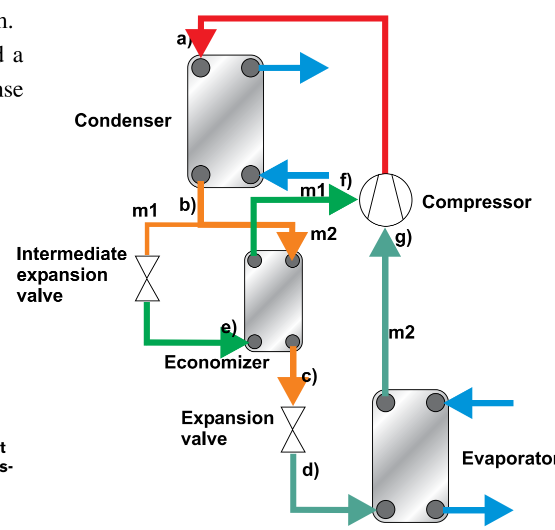
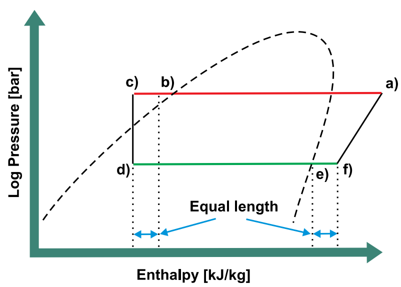


Figure 10.13 An economizer system (right) and a log P/h diagram (above). The refrigerant flow is divided after point (b). The smaller flow (m1) is evaporated at an intermediate pressure to cool the main flow (m2). The cold intermediate gas (f) enters the compressor in a special economizer port.

10.7 Suction Heat Exchanger

The hot condensate liquid from the condenser can be utilized to superheat the cold vapor from the evaporator in a suction heat exchanger, with two positive effects. The higher level of sub-cooling increases the evaporator capacity, as discussed above. At the same time, the superheating in the evaporator can be minimized, because the suction heat exchanger ensures that no liquid enters the compressor. This results in a more efficiently utilized heat surface inside the evaporator, allowing a higher evaporation temperature or the use of a smaller CBE. Because the refrigerant mass flow is the same on both sides of the suction heat exchanger, the enthalpy decrease of the condensate exactly corresponds to the enthalpy increase of the vapor. An excessively high level of vapor superheating may create problems with elevated discharge gas temperatures, which may limit the level of sub-cooling.

Systems with suction gas heat exchangers can become unstable if the system load fluctuates, so this system solution is favored for stable systems. Alternatively, an electronic expansion valve can be used to handle the fluctuations. A suction heat exchanger system is shown in Figure 10.14.



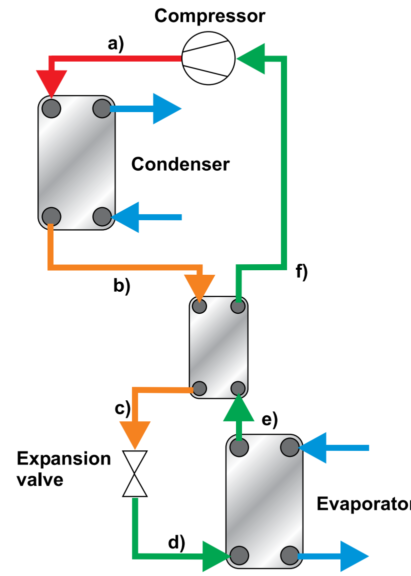


Figure 10.14. A liquid suction heat exchanger system with a log P/h diagram. The suction heat CBE exchanger simultaneously superheats the evaporator vapor and subcools the condensate.

10.8 Low Temperature Systems

Operating at very low evaporation temperatures (e.g. when providing cooling for a deep-freezer and still using ambient air or similar as a heat sink for the condenser) often requires special system solutions.

Low-temperature evaporation requires low evaporation pressures while the condensing pressure is at normal levels. It is often beneficial, and in some cases necessary, to separate the evaporation and condensing pressure levels by more than one compressor step. This is because when the pressure ratio over the compressor increases, the discharge temperature out of the compressor will also increase. Simultaneously, the compressor efficiency decreases, which increases operating costs. High discharge temperatures may cause both the refrigerant and the lubrication oil to decompose. This in turn will shorten the life of the compressor. Figure 10.15, point (a1), shows the higher discharge temperature of a single-step refrigeration system with low evaporation temperature.

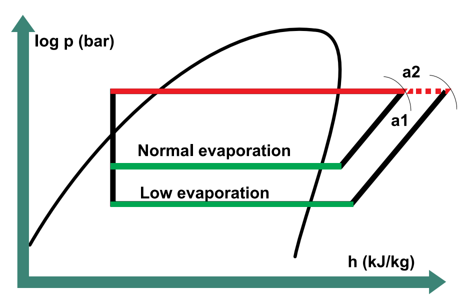


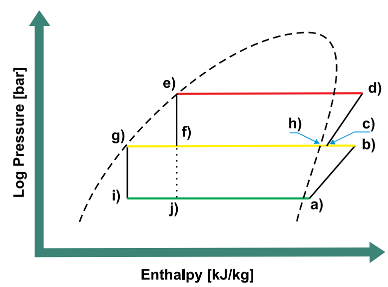
Figure 10.15 A log P/h diagram showing the resulting higher discharge temperature for a larger compression step.

Two-Stage Systems

Intermediate gas cooling is often used between the two compressor steps. By cooling the refrigerant vapor after the first compressor, the discharge gas leaving the high-stage compressor can be kept at an acceptable temperature level. The intermediate cooling also increases the compressor efficiency, which reduces the compressor power consumption.

A two-stage system is a refrigeration system working with a two-stage compression and mostly also with a two-stage expansion. A schematic system layout and the corresponding process in a log P/h diagram are shown in Figure 10.16. Flash gas is separated from liquid refrigerant in an intermediate receiver between the two expansion valves. The high-stage compressor will then remove the flash gas, as shown in Figure 10.16. The removal of the gas between the expansion stages reduces the quality of the refrigerant vapor that enters the evaporator from the state ‘j’ (which would be the vapor quality if only one expansion valve were used) to the state ‘i’, as shown in the log P/h diagram of Figure 10.16.

Due to a lower quality entering vapor, each mass unit of refrigerant passing through the evaporator will be able to absorb more heat, reducing the required refrigerant mass flow rate for a given cooling capacity. This in turn reduces the required low-stage compressor size. Because of the enhanced heat transfer coefficient in the evaporator, the heat transfer area needed is also reduced.



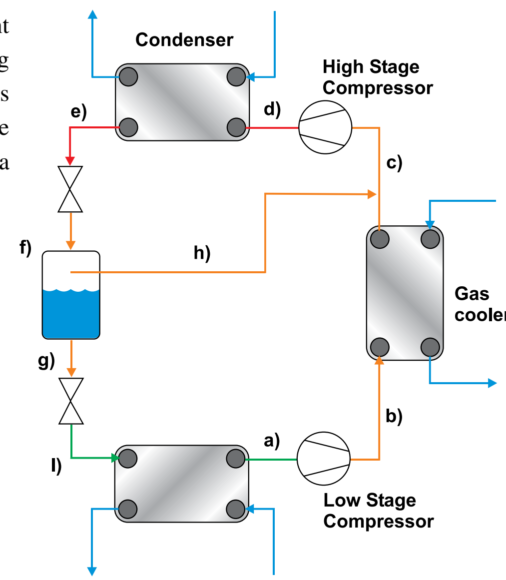
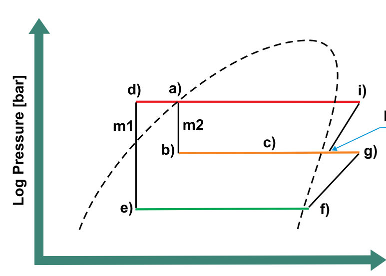


Figure 10.16 Schematic layout of a two-stage low-temperature refrigeration system (right) with a log P/h diagram (above).

Intercooler System

An intercooler system uses an intermediate evaporation step, similar to the economizer system, to cool the discharge gas from the first compressor step. The two-stage intercooler system is shown together with a corresponding log P/h diagram in Figure 10.17.



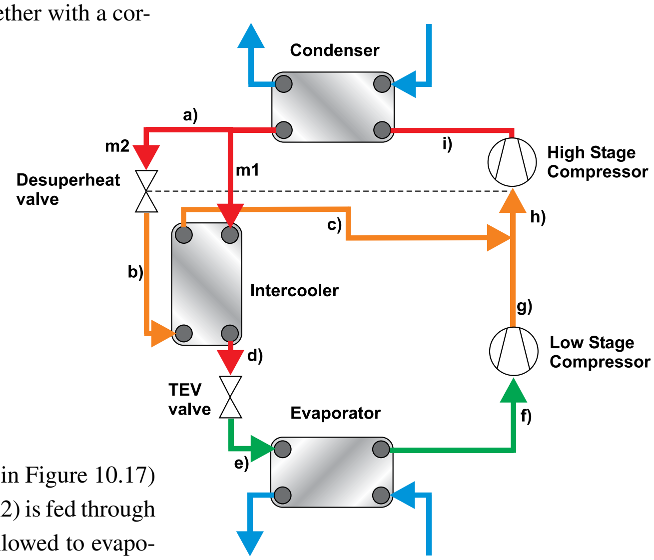


Figure 10.17 Schematic layout of an intercooler system (right) with a log P/h diagram (above).

The refrigerant liquid leaving the condenser (state ‘a’ in Figure 10.17) is split into two streams. The smaller part of the liquid (m2) is fed through an intermediate expansion valve (‘a’ to ‘b’), and then allowed to evaporate on one side of the CBE intercooler (‘b’ to ‘c’). The main flow (m1) is sub-cooled by leading it through the other side of the CBE intercooler (‘a’ to ‘d’). The sub-cooled refrigerant liquid leaving the intercooler is fed through the main expansion valve (‘d’ to ‘e’) and then through the main evaporator ('e' to 'f'). The sub-cooling decreases the inlet vapor quality, which reduces the refrigerant mass flow rate through the evaporator and the required low-stage compressor size for a given cooling capacity.

The intermediate refrigerant stream (m2) is not completely vaporized when leaving the intercooler (state 'c'). The remaining liquid is evaporated when it is mixed with the hot discharge gas from the low-stage compressor. This results in efficient gas cooling ('g' to 'h'). The discharge gas from the high stage compressor can be kept at an acceptable temperature (state 'i'), and the compressor efficiency is increased.

The high efficiency of the SWEP CBE minimizes the temperature difference between the evaporating stream (m2) and the sub-cooled stream (m1), which increases the overall efficiency of the system.

Cascade Systems

The cascade system consists of two separate refrigeration circuits connected only by an intermediate cascade heat exchanger. As shown in Figure 10.18, the high-temperature circuit is cooled by an air condenser (2) at ambient temperature, and uses the cascade heat exchanger (1) as the system evaporator. The low-temperature system produces the low-temperature cooling in the cold evaporator (3), and uses the cascade exchanger as a condenser. The corresponding outline in a log P/h diagram is shown in Figure 10.19. The cascade heat exchanger connects the two refrigerant circuits thermally by acting simultaneously as an evaporator and a condenser.

The primary advantage of a cascade system is that the two stages do not necessarily contain the same refrigerants. A refrigerant with a higher vapor pressure can be used in the low-temperature system, while a refrigerant with a lower vapor pressure is suitable for the high-temperature system.

Multi-stage refrigeration cycles can also achieve very low temperatures efficiently, but there are some major disadvantages compared with the cascade cycle. In multi-stage refrigeration, the same refrigerant must work at the highest and the lowest pressure levels. The selection of refrigerant to avoid excessively large pressures in the ambient condenser, and evaporation pressures below one atmosphere in the cold evaporator, can be difficult. Vacuum should always be avoided, because this increases the risk of air and moisture leaking into the system, leading to reduced system performance and increased wear on components. Because the refrigerant oil has a higher solubility in the refrigerant at higher temperatures, a multi-stage system also has a higher risk of uneven oil distribution, giving lubrication problems in the low-stage compressor.

By contrast, refrigerant selection and oil distribution for a cascade system can be dealt with separately for each circuit. It is important to note that the cascade heat exchanger will be exposed to temperature and pressure fluctuations. In the cascade unit, the evaporating side typically operates at -10 to -20°C. The discharge gas from the low-temperature compressor may very well be 80°C or higher. To avoid the risk of thermal fatigue inside the cascade unit due to the very high temperature differences, the

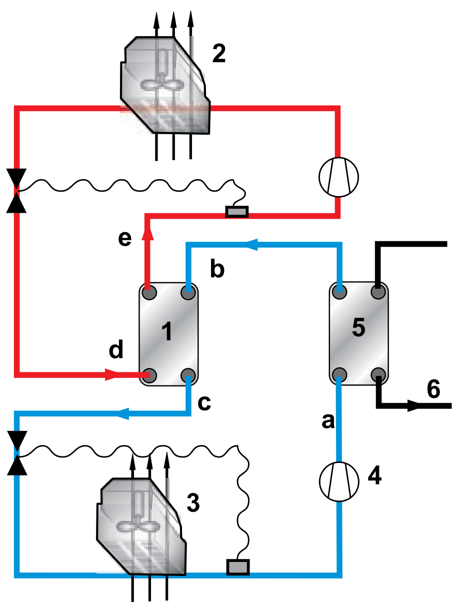


Figure 10.18 Schematic layout of a cascade system. (1) Cascade unit (2) Ambient air condenser (3) Low-temperature air evaporator (4) Low-temperature compressor step (5) CBE desuperheater (6) Hot tap water.

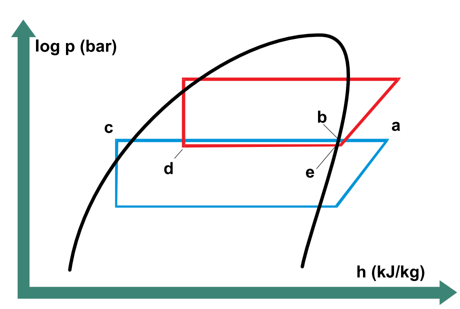


Figure 10.19 A log P/h diagram of a cascade system showing the low-temperature cycle in blue and the warm cycle in red. (a) Discharge temperature for the low-temperature cycle (b) Inlet vapor temperature to cascade unit after desuperheater (c) outlet condensate temperature for the cold cycle (d) Temperature after expansion valve in the warm cycle (e) evaporation temperature in the warm cycle.

installation of a desuperheater (5) is recommended before the inlet on the condensing side, i.e. in the “cold” circuit.

The desuperheater reduces the inlet gas temperature of the condensing side of the cascade unit, while utilizing the superheating energy to produce high-temperature water (6). The main duty, however, is to reduce the gas temperature. The desuperheater must therefore not be bypassed.

10.9 Absorption Chillers

Conventional compressor air-conditioning chillers are powered by electricity. Absorption chillers/heaters, on the other hand, use high-temperature heat as their main energy source. A very small amount of electricity is needed in absorption systems compared with compression cycle systems, because only the pumps are operated by electricity. However, the investment cost of the system is much higher than for a compressor system. Absorption chillers are therefore used mostly for large installations when electricity is limited and/or heat is abundant. Absorption chillers can be used for both heating and cooling purposes simultaneously, by using the process cooling water from the absorber and the condenser.

The most efficient modern absorption cycle chillers use water as the refrigerant and a solution of lithium bromide (LiBr) as the absorbent. The LiBr concentration is typically around 64% after the generator and approximately 60% after the absorber. The simplest absorption system is a one-stage system with one absorber and one generator. The benefits include lower investment costs, but the trade-off is the lower efficiency. The use of multistage absorbers or generators increases the system performance, but also the investment cost. An absorption chiller system is shown in Figure 10.20.

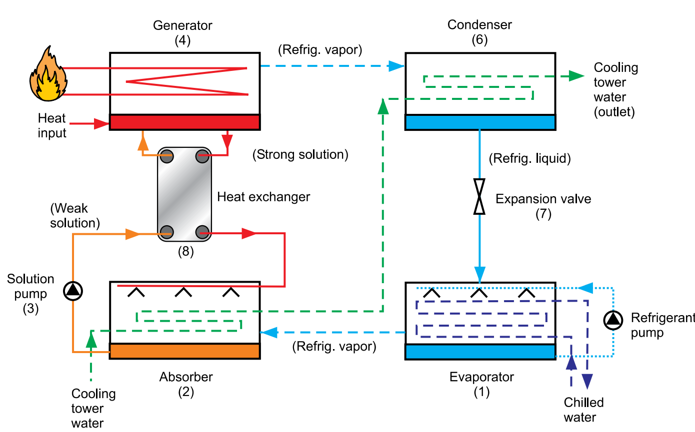


Figure 10.20 A one-stage absorption system. Water is chilled in the evaporator (1) by evaporating water at low pressure, circulated by the refrigerant pump. The water vapor enters the absorber (2) due to the slightly lower pressure. A concentrated LiBr solution absorbs water, and the temperature increases slightly due to the heat of absorption. This heat is carried away by process cooling water. The diluted LiBr solution, containing the water vapor, is preheated in a heat exchanger (8) before it is pumped (3) to the generator (4) for evaporation at high pressure (~1 atm). Refrigerant vapor (water) enters the condenser at the lower pressure. The condensation energy is transferred to the second loop by the process cooling water. This can be cooled in a liquid chiller or used as heat source. The pressure is reduced by the expansion valve (7), and the refrigerant flow then enters the evaporator (1).

CBEs are used to exchange heat between different flows in the system. Typical applications include heat exchange in the circulating stream between the hot LiBr stream from the generator and the low-temperature stream from the absorber. This heat exchanger is often referred to as "high-temperature". Other CBEs can be used to cool further the LiBr stream entering the absorber, "low-temperature", and another to preheat further the stream entering the generator.

Due to the true counter-current flow and highly efficient heat transfer surface in a CBE, the temperature driving force of heat exchange can be kept at a minimum, which is important for the system performance. The efficiency of CBEs is also less sensitive to the low flow velocities than that of Shell and Tube (S&T) heat exchangers.

Lithium bromide in concentrations used currently (60-65%) is potentially very corrosive to stainless steel (AISI 316) and copper. The risk of corrosion also increases strongly with temperature. However, with added corrosion inhibitors (molybdates), empirical experience from customer installations shows that the copper-brazed CBE technology can withstand corrosion provided no air (oxygen) enters the system and the temperature stays below ~110°C. Some as yet unconfirmed field results also suggest that this temperature limit may be higher.

A nitrogen blanket before charging the system, and vacuum purge during operation, are required to ensure that there is no air contact. Because the system operates below atmospheric pressure, it is also imperative that the system be as well-sealed as possible.

10.10 Air Dryers

In several types of automated industries, compressed air is used for various purposes such as spray painting or pneumatic tools (compressed air equipment). A common problem in compressed air systems is the condensation of humidity. It appears along distribution pipes, blocking filters and machinery, and causes malfunction of the system. The water can cause corrosion and considerable damage to the equipment using the air. The condensation is due to the water vapor in the compressed air. This vapor is cooled along the pipes, and thus transformed to water droplets.

To avoid water condensation in a pneumatic system, an air dryer can be installed in the system (see Figure 10.21). The basic function of the air dryer is to remove moisture from the air by cooling it with a refrigerant. Thus, the water vapor is condensed, and the air can be compressed. The result is dry compressed air, which can be used in compressed air equipment without causing any damage.

In a pneumatic system, the air leaving the compressor (point 1 in Figure 10.21) contains a significant amount of water. In the after-cooler (2), up to 70% of the moisture can be removed. After the separator (3), the compressed air is still saturated with moisture. The air is then stored in a receiver (4), which can be installed before or after the air dryer (6). The goal of the air dryer is to eliminate the remaining 30% of the moisture.

The system design with the air dryer after the receiver, as shown in Figure

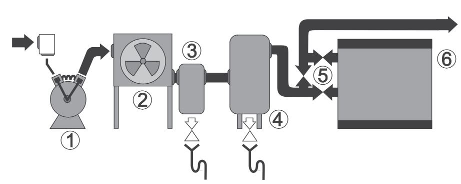


Figure 10.21 A pneumatic system with an air dryer. The components are in order a compressor, an after-cooler, a separator, a receiver, a bypass and an air dryer.

10.21, is recommended when the compressor operates intermittently and the air demand is not more than the maximum compressor capacity. If the air demand can exceed the maximum capacity of the compressor, the installation of the air dryer before the receiver is recommended.

The cooling conditions in a pneumatic system can be demonstrated by monitoring the pressure, temperature and humidity during the compressing and cooling stages, as shown in Figure 10.22.

In the compressor, free air at 20°C, 1 bar and 50% relative humidity is compressed to typically 7 bar g. During compression, the temperature increases (from 20 to 80 °C), and the relative humidity decreases. Because no vapor condensation takes place, the absolute humidity is constant. When the air is cooled in the after-cooler, the relative humidity increases to 100%. After the "dew point" (where the air is saturated), the vapor condenses with a reduced absolute humidity as a result. Most of the remaining humidity is condensed in the air dryer. During this condensing process, the air is in saturated conditions, which correspond to a relative humidity of 100%. When re-heating the air in the air dryer, the relative humidity will again decrease, as shown in Figure 10.22.

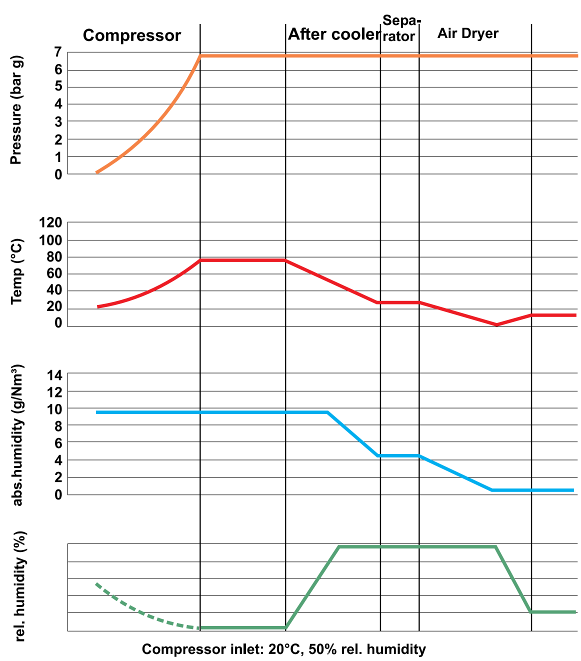


Figure 10.22 Pressure, temperature and humidity changes along a pneumatic system.

SWEP air dryers are an ingenious version of the compact brazed heat exchanger. They transform the moist warm air from an ordinary air compressor to dry air at ambient temperature. Figure 10.23 explains the flow pattern, and shows the port locations.

The warm, moist air from the compressor (point A in Figure 10.23) is pre-cooled in an air-to-air heat exchanger (the air from the compressor being cooled by the leaving dry air). Then the moist air enters the second chamber of the CBE where it is further cooled by the refrigerant from the refrigerant compression cycle (E-F). The condensate is then separated from the air stream at point (B). The cold dry air then makes a second pass through the air-to-air heat exchanger to be reheated before leaving the dryer.

The SWEP CBE air dryer offers a very compact and cost-effective alternative to other heat exchange solutions by combining high heat transfer with stable performance and creative engineering.

The B12 has been specifically developed as a heat exchanger for air dryers. The majority of the conventional CBE models create relatively high pressure-drop in air dryer applications. If the pressure drop in the heat exchanger is too high, it will reduce the overall performance of the pneumatic system, i.e. the system will not be able to deliver the required pressure to the pneumatic tools. The B12, however, has a low pressure-drop. It can handle asymmetrical flows and can be used as a combined air-to-air and refrigerant-to-air heat exchanger.

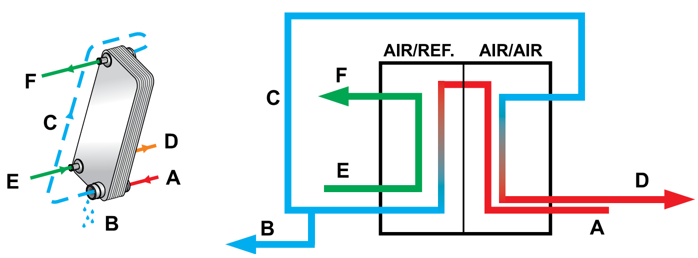


Figure 10.23 A SWEP air dryer. The warm, moist air from the compressor (A) first enters the air-to-air part of the dryer, where it is pre-cooled by the leaving dry air. After this passage, the moist air enters the second chamber where a refrigeration circuit (E-F) evaporates, thus reducing the air temperature dramatically. The corrugated CBE plates aggregate the water mist into droplets that are separated outside the unit at point (B). The cold dry air finally re-enters the air-to-air part of the heat exchanger, where it is heated to ambient temperature by the entering moist, hot air.

Appendix

A – Guide to the resistance of copper and stainless steel in CBEs

The guide to resistance in Table A.1 is an attempt to give a picture of the resistance of AISI 316 stainless steel and pure copper (99.9%) in water to corrosion by some important chemical factors. However, corrosion is actually a very complex process influenced by many different factors in combination. This table is therefore a considerable simplification, and its value should not be overestimated.

Explanations:

+ Good resistance under normal conditions
0 Corrosion problems possible, particularly when there are other factors rated 0
- Use is not recommended

Table A.1 Guide to resistance of copper and stainless steel in CBEs

WATER CONTAINING	CONCENTRATION (mg/l or ppm)	AISI 316	COPPER
Alkalinity (HCO_3^-)	<70	+	0
	70-300	+	+
	>300	+	0/+
Sulfate (SO_4^{2-})	< 70	+	+
	70-300	+	0/-
	> 300	0	-
HCO_3^- / SO_4^{2-}	> 1.0	+	+
	< 1.0	+	0/-
Electrical conductivity	< 10 µS/cm	+	0
	10 - 500 µS/cm	+	+
	> 500 µS/cm	+	0
pH	< 6.0	0	0
	6.0 - 7.5	0/+	0
	7.5 - 9.0	+	+
	> 9.0	+	0
Ammonia (NH_3)	< 2	+	+
	2-20	+	0
	> 20	+	-
Chlorides (Cl^-)	< 300	+	+
(Please also see table B.2)	> 300	0	0/+
Free chlorine (Cl_2)	< 1	+	+
	1-5	+	0
	> 5	0/+	0/-
Hydrogen sulfide (H_2S)	< 0.05	+	+
	> 0.05	+	0/-
Free (aggressive)	< 5	+	+
carbon dioxide (CO_2)	5-20	+	0
	>20	+	-
Total Hardness (°dH)	4.0 - 8.5	+	+
Nitrate (NO_3^-)	< 100	+	+
	> 100	+	0
Iron (Fe)	< 0.2	+	+
	> 0.2	+	0
Aluminum (Al)	< 0.2	+	+
	> 0.2	+	0
Manganese (Mn)	< 0.1	+	+
	> 0.1	+	0

Table B.2 Choice of plate material

CHLORIDE	MAXIMUM TEMPERATURE			
CONTENT	60°C	80°C	120°C	130°C
≤ 10 ppm	SS 304	SS 304	SS 304	SS 316
≤ 25 ppm	SS 304	SS 304	SS 316	SS 316
≤ 50 ppm	SS 304	SS 316	SS 316	Ti / 254 SMO
≤ 80 ppm	SS 316	SS 316	SS 316	Ti / 254 SMO
≤ 150 ppm	SS 316	SS 316	Ti / 254 SMO	Ti / 254 SMO
≤ 300 ppm	SS 316	Ti / 254 SMO	Ti / 254 SMO	Ti / 254 SMO
> 300 ppm	Ti / 254 SMO	Ti / 254 SMO	Ti / 254 SMO	Ti / 254 SMO

SS = Stainless steel grade

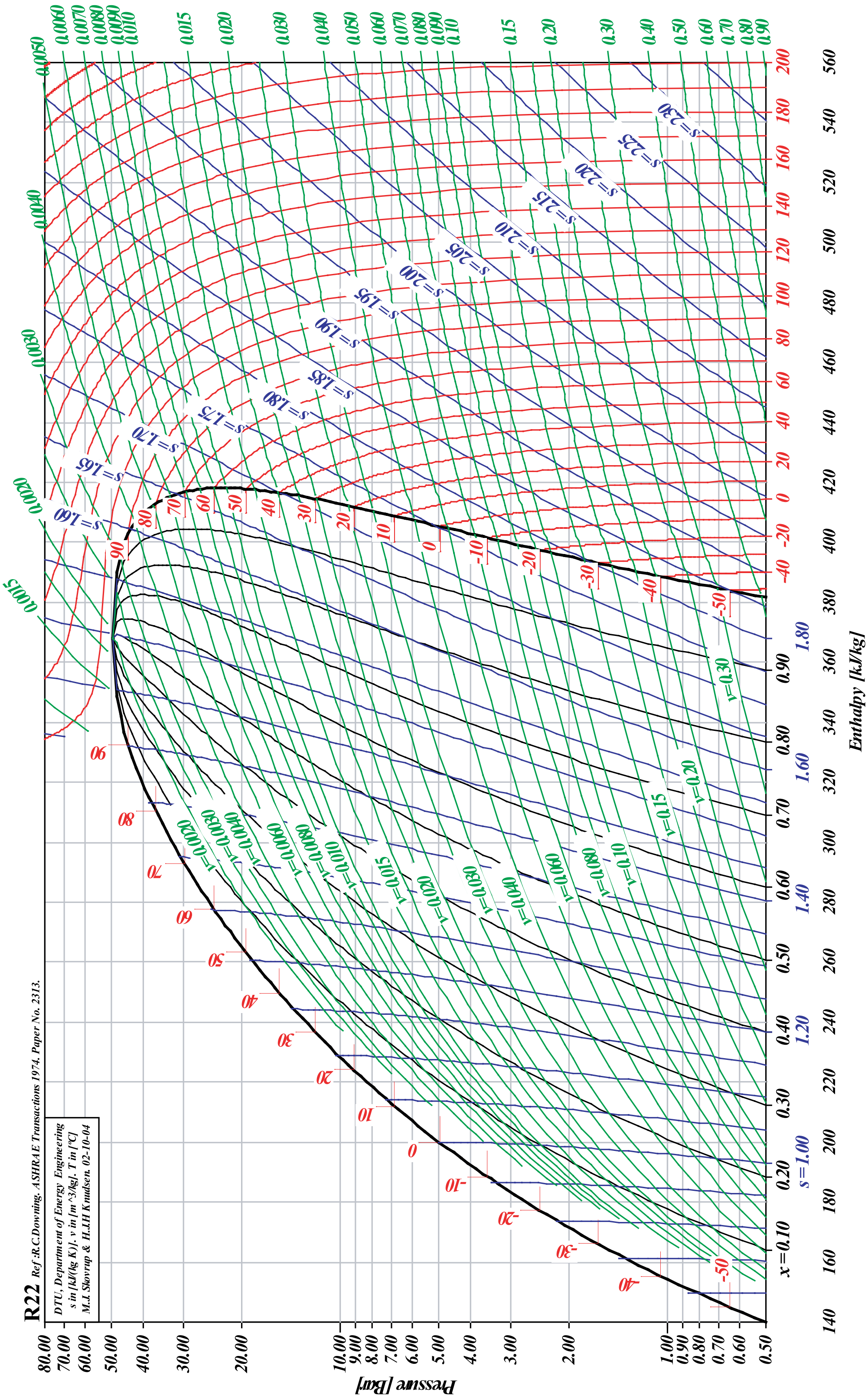
Ti = Titanium

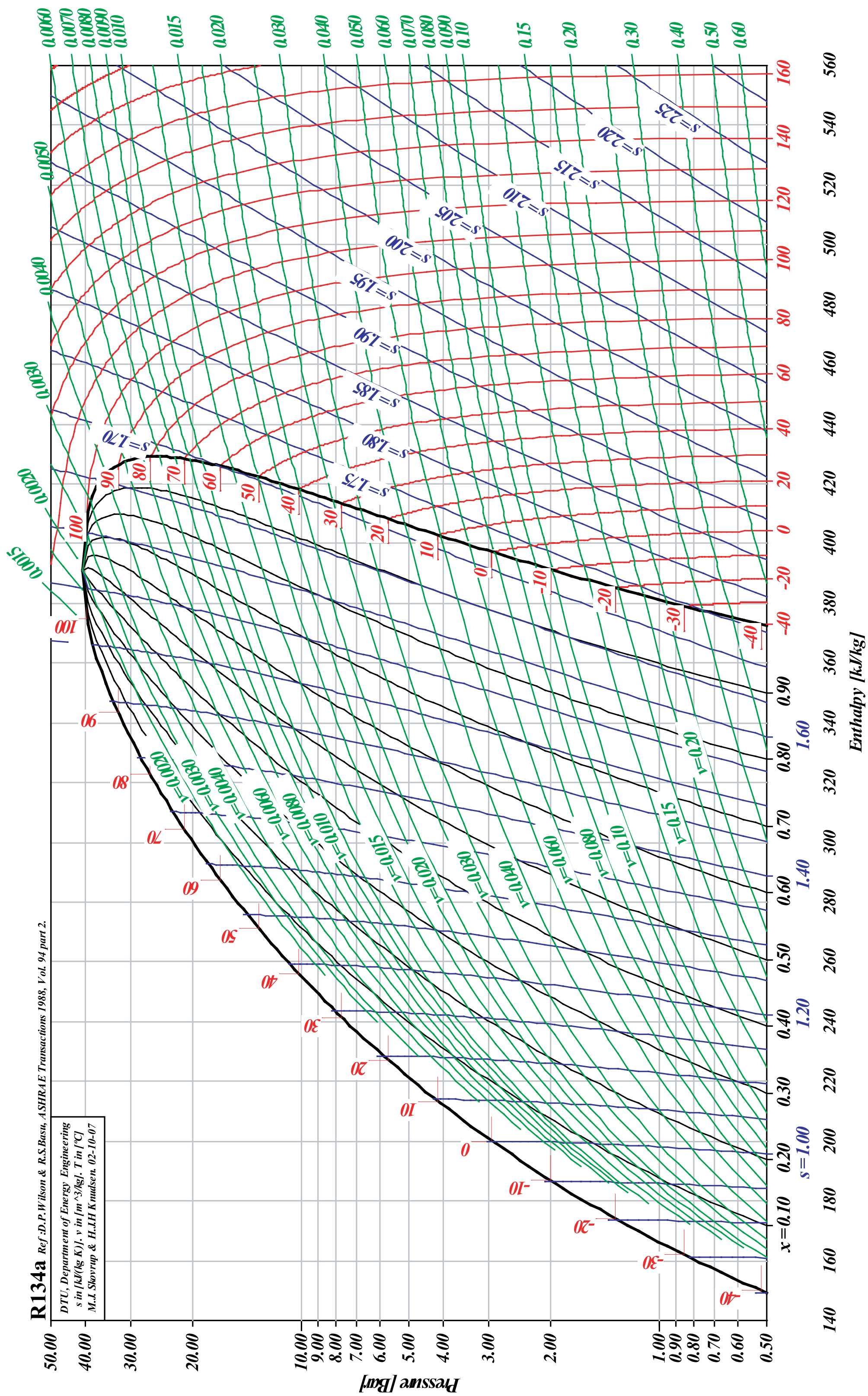
B - Log P/h diagrams for refrigerants

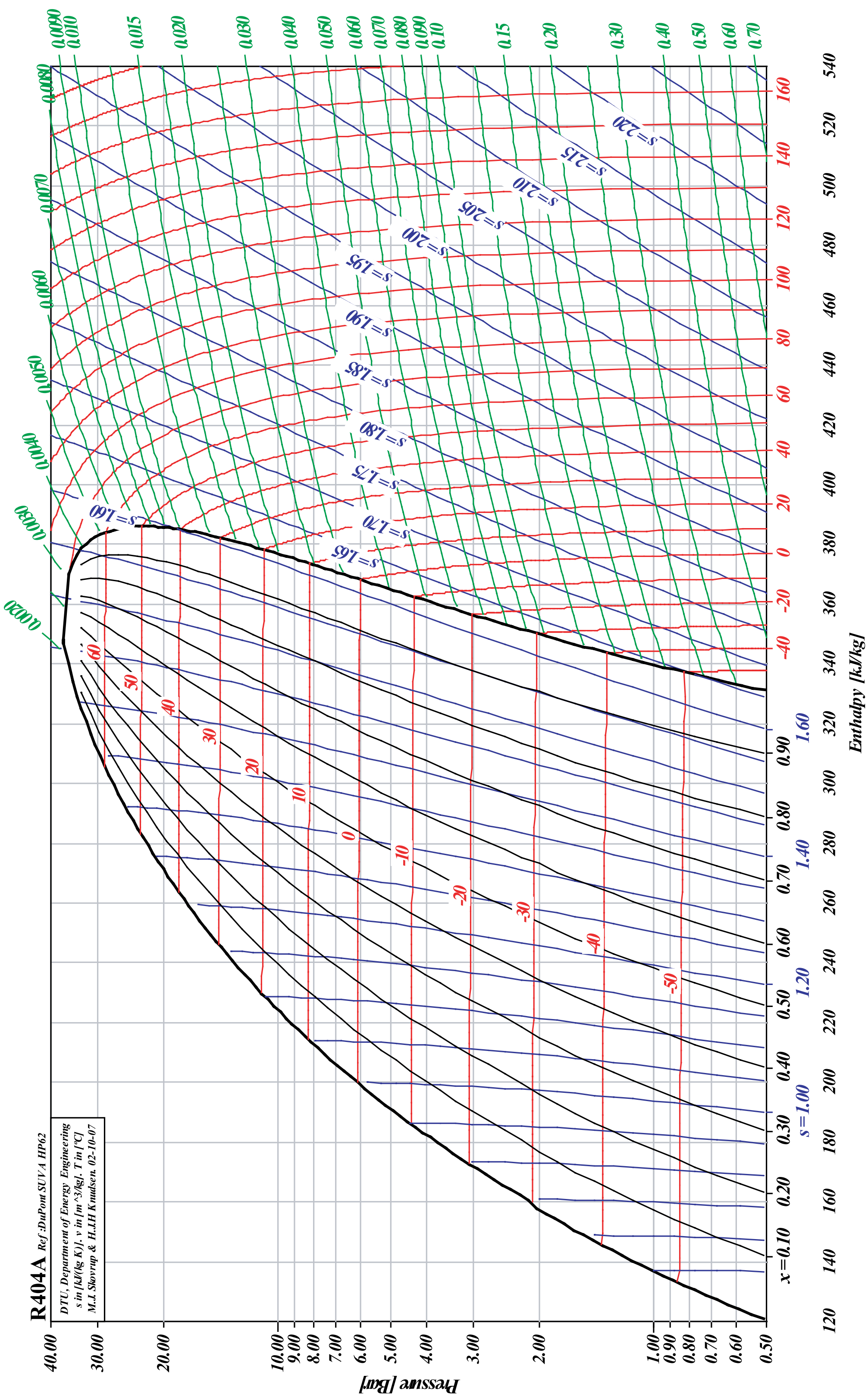
The log P/h diagrams in this section have been retrieved from the Coolpack program, which can be downloaded at the following address: http://www.et.dtu.dk/CoolPack/

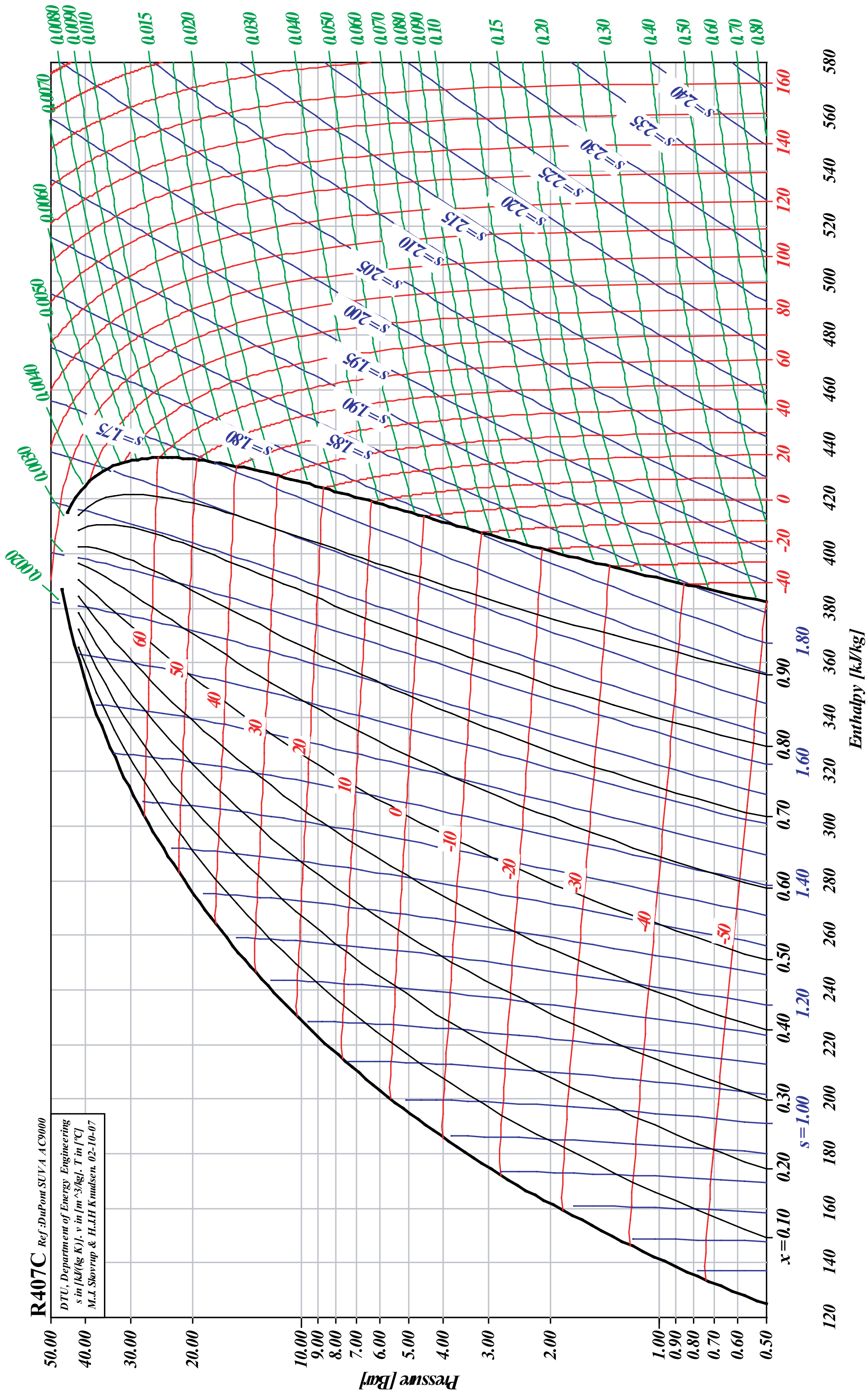
The log P/h diagram of the following refrigerants can be found:

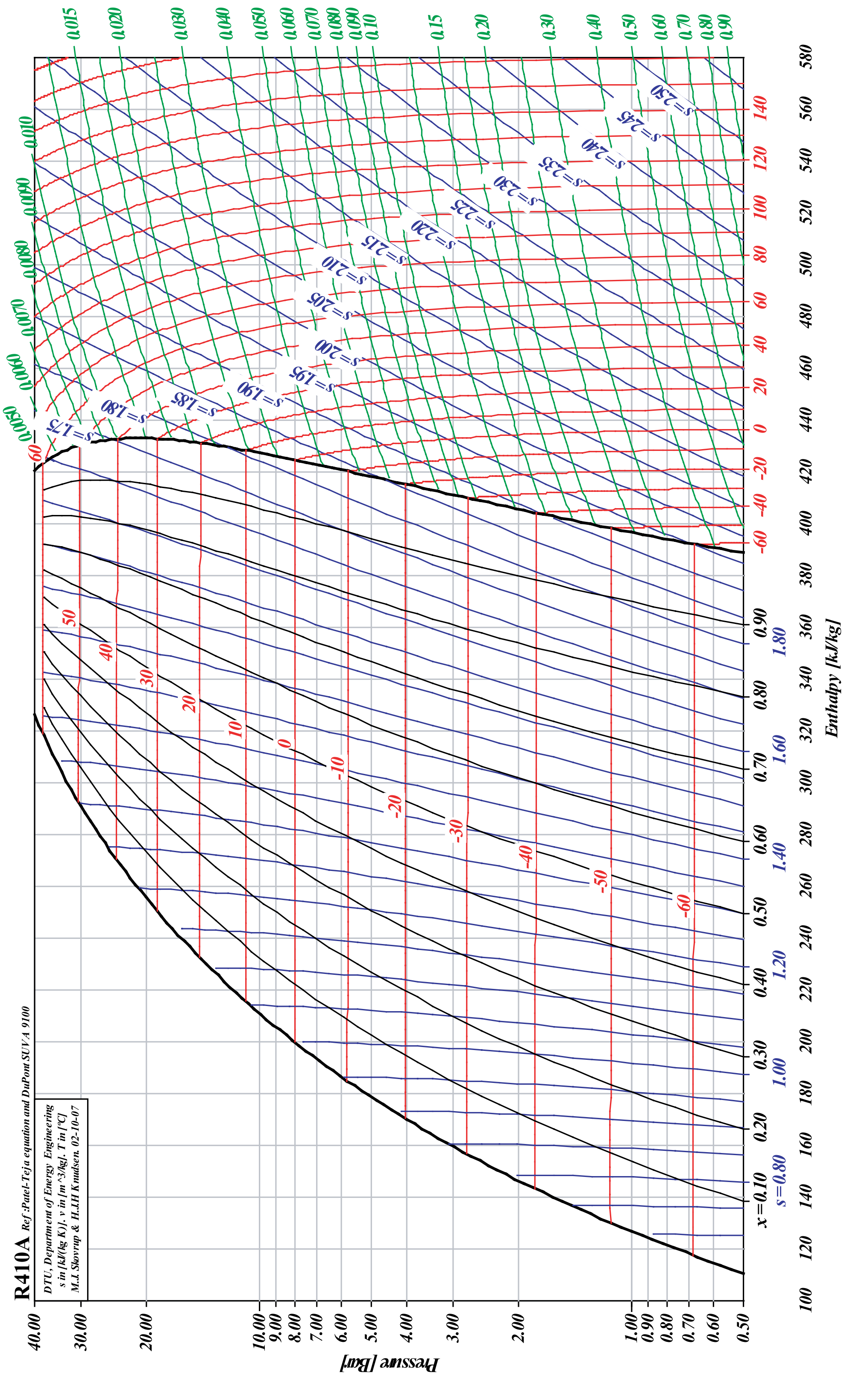
- R22
- R134a
- R404A
- R407C
- R410A
- R717 (Ammonia)
- R744 (Carbon dioxide)
- R507

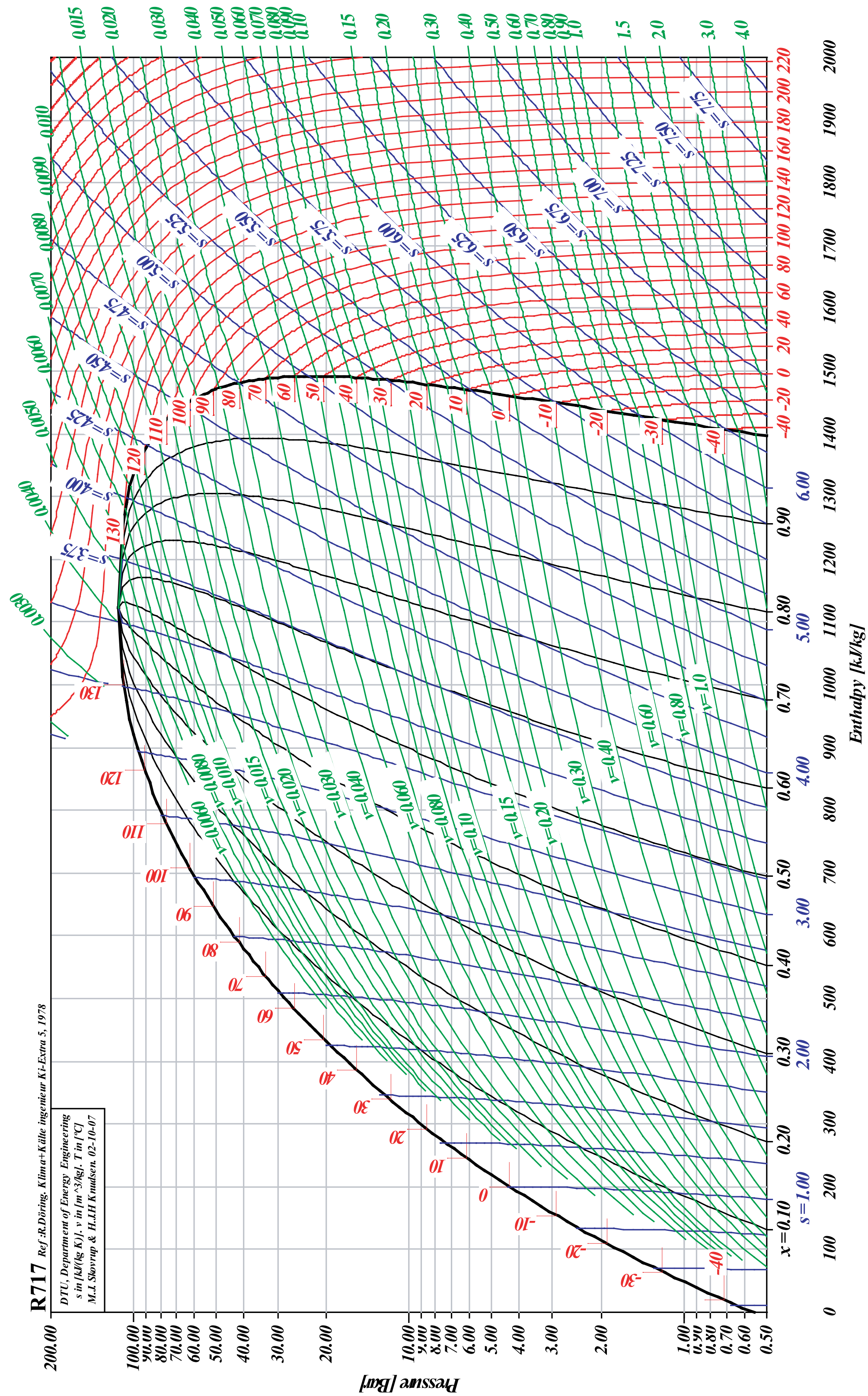

R22 Ref :R.C.Downing. ASHRAE Transactions 1974. Paper No. 2313.
DTU, Department of Energy Engineering
s in [kJ/(kg K)], v in [m^3/kg], T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-04
Pressure [Bar]
Enthalpy [kJ/kg]

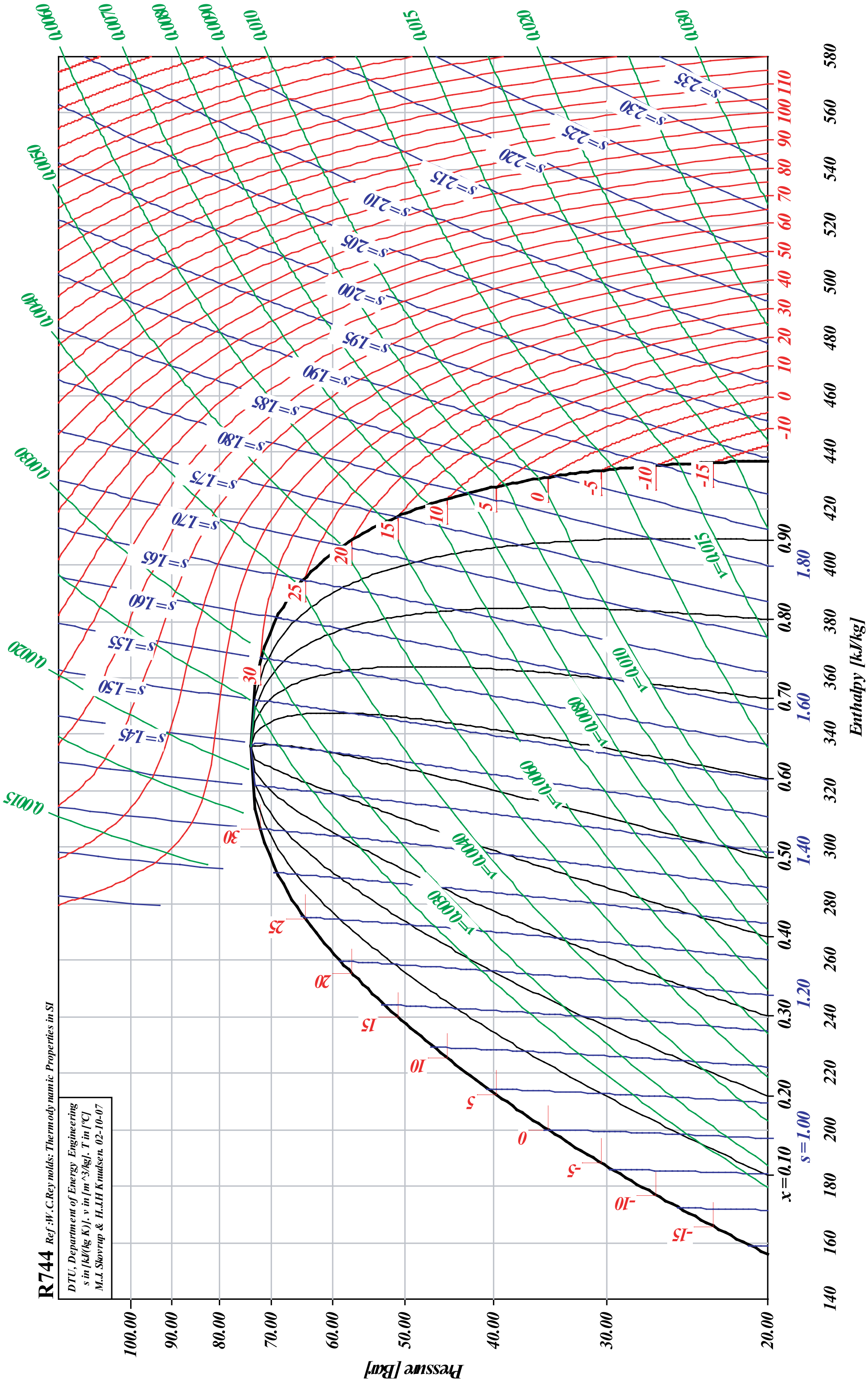

R134a Ref :D.P.Wilson & R.S.Basu, ASHRAE Transactions 1988, Vol. 94 part 2.
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Pressure [Bar]
Enthalpy [kJ/kg]


R404A Ref :DuPont SUVA HP62
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Pressure [Bar]
Enthalpy [kJ/kg]
x=0.10
s=1.00
s=1.60
s=2.20

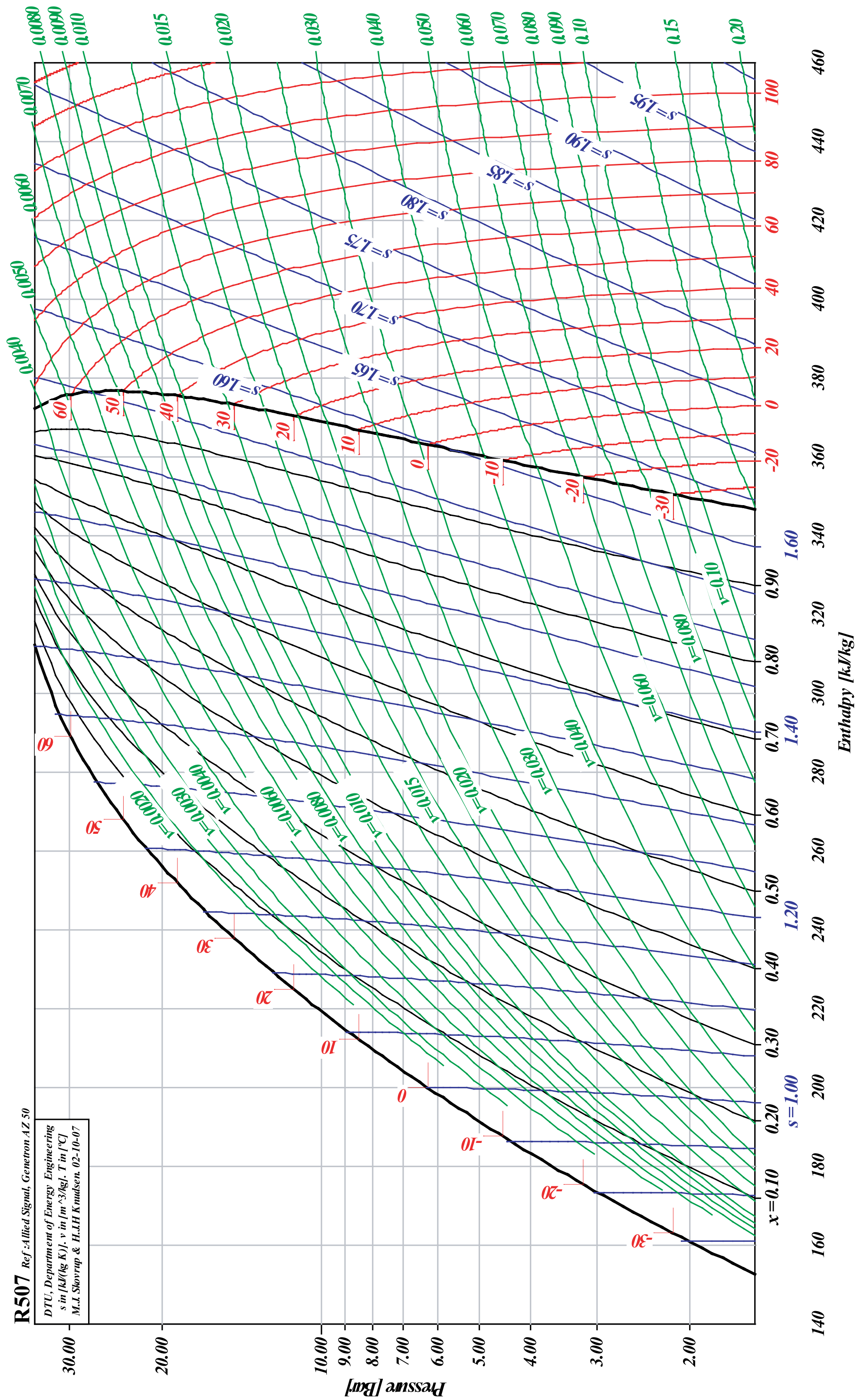

R407C Ref:DuPont SUVA AC9000
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Enthalpy [kJ/kg]
Pressure [Bar]


R410A
Ref.:Patel-Teja equation and DuPont SUVA 9100
DTU, Department of Energy Engineering
s in [kJ/(kg K)], v in [m^3/kg], T in [°C]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Enthalpy [kJ/kg]
Pressure [Bar]


R717 Ref :R.Döring, Klima+Kälte ingenieur Ki-Extra 5, 1978
DTU, Department of Energy Engineering
s in [kJ/(kg K)], v in [m^3/kg], T in [°C]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Pressure [Bar]
Enthalpy [kJ/kg]



R744 Ref.:W.C.Reynolds: Thermodynamic Properties in SI
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Pressure [Bar]
Enthalpy [kJ/kg]


R507 Ref.: Allied Signal, Genetron AZ 50
DTU, Department of Energy Engineering
s in [kJ/(kg K)]. v in [m^3/kg]. T in [ºC]
M.J. Skovrup & H.J.H Knudsen. 02-10-07
Pressure [Bar]
Enthalpy [kJ/kg]

C - Conversion Tables

Length

m	ft
1	3.281
0.3048	**1**

Area

m^2	ft^2
1	10.764
0.0929	**1**

Mass

kg	lb
1	2.205
0.4536	**1**

Volume

m^3	ft^3	US Gal	Imp Gal
1	35.32	264.2	220
$2.832 \cdot 10^{-2}$	**1**	7.481	6.229
$3.785 \cdot 10^{-3}$	0.134	**1**	0.833
$4.546 \cdot 10^{-3}$	0.161	1.201	**1**

Energy

J, Nm, Ws	kJ	kWh	kpm	kcal	ft • lbf	BTU
1	10^{-3}	$2.78 \cdot 10^{-7}$	0.102	$2.39 \cdot 10^{-4}$	0.738	$0.948 \cdot 10^{-3}$
10^3	**1**	$2.78 \cdot 10^{-4}$	102	0.239	738	0.948
$3.6 \cdot 10^6$	$3.6 \cdot 10^3$	**1**	$3.67 \cdot 10^5$	860	$2.66 \cdot 10^6$	$3.41 \cdot 10^3$
9.81	$9.81 \cdot 10^{-3}$	$2.72 \cdot 10^{-6}$	**1**	$2.34 \cdot 10^{-3}$	7.23	$9.29 \cdot 10^{-3}$
$4.19 \cdot 10^3$	4.19	$1.16 \cdot 10^{-3}$	427	**1**	$3.09 \cdot 10^3$	3.97
1.36	$1.36 \cdot 10^{-3}$	$0.377 \cdot 10^{-6}$	0.138	$0.324 \cdot 10^{-3}$	**1**	$1.29 \cdot 10^{-3}$
$1.06 \cdot 10^3$	1.06	$0.293 \cdot 10^{-3}$	108	0.252	778	**1**

Pressure

$Pa=N/m^2$	kPa	bar	mwg	kp/cm^2 = at	psi	atm
1	10^{-3}	10^{-5}	$1.02 \cdot 10^{-4}$	$1.02 \cdot 10^{-5}$	$1.45 \cdot 10^{-4}$	$9.87 \cdot 10^{-6}$
10^3	**1**	10^{-2}	0.102	$1.02 \cdot 10^{-2}$	0.145	$9.87 \cdot 10^{-3}$
10^5	10^2	**1**	10.2	1.02	14.5	0.987
$9.81 \cdot 10^3$	9.81	$9.81 \cdot 10^{-2}$	**1**	0.100	1.42	$9.68 \cdot 10^{-2}$
$9.81 \cdot 10^4$	98.1	0.981	10.0	**1**	14.2	0.968
$6.89 \cdot 10^3$	6.89	$6.89 \cdot 10^{-2}$	0.703	$7.03 \cdot 10^{-2}$	**1**	$6.80 \cdot 10^{-2}$
$1.013 \cdot 10^5$	$1.013 \cdot 10^2$	1.013	10.3	1.03	14.7	**1**

Power

W, J/s	kW	kpm/s	kcal/h	hp (metric)	hp (UK, US)	ft • lbf/s	BTU/h
1	10^{-3}	0.102	0.860	$1.36 \cdot 10^{-3}$	$1.34 \cdot 10^{-3}$	0.738	3.41
10^{3}	**1**	102	860	1.36	1.34	738	$3.41 \cdot 10^{3}$
9.81	$9.81 \cdot 10^{-3}$	**1**	8.43	$1.33 \cdot 10^{-2}$	$1.32 \cdot 10^{-2}$	7.23	33.5
1.16	$1.16 \cdot 10^{-3}$	0.119	**1**	$1.58 \cdot 10^{-3}$	$1.56 \cdot 10^{-3}$	0.858	3.97
735.5	0.7355	75	632.4	**1**	0.986	542.5	$2.51 \cdot 10^{3}$
745.7	0.7457	76.04	641.2	1.014	**1**	550	$2.54 \cdot 10^{3}$
1.36	$1.36 \cdot 10^{-3}$	0.138	1.166	$1.84 \cdot 10^{-3}$	$1.82 \cdot 10^{-3}$	**1**	4.63
0.293	$2.93 \cdot 10^{-4}$	$29.9 \cdot 10^{-3}$	0.252	$0.398 \cdot 10^{-3}$	$0.393 \cdot 10^{-3}$	0.216	**1**

Specific heat

kJ/(kg °C)	kcal/(kg °C)	BTU/(lb°F)
1	0.2388	0.2388
4.187	**1**	1
4.187	1	**1**

Latent heat

kJ/kg	kcal/kg	BTU/lb
1	0.2388	0.430
4.187	**1**	1.80
2.326	0.556	**1**

Thermal conductivity

W/(m °C)	kcal/(m °C h)	BTU/(ft°F h)
1	0.860	0.578
1.163	**1**	0.672
1.731	1.488	**1**

Heat transfer coefficient

W/(m²°C)	kcal/(m² °C h)	BTU/(ft² °F h)
1	0.860	0.176
1.163	**1**	0.205
5.677	4.881	**1**

Temperature and Temperature Difference

Quantity	Kelvin scale	Celsius scale	Fahrenheit scale	Physical fixed point
Associated	0 K	-273.15 °C	-459.67 °F	Absolute zero
Temperatures	255.3722 K	-17.7778 °C	0 °F	
	273.15 K	0 °C	32 °F	Ice fusing point
	273.16 K	0.01 °C	32.018 °F	Water triple point
	373.15 K	100 °C	212 °F	Water boiling point (1atm)
Associated	1 K	1 °C	1.8 °F	
Temperature differences	0.555 556 K = 5/9 K	0.555 556 °C = 5/9 °C	1 °F	

Glossary

A

adsorption charge
In addition to refrigerant, the bulb also contains a solid adsorbent such as charcoal or silica gel. The adsorbed refrigerant reacts more slowly to temperature changes than direct-charged bulbs, and gives a slower response.

alkalinity
The quantitative capacity of water to neutralize an acid, i.e. the measure of how much acid can be added to a liquid without causing a significant change in pH. Expressed as mg $CaCO_3$/l.

AMP
Adenosine monophosphate.

ASHRAE
American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

asymmetric operation
When the temperature change in a fluid (or the flow) on one side of the CBE is much larger than on the other side.

B

B-model
SWEP's standard compact brazed heat exchanger, made of stainless steel 316 and copper.

back pressure
Higher pressure in the system after the heat exchanger than in the heat exchanger. This pressure prevents the condensate from being drained in steam applications.

boiling point
See: bubble point.

bubble point
The temperature (pressure related) where boiling starts. 100% saturated liquid.

bulb
A pressure probe connected to the expansion valve in order to control the amount of superheating.

burnout point
At this point in the boiling process, vapor will form to such an extent that it hinders contact between the liquid refrigerant and the heat transfer area, leading to lower heat transfer efficiency.

C

CBE
Compact Brazed heat Exchanger.

centrifugal compressor
Also called a turbo compressor. Compresses the gas by accelerating it with an impeller. The velocity is then transformed to pressure.

CHP
Combined Heat and Power plant, producing both electricity and heat.

circulation number
In flooded systems, the number of times a specific volume has to pass the evaporator to be fully evaporated.

cleaning in place (CIP)
The mineral scaling and other fouling that may form in a CBE can be removed using this cleaning method.

close temperature programs
When the primary and the secondary fluid temperatures are close to each other.

CFC
ChloroFluoroCarbon

compression ratio
Ratio of condenser and evaporator pressures.

compressor
Removes refrigerant vapor formed in the evaporator and brings it to a higher pressure. The compressor is usually powered by electricity.

compressor shut-off valves
Safety devices to protect the system from excessively high or low compressor pressures, i.e. low/high evaporation temperatures. Also: service valves.

condenser
A heat exchanger turning gas into liquid.

conduction
Heat transfer through a solid material.

convection
Heat transfer by eddies in a fluid.

COP
Coefficient Of Performance. COP is a measure of efficiency, and can be calculated for the chiller, the condenser and the heat pump. COP_{ref} = THA/W (=compressor work) and COP_{hp} = THR/W. $COP_{hp} = COP_{ref} + 1$

critical heat flux
At this point, vapor will form to such an extent in the evaporator that it hinders contact between the liquid refrigerant and the heat transfer area, leading to lower heat transfer efficiency.

cryogenic
Very low-temperature applications, e.g. liquid nitrogen at -120°C.

D

detergents
Detergents are surface-active agents, substances that lower the surface tension of water.

dew point
At the dew point, all liquid has evaporated and superheating starts. The gas is 100% saturated.

direct expansion (DX) system
The basic compression cycle is a DX system. The expansion valve is mounted directly before the evaporator.

E

electronically controlled ON/OFF valves
The electronic ON/OFF valve is actually an electronically controlled solenoid valve that functions both as an expansion valve and as a solenoid valve. When functioning as an expansion valve, ON/OFF control is used.

evaporator
Heat exchanger turning liquid into gas.

EWT
Entering Water Temperature

expansion valve
A valve that maintains the pressure difference between the high-pressure and low-pressure sides in a condenser/evaporator system. See also: valves.

external pressure equalization
An expansion valve with external pressure equalization compares the bulb pressure with the suction line pressure in order to control the superheat. Recommended in connection with CBE evaporators due to the extra V-ring pressure drop.

externally equalized expansion valve
An expansion valve with external pressure equalization compares the bulb pressure with the suction line pressure in order to control the superheat. Recommended in connection with CBE evaporators due to the extra V-ring pressure drop.

F

flooded (wet) evaporator
Generic term for thermosiphons and forced-flow systems. The forced-flow system is similar to a thermosiphon, but the forced-flow system has a pump installed before the evaporator.

flooded systems
Flooded evaporation is also called wet evaporation. In flooded systems, the refrigerant enters the evaporator as 100% liquid, and the outlet flow may contain liquid. The outlet flow must not be superheated.

foaming
Droplets of refrigerant entering the oil sump, where they are immediately evaporated.

footprint
In this context, the floor area underneath a heat exchanger.

forced circulation systems
The forced-flow system is similar to a thermosiphon, but the forced-flow system has a pump installed before the evaporator.

forwarding temperature
Temperature of the hot fluid when it enters the device in question (radiator, heat exchanger)

fouling factors
Contamination factors. Used as a measure of the contamination of (amount of dirt in) a CBE.

G

glide
With reference to refrigerants, glide is a phenomenon that occurs with some mixed refrigerants due to different refrigerants boiling off at different temperatures, i.e. at constant evaporation pressure, the temperature varies.

GWP
Global Warming Potential

H

halons
Generic term for carbon compounds containing halogens (F, Cl, Br, I)

HCFC
HydroChloroFluoroCarbon

heat flux
A measure to describe the heat transfer efficiency. The unit for this measure is transferred energy per unit area (kW/m2).

hermetic compressor
Houses both motor and compressor house inside a welded shell, which provides a truly hermetic seal to the surroundings. The welded shell of a hermetic compressor does not open.

HFC
HydroFluoroCarbon

high-pressure float valve
Located on the high-pressure side of the system and in open connection to the condenser. It controls the evaporator level indirectly by maintaining a constant level of refrigerant inside the float chamber. Float valves can be found in flooded systems.

high-theta pattern
Large-angled "herringbone" pattern. The pressure drop is higher in a CBE with a high-theta pattern than in a CBE with a low-theta pattern. Increased pressure drop leads to increased turbulence in the fluid = higher heat transfer efficiency.

hunting
A phenomenon that can happen in (e.g.) steam systems if the condensate drainage is incorrect. Condensate backs-up in the heat exchanger and induces on/off regulation. This leads to wear on the control valve, the risk of cavitation, and temperature fluctuations in the secondary fluid.

hydraulic block
In a boiler, block consisting of connections to the CBE, instead of pipes, in order to save space. It contains a number of functions, such as three-way valve, bypass valve, fill-up valve and temperature sensors.

hygroscopic
Absorbs water.

I

indirect
In an indirect system, the fluid is not directly heated by the heating source. The heating source heats an intermediate fluid that heats the fluid by means of a heat exchanger.

inhibitors
Compounds that make a chemical reaction go more slowly, by obstructing the reaction.

instantaneous tap water
Hot tap water produced only when the demand occurs (no accumulation).

internally equalized expansion valve
An expansion valve with internal pressure equalization compares the bulb pressure with the pressure in the gas/liquid line before the evaporator inlet in order to control the superheat. NOT recommended in connection with CBE evaporators due to the extra V-ring pressure drop.

isentropic efficiency
Isentropic efficiency = actual work/isentropic work. Isentropic work = no heat losses to the surroundings, no heat gained from the surroundings, no change in entropy during the work process.

L

lag time
The time a process takes to react to various changes.

latent energy
The energy that is absorbed or rejected when a phase change occurs between two states of aggregation (gas/liquid/solid).

latent heat
The heat that is absorbed or rejected when a phase change occurs between two states of aggregation (gas/liquid/solid).

Legionella
Bacteria that develop in stagnant hot water and cause the deadly Legionnaires' disease.

liquid cooler
Here, a CBE being used to cool liquid.

liquid hammer
In compressors, if a considerable amount of liquid enters the compressor house, a very large pressure can be built up when the piston reaches its top position. This phenomenon is called liquid hammer, and may cause severe damage to the valves or crankshaft.

liquid heat capacity
Amount of energy needed to increase the temperature of a specific liquid 1 Kelvin.

liquid line
That part of the refrigeration cycle where the refrigerant is in the liquid phase (after the condenser, before the evaporator).

Liquid line filter driers
Positioned in the liquid line to protect the expansion valve from particle contamination and to absorb potential humidity in the refrigerant. N.B. Creates additional pressure drop.

liquid static head
A pressure head created by liquid.

liquid-charged bulb
A liquid-charged bulb has a large charge of refrigerant and will never "run dry". It will always contain both liquid and gaseous refrigerant (see: bulb).

LMTD
Logarithmic Mean Temperature Difference.

low-pressure float valve
Controls the liquid level in the receiver, and is normally mounted in a chamber parallel to the liquid/vapor separator. Float valves can be found in flooded systems.

low-theta pattern
Small-angled "herringbone" pattern. The pressure drop is lower in a CBE with a low-theta pattern than in a CBE with a high-theta pattern. Reduced pressure drop leads to reduced turbulence in the fluid = lower heat transfer efficiency.

LWT
Leaving Water Temperature.

M

minimum stable signal (MSS)
Thermal expansion valves must always operate with a minimal working superheating to achieve stable regulation. The minimum stable signal (MSS) is sent to the TEV at the minimum superheating.

modulating electronic expansion valves
Modulating electronic expansion valves are controlled by temperature or pressure sensors.

MOP bulb
An MOP (Maximum Operating Pressure) bulb, also called a gas-charged bulb, has a much smaller quantity of refrigerant mixture inside the bulb than a liquid-charged bulb (see: bulb).

multi-pass
The fluid passes through more than one channel length before leaving the CBE.

N

nominal operation point
The most usual operating case. The system design case.

NTU
Number of heat Transfer Units, an expression for the thermal length of a plate.

O

ODP
Ozone Depletion Potential

oil filters
Often installed in the oil return line between the oil separator and the compressor to protect the compressor from contaminants. If this filter is blocked, it can lead to excessive oil carry-over, which affects heat transfer in the evaporator.

oil separator
Separates oil from refrigerant and returns it to the compressor.

open compressor
The motor and compressor house are mounted without a shell, although the shaft has a seal.

opening superheating
The additional superheating (in addition to the static superheating) required to open the expansion valve for operation.

operating point
In a refrigerant system: the equilibrium point, where the performance of the evaporator matches the performance of the compressor.

oversurfacing
The overdimensioning of a heat exchanger as a percentage. A heat exchanger with oversurface is larger than is necessary to accomplish the thermal duty.

P

pAlk
The negative logarithm of the alkalinity, expressed as mg $CaCO_3$ /L.

pCa
The negative logarithm of the calcium concentration, expressed as mg $CaCO_3$ /L.

pH
The pH of a solution is the negative common logarithm of the hydrogen ion activity, aH^+: $pH = -\log_{10}(aH^+)$

pinch
The minimum temperature difference between the refrigerant and the secondary fluid in a counter-current condenser.

positive displacement compressors
Compressors where a certain volume of gas is trapped in a space that is continuously reduced by the compressing device (piston, scroll, screw or similar) inside the compressor.

pressure gauges
Permanently installed to monitor compressor suction and discharge pressures.

primary circuit
Intermediate fluid transporting the heat from the heating source to the fluid required to be heated (in heating systems).

pS
The negative logarithm of the content of solid particles in a fluid.

PV
Photovoltaic. A photovoltaic panel converts sunlight into electricity.

R

Radiation
No medium need exist between the two bodies for heat transfer to take place (cf. conduction and convection). In radiation, the intermediaries are photons that travel at the speed of light.

receiver
Positioned after the condenser. Compensates for varying refrigerant flow in evaporator and condenser systems. Makes subcooling impossible.

reciprocating compressors
Another name for piston compressors.

recirculation type evaporator
Flooded evaporator.

refrigerant
A fluid used in cooling systems, usually with a low boiling point at atmospheric pressure.

S

saturation point
A state where a phase change may occur without a change in temperature and pressure.

scaling
A fouling phenomenon: the inorganic salts dissolved in water precipitate and form a scale on a surface.

scroll compressors
These capture the gas in the volume formed between one fixed and one orbiting scroll.

secondary distribution loop
Safety circuit between corrosive/explosive, etc., fluids and the primary circuit. In the event of leakage, the hazardous fluid is confined in an area where it will not do any harm.

secondary heat exchanger
Transfers the heat from the primary to the secondary circuit.

semi-hermetic compressor
The motor and the compressor house are located in a two-piece shell. The covers are bolted together.

semi-instantaneous tap water
Hot water accumulated in a tank to ensure a short response time when a demand occurs.

sensible energy
The energy added to or rejected from a gas, liquid or solid without a phase change.

sensible heat
The heat absorbed by or rejected from a gas or liquid without any phase change.

sight glass
Used for inspection of the refrigerant flow before the expansion valve.

single-pass
A CBE where the fluids pass through the channels only once.

single-screw compressor
A configuration of a screw compressor consisting of a single screw rotor.

solenoid valve
The solenoid valve will maintain the pressure difference between the condenser and the evaporator sides during off periods and thereby prevent liquid from flowing into the evaporator.

SSP
The SWEP Software Package.

stalling point
Condition where the steam pressure inside the heat exchanger is equal to the back pressure in the condensate system.

static head
Liquid column in a vertical pipe.

static superheating
The minimum level of superheating that is required to allow the pressure from the bulb to start pushing back the spring and thus opening the expansion valve.

sub-cooled liquid
When saturated liquid is cooled at constant pressure, its temperature decreases and it becomes subcooled.

substation
Installation transferring heat from the district heating/cooling network (primary system) to the building network (secondary system).

suction line
The part of the refrigerant cycle that is located between the evaporator and the compressor. The refrigerant is usually in the state of superheated refrigerant here.

suction line accumulators
Installed to avoid liquid carry-over to the compressor, which could affect lubrication and, in the worst-case scenario, lead to damage if liquid or oil foam enters the compression chamber.

suction line filters
Protects the compressor from particle contamination and absorbs potential humidity in the refrigerant. N.B. induces additional pressure drop. The evaporation pressure must be measured before the filter.

superheated gas
When a dry saturated gas is heated at constant pressure, its temperature rises and it becomes superheated.

surfactants
Any substance that will reduce water's surface tension is called a surface-active agent, or surfactant. Using the analogy of the thin, elastic surface membrane, a surfactant cuts that membrane to ribbons.

system hunting
On-off regulation. See: hunting.

T

tandem compressors
Two or more compressor working in parallel.

TDS
Total Dissolved Solids

TEV
Thermal Expansion Valve.

TEWI
Total Environmental Warming Impact

THA
Total Heat of Absorption.

thermosiphon systems
A flooded system where no pump is needed.

THR
Total Heat of Rejection.

triplet compressors
Three compressors in parallel (also called trio).

twin-screw compressor
Consists of two rotors with matching profiles. The rotor profiles are designed to decrease the volume between them continuously from the inlet to the outlet of the compressor.

two-pass over one-pass
A CBE where one of the fluids passes through the channels only once but the other passes through the channels twice.

two-phase static head
A pressure head created by a liquid/gas mixture.

V

V-ring
A SWEP distribution device, securing a good distribution of gas and liquid in SWEP evaporators.

volumetric efficiency
The ratio between the actual vapor volume and the maximum volume that can be theoretically contained in the compressor cylinder.

W

water hammer
Water propelled in the heat exchanger with a high velocity, which causes damage.

water hammer arrester
Absorbs unstable pressure due to sudden closing of gate, and moderates the impact of the fluid inside the pipe.

water heater
Provides only hot tap water (cf. boiler, which provides hot tap water and space heating).

working superheating
The real superheating that can be measured in the system. Working superheating = opening superheating + static superheating.