

## Chapter 2

# Types of Compression Heat Pumps and Their Main Components

**Abstract** The main components of compression heat pumps are treated herein. Their salient features, working principles and roles are dealt with, stressing their contribution to heat pumps operation and efficiency. Furthermore, products existing on the market are often referred to in order to allow the interested reader to have, at least, a rough idea of the available equipment, nowadays. Engine driven heat pumps, also named Gas Heat Pumps (GHP) are described in addition to the most commonly used electric heat pumps (EHP). Except for the driving motor, GHPs differ from EHPs both from the thermodynamic point of view, as they interact with three heat sources (they are a three-thermal-system), and for the possibility of using heat recovered by engine cooling. Last but not least, part of this chapter is devoted to describe CO<sub>2</sub> heat pumps, due their peculiarity. In fact carbon dioxide has a very low critical temperature and, thus, they operate in hyper-critical condensations in most cases. Due to this a gas cooler is employed instead of a classical condenser.

### 2.1 Main Components of Compression Heat Pumps

From the scheme we have referred to so far, it clearly comes out that the main components of compression heat pumps are:

- the compressor, that keeps the right pressure drop between evaporator and condenser to maintain the proper phase change temperatures to interact with the external (to the HP) sources;
- the expansion valve, irreversibly taking the refrigerant from the condenser pressure to the one of the evaporator;
- the condenser, where the superheated vapor coming from the compressor is de-superheated, first, then condensed to liquid with some degree of sub cooling to prevent vapor from entering the expansion valve;
- the evaporator, where the mixture coming from the expansion device vaporizes. The exiting vapor can be either saturated (wet evaporator) or superheated (dry evaporator). In the former case a proper device (separator) is needed to prevent liquid from entering the compressor. In the latter case the vapor leaving the evaporator has a superheat of few degrees Celsius for the same purpose.

## 2.2 Compressor

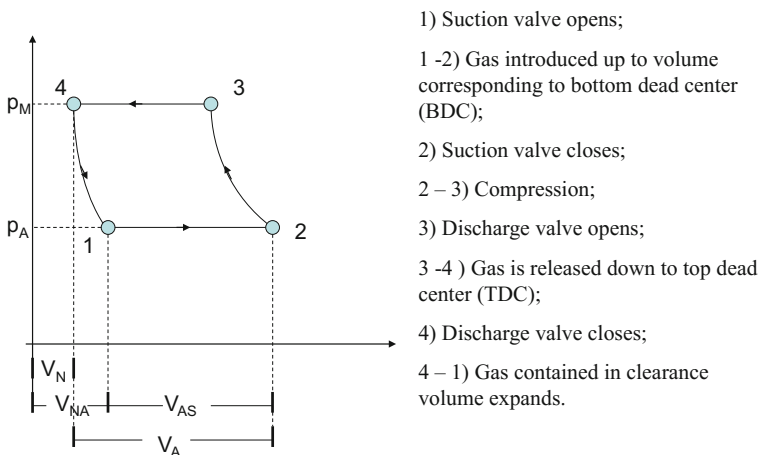
Heat pumps mainly adopt volumetric compressors. They may be both reciprocating and rotary compressors. We will refer to the former ones to describe the main features of this type of device.

### 2.2.1 Reciprocating Compressor and Basic Concepts

Those we are referring in the following consist of cylinders where the vapor coming from the evaporator is sucked due to piston motion, and then released to the condenser. Figure 2.1 show the ideal cycle followed by a reciprocating compressor in the pressure volume plane.

As well known, piston movement does not cover the whole cylinder volume. When it reaches the highest possible position, commonly referred to as top dead center, both suction and discharge valves are closed and discharge pressure,  $p_M$ , is reached. At this point, a volume is left as no further compression is allowed due to the valve plate, named clearance volume,  $V_N$  ( $V_4$  in Fig. 2.1). The smaller this volume, the more efficient the compression is.

If  $V_C$  ( $V_3$  in Fig. 2.1) is the cylinder volume at the end of compression, the volume of the fluid discharged, after the discharge valve opening, is  $V_M = V_C - V_N$ . Then, the above valve closes and the re-expansion process of the clearance vapor occurs down to pressure  $p_A$  (point 1 in the figure). The new volume  $V_{NA}$  at pressure  $p_A$  corresponds to the volume occupied by the clearance vapor  $V_N$  at the discharge pressure.



**Fig. 2.1** Reversible cycle of a reciprocating compressor

The suction valve opens at pressure  $p_A$  (BTD bottom dead center) and vapor enters the compressor. The volume  $V_A = V_2 - V_N$  is the theoretical volume that could be sucked and  $V_{AS} = V_2 - V_{AN} = V_A - (V_{AN} - V_N)$  is the actual volume sucked by the compressor. The ratio  $V_{AS}/V_A$  is named the compressor volumetric efficiency.

*Example 2.1* Let us consider the isentropic compression of an ideal gas, in a cylinder with a clearance volume  $V_N$ . Let us identify with  $V_A$  the available volume at suction and with  $p_A$ ,  $T_A$ ,  $p_B$ , and  $T_B$  respectively the pressures and temperatures (K) at the suction and discharge points.

The number of moles,  $n_N$ , contained in the clearance volume is

$$n_N = \frac{p_M V_N}{RT_M}$$

At suction, in the absence of clearance volume the numbers of moles that could be sucked would be:

$$n_A = \frac{p_A V_A}{RT_A}$$

Actually (in the presence of clearance volume) we can suck a number of moles,  $n_{AS}$ , equal to the difference between these latter diminished by the number of moles contained in the clearance volume.

$$n_{AS} = n_A - n_N = \frac{p_A V_A}{RT_A} - \frac{p_A V_{NA}}{RT_A} = \frac{p_A V_{AS}}{RT_A}$$

The volumetric efficiency,  $\eta_v$ , of the compressor is:

$$\eta_v = \frac{V_{AS}}{V_A} = 1 + \frac{V_N}{V_A} (1 - \beta^{1/k})$$

where  $\beta = p_M/p_A$  is the barometric compression ratio (simply called compression ratio) and  $k$  is the ratio between the gas specific heats or, more in general the polytrophic exponent. The clearance volume commonly varies between 2 and 5% of  $V_A$ . Thus, if we assume a value of 5% and  $k = 1.4$ , values can be calculated by the following formula:

$$\eta_v = 1 + 0.05(1 - \beta^{0.714})$$

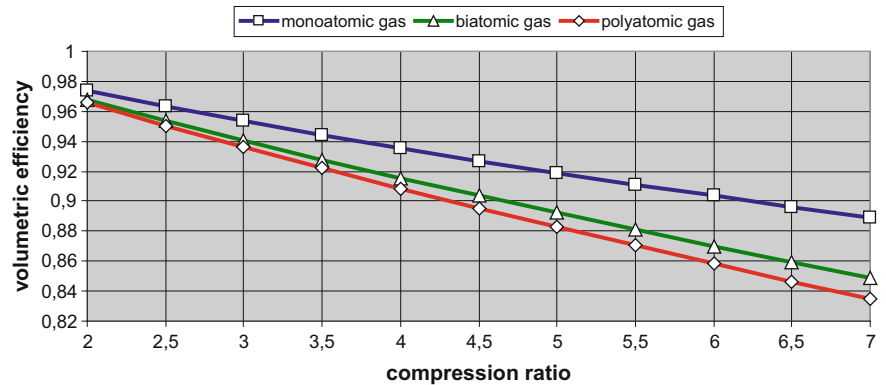
| $\beta$ | $\eta_v$ |
|---------|----------|
| 2       | 0.97     |
| 3       | 0.94     |
| 4       | 0.92     |
| 5       | 0.89     |
| 6       | 0.87     |
| 7       | 0.85     |
| 8       | 0.83     |

The volumetric efficiency reduction due to increasing of the pressure ratio can be easily explained as follows. The larger the compression ratio, the larger is the gas volume,  $V_{NA}$ , after the expansion of the clearance volume. Consequently  $V_{AS}$  decreases, as the mass sucked and then compressed during a cycle is given by  $\rho_2 V_{AS} = \eta_v \rho_2 V_A$  and the mass flow rate by  $m = n_{cy} \eta_v \rho_2 V_A$ , if  $n_{cy}$  is the number of cycles per second.

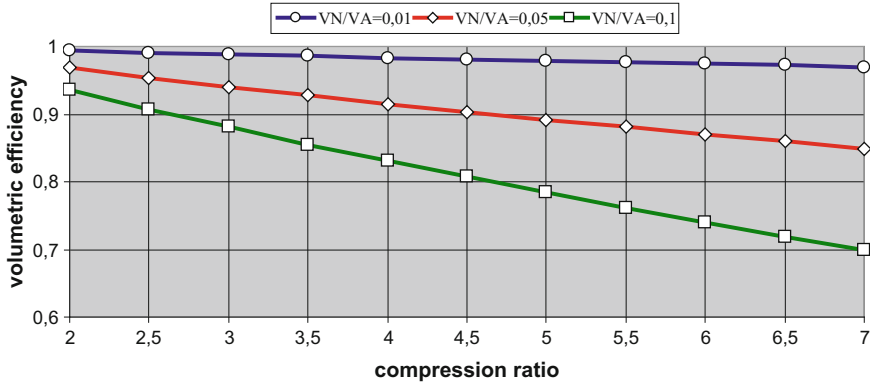
The following figures show the trend of volumetric efficiency versus compression ratio for monatomic, diatomic and polyatomic ideal gases (Fig. 2.2) with  $V_N/V_A = 0.05$ , and (Fig. 2.3) for a diatomic gas with different values of  $V_{AS}/V_A$ . It is, therefore, clear how both gas structure (even if ideal) and clearance volume affect volumetric efficiency.

In reality, several phenomena have to be accounted for, causing irreversibilities. Among them we recall friction losses in the mechanical compressor’s components, heat losses along the compression (that we have previously assumed adiabatic), fluid friction losses and fluid leaks towards the suction valve through some seal flaws between the rotating (or alternating) and fixed parts of the casing.

A very significant role is played by pressure losses in suction and discharge valves. They lower the suction pressure taking this to  $p_2'$  instead of the one required



**Fig. 2.2** Volumetric efficiency of an ideal gas (monatomic, diatomic and polyatomic) versus compression ratio



**Fig. 2.3** Volumetric efficiency versus compression ratio for various  $V_N/V_A$

by the heat exchange with the cold source  $p_2 = p_A$ , and cause the discharge pressure to be increased (to a value  $p_3'$ ) instead of the one required by the heat exchange with the hot source,  $p_3 = p_M$ .

There exist some other reasons (see also the screw and scroll compressors) causing a difference between the pressures actually imposed by compressor and those imposed by external thermal sources. The actual pressure ratio occurring in the compressor,  $p_3'/p_2' = \beta'$ , is called the internal compressor ratio and may be different from the previously defined  $\beta = p_3/p_2$ , also named the external compressor ratio. The above mentioned phenomena contribute to modify the volumetric efficiency and the work achievable.

All this leads to introduce the isentropic efficiency,  $\rho_c$ , defined as the ratio of the ideal enthalpy difference between discharge and suction,  $\Delta h$ , and the actual one,  $\Delta h'$  (Fig. 2.4).

$$\rho_c = \frac{\Delta h}{\Delta h'} = \frac{h_3 - h_2}{(h_{3'} - h_{3M}) + (h_{3M} - h_{2A}) + (h_{2A} - h_2)}$$

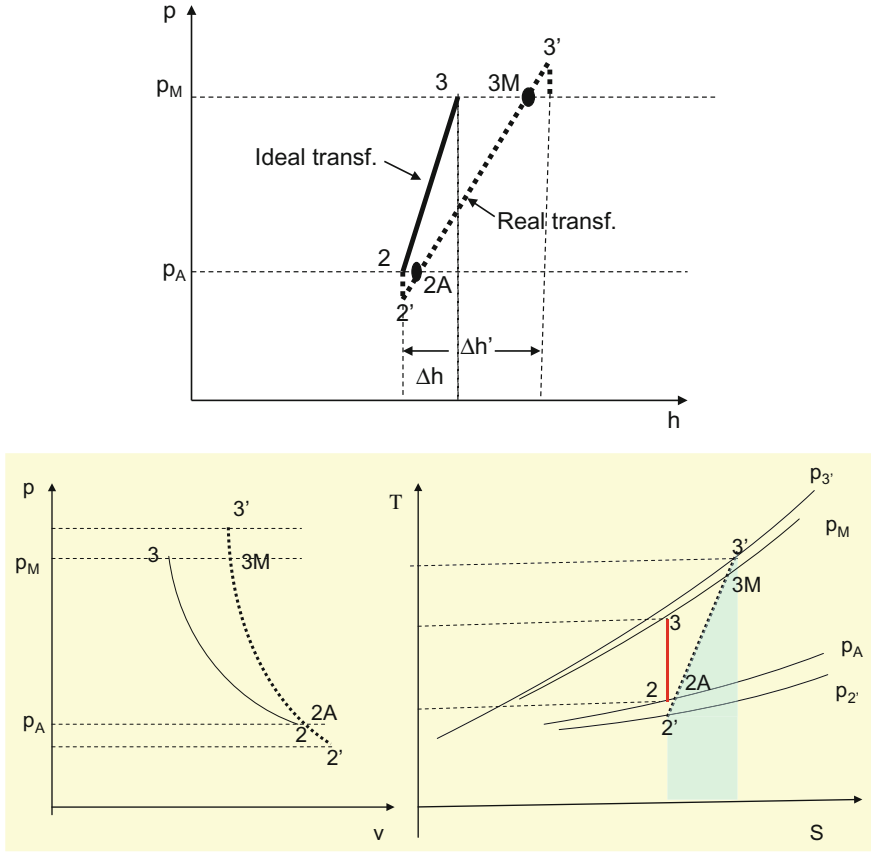
In the case where external compression ratio is equal to the internal one, the work supplied to compressor in the presence of friction ( $l_a$ ) is given by:

$$l = - \int_2^{3M} v dp - l_a = c_p(T_2 - T_{3M}) = c_p T_2 \left[ 1 - \left( \frac{p_M}{p_A} \right)^{\frac{\pi-1}{\pi}} \right]$$

Points 3 M and 2A respectively are at the same pressures as 3 and 2,  $\pi$  is the exponent of the real adiabatic transformation, and the work is negative as supplied to the system and  $T$  the absolute temperature (K).

Recalling that, on the polytropic 2'-3'<sup>1</sup>

<sup>1</sup>Remember we are referring to an equivalent reversible transformation.



**Fig. 2.4** Ideal and real compression in the planes p,h, p,v and T,S

$$\int_2^{3M} T ds = l_a$$

We can display friction losses on the plane T, s as the area underneath the curve 2–3 M. Area 2–3–3 M represents the energy related to the compressed gas heating up:

$$Area(2 - 3 - 3M) = \int_2^{3M} T ds - l_a = c_p(T_{3M} - T_2) - l_a$$

This phenomenon is called “thermal recovery”.

**Table 2.1** Values of  $k = c_p/c_v$  for some refrigerants

| Fluid           | k    | T (°C) | p (bar) |
|-----------------|------|--------|---------|
| NH <sub>3</sub> | 1.31 | 0      | 1       |
| CO <sub>2</sub> | 1.29 | 27     | 1       |
| R134a           | 1.11 | 30     | 1       |
| R437            | 1.15 | 25     | 1       |

To better describe the compressor technical features, a hydraulic efficiency is defined as:

$$\begin{aligned}\rho_y &= \frac{l + l_a}{l} = \frac{-\int_2^{3M} v dp}{c_p T_2 \left[ 1 - \left( \frac{p_M}{p_A} \right)^{\frac{\pi-1}{\pi}} \right]} = \frac{\frac{\pi}{\pi-1} p_A v_2 \left[ 1 - \left( \frac{p_M}{p_A} \right)^{\frac{\pi-1}{\pi}} \right]}{\frac{k}{k-1} p_A v_2 \left[ 1 - \left( \frac{p_M}{p_A} \right)^{\frac{\pi-1}{\pi}} \right]} \\ &= \frac{\frac{\pi}{\pi-1}}{\frac{k}{k-1}} = \frac{\pi k - 1}{k \pi - 1}\end{aligned}$$

Such a parameter does not depend on compression ratio. Once the hydraulic or polytropic efficiency<sup>2</sup> is known, the exponent of the polytropic curve can be obtained and viceversa. Therefore the isentropic efficiency can be written as:

$$\rho_c = \frac{h_2 - h_3}{h_2 - h_{3M}} = \frac{T_3 - T_2}{T_{3M} - T_2} = \frac{\beta^{\frac{k-1}{k}} - 1}{\beta^{\frac{1-k}{k}} - 1}; \quad \beta = \frac{p_M}{p_A}$$

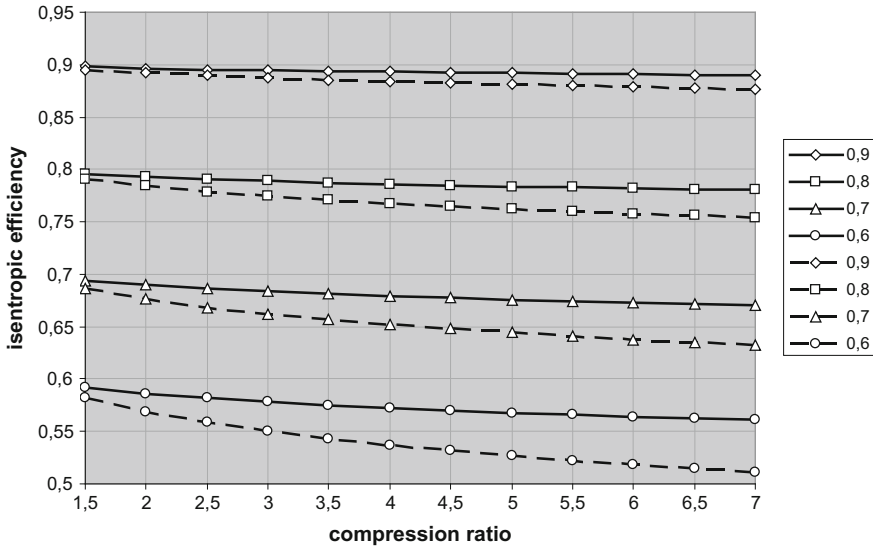
It decreases with the compression ratio and depends on the type of fluid through  $k$ , ratio between the specific heats at constant pressure and volume. After recalling that  $k = 1.4$  for standard air, some values of this parameter are given in Table 2.1 for four gas used in heat pumps.

Figure 2.5 shows the theoretical trend (calculated here in) of the isentropic efficiency versus the compression ratio at constant values of the hydraulic efficiency, with  $k = 1.11$  (continuous line), and  $k = 1.29$  (dotted line).

As already said, the pressure existing in the compressor, both at suction and discharge, is not the same as the one present in the circuit just before the suction and after the discharge. Therefore we introduced two compression ratios the internal,  $\beta'$ , and the external,  $\beta$ , ones.

Three different cases can occur, for reciprocating, screw and scroll compressors, as follows.

<sup>2</sup>The name “hydraulic efficiency” refers to the fact that the thermal recovery is negligible in the hydraulic machines, so that this efficiency is equal to 1. This parameter is also called “the polytropic efficiency” because a reference reversible polytropic is usually considered, with an average exponent equal to that of the actual transformation.



**Fig. 2.5** Isentropic efficiency versus compression ratio for  $k = 1.11$  (continuous line) and  $k = 1.29$  (dotted line) and for several hydraulic efficiencies, as indicated in the legend

- $\beta = \beta'$ —the discharge opening opens exactly when the refrigerant pressure (at the compressor exit) matches the one of the discharge line and the gas is immediately sent to it. This event very seldom occurs.
- $\beta' < \beta$ —the compressed gas has not yet reached the pressure of discharge line, sub-compression. This causes a sudden refrigerant flow towards the compressor, with an abrupt pressure increase, over compression. After that gas is expelled to the circuit.
- $\beta' > \beta$ —the compressed refrigerant pressure is larger than the one of the discharge line. This originates a sudden flow leaving the compressor.

In both the last two cases some uncontrolled expansions occur and introduce additional losses. The most critical is the over compression, as gas expands within a larger volume while discharging.

The trend of the isentropic efficiency versus the pressure ratio is qualitatively depicted in Fig. 2.6, and a peak of its value is easily identifiable. Figure 2.7 shows these trends for different types of compressors. They basically have the same shape even if some peculiar behavior occurs, depending on the type of compressor, in particular for screw machines.<sup>3</sup> The above graphs show how the isentropic

<sup>3</sup>A screw compressor can be optimized so that its isentropic efficiency is maximized in correspondence to a given value of compression ratio. Thus the maximum of the curve coincides with this ratio. During operation compression ratio can change (e.g., at partial load) and the curve can shift either leftward or rightward. To restore the optimum, compression ratio should be increased or decreased correspondingly.



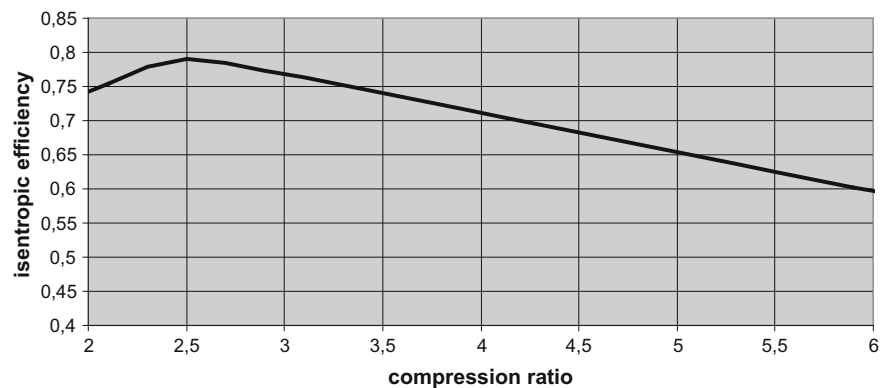
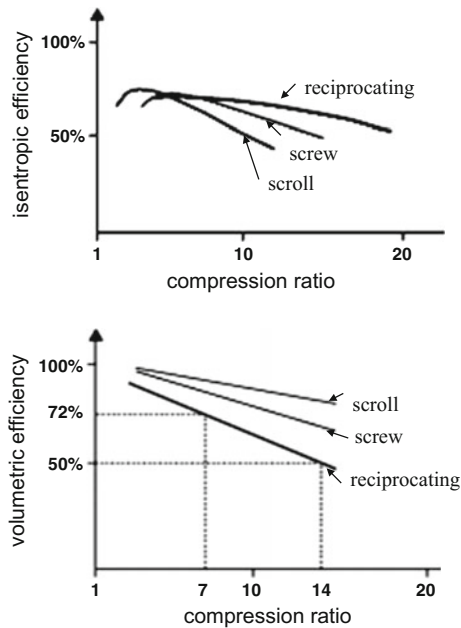


Fig. 2.6 Typical trend of a compressor isentropic efficiency

Fig. 2.7 Typical trends of isentropic efficiency (*top graph*) and of volumetric efficiency (*lower graph*) versus compression ratio for reciprocating, screw and scroll compressors



efficiency could also increase with a reduction of the compression ratio. Furthermore the typical trends of the volumetric efficiency are displayed in the same figure.

The thermodynamic cycle irreversibilities play a different role on heat pumps performances in winter and in summer. In fact, in winter, the useful effect (i.e., the useful heating output)  $h_2 - h_3$ , increases, due to the increase of the compression work,  $h_2' - h_2$ . Thus the COP changes as below:

*ideal case*

$$COP_{id} = \frac{h_2 - h_3}{h_2 - h_1}$$

*real case*

$$\begin{aligned} COP_{re} &= \frac{h_{2'} - h_3}{h_{2'} - h_1} = \rho_c \frac{(h_2 - h_3) + (h_{2'} - h_2)}{(h_2 - h_1)} \\ &= \rho_c COP_{id} + \rho_c \frac{(h_{2'} - h_1) - (h_2 - h_1)}{(h_2 - h_1)} \\ &= \rho_c COP_{id} + 1 - \rho_c \end{aligned}$$

On the other hand, in summer:

*ideal case*

$$EER_{id} = \frac{h_1 - h_4}{h_2 - h_1}$$

**real case**

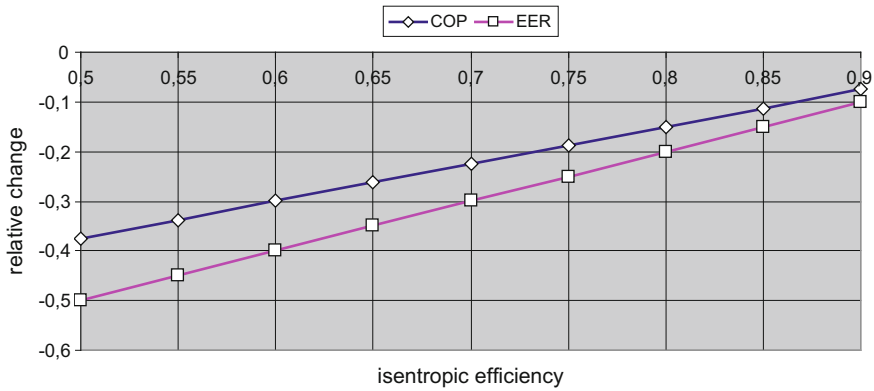
$$EER_{re} = \frac{h_1 - h_4}{h_{2'} - h_1} = \rho_c EER_{id}$$

Figure 2.8 shows the trend of the relative change of the performance parameters (ratio of the difference between the real value minus the ideal one and the ideal value) versus the isentropic efficiency, to give an idea of its influence on summer and winter performances.

Figure 2.9 qualitatively shows the difference between a cycle with an ideal compressor and a cycle with a real one.

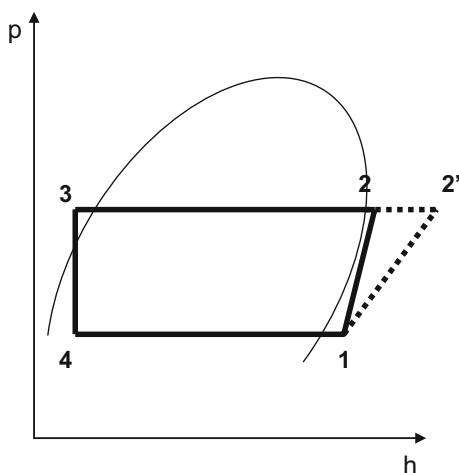
Another typical curve of volumetric compressor is the one linking the flow rate to the pressure head supplied. It is often provided as compression ratio versus volumetric flow rate, as in Fig. 2.10.

For volumetric machines, the pressure supplied by compressors is practically independent from the flow rate, and only related to the circuit hydraulic characteristic curve. Thus the curve compression ratio-flow rate is usually represented by a

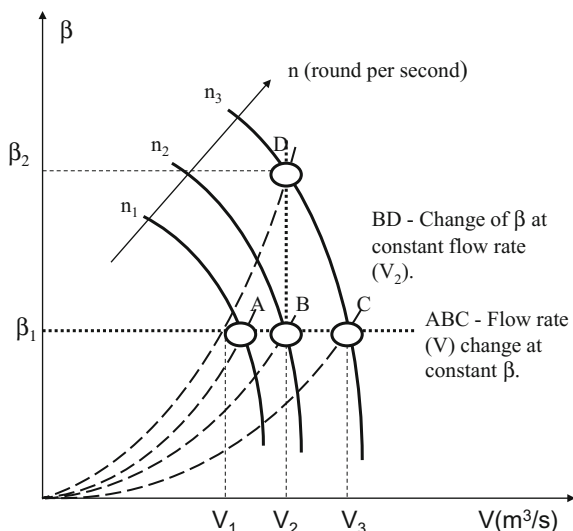


**Fig. 2.8** Relative change of summer and winter performance coefficient versus isentropic efficiency

**Fig. 2.9** Cycles with ideal (continuous line) and real (dotted line) compressor



**Fig. 2.10** Compression ratio versus flow rate of a volumetric compressor.  $V$  is the volumetric flow rate, in this case



straight line with a negative slope (not exactly vertical due to the volumetric efficiency decrease with increasing the compression ratio).

Figure 2.10 also displays the effect of the number of rounds per second of the compressor shaft. An increase of this number moves the curve to the right, toward larger flow rates (from A to C in the figure) at the same compressor ratio. In this case the circuit hydraulic losses have to be decreased. If pressure has to be augmented at constant flow rate (from B to D in the figure), the pressure losses must increase. The corresponding decrease or augmentation of the pressure losses is obtained by opening or closing the metering device (expansion valve).

The characteristic number of rounds per minute of compressors may be also very different for the different types. For example, for reciprocating compressors, they roughly go from a hundred for large and slow compressors with compression ratio 2–3, to a thousand for the smallest ones with a compression ratio around 10.

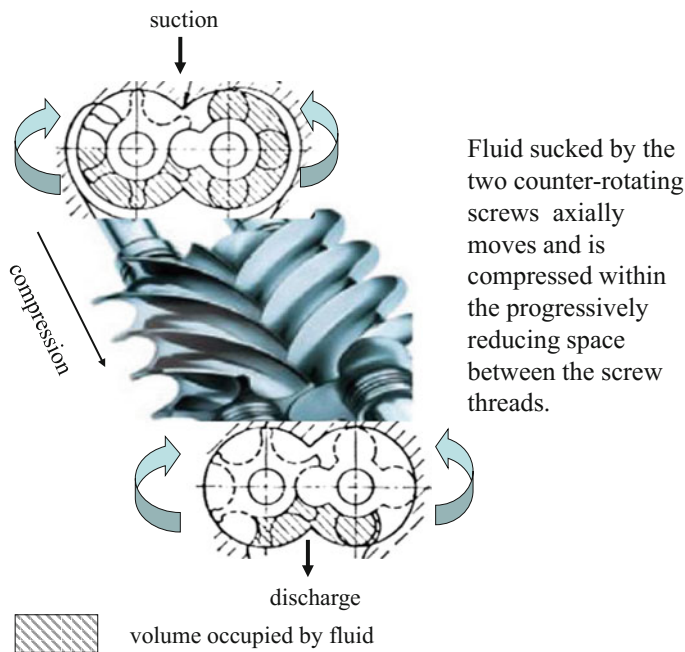
In rotary compressors there might be several thousands rounds per minute.

### 2.2.2 Screw Compressors

Thanks to the technological progress in heating and cooling applications, rotary compressors are often employed instead of reciprocating compressors.

Among other things this is due to their smaller size, larger silentness, smoothly running, low vibration and better control and modulation capability. They can be roughly divided in compressors with a single rotating axis (vane and scroll) and with two rotating axes. Among them we include vane, lobe, screw and scroll compressors. Vane and scroll compressors have a single shaft (single rotation axis), while the others can have both one and two axes. In general, screw compressors have two axes, i.e., two screws.

Screw compressors are used for power values above 50 kW, where they have a better efficiency than the reciprocating ones, even if low power screw compressors (down to 2.25 kW) are available for small applications. Their working principle is shown in Fig. 2.11: two meshing helical screws of different diameters constitute the



**Fig. 2.11** Screw compressor working scheme

compressor rotors. Gas enters at the suction side and moves through the threads as the screws rotate. They force the gas to the discharge port at the end of the screws, progressively reducing the gas volume.

Generally they have smaller compression ratios ( $\beta = 3:4$ ) than the reciprocating compressors, they can be used with several stages in series.

The most common configuration consists of a male rotor with four lobes and a female one with six indentations. Other possible configurations are 3(lobes)/5(indentations) and 5/7. Rotor diameters commonly ranges from 12 to 32 mm.

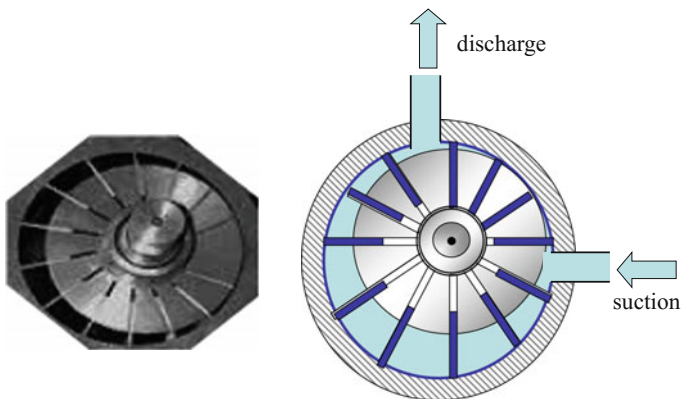
Rotors are located in horizontal cylindrically shaped casings provided with suction and discharge ports. Lubricating oil is injected on the threads to prevent refrigerant leakages, thanks to the presence of an oil film. It is then recovered in an oil separator located close to the discharge port.

Suction phase begins when the two moving rotors leave the suction port open. Fluid enters the compression region and moves along the screw axes. The suction port is closed by the engaging rotors and compression starts, with the discharge port closed.

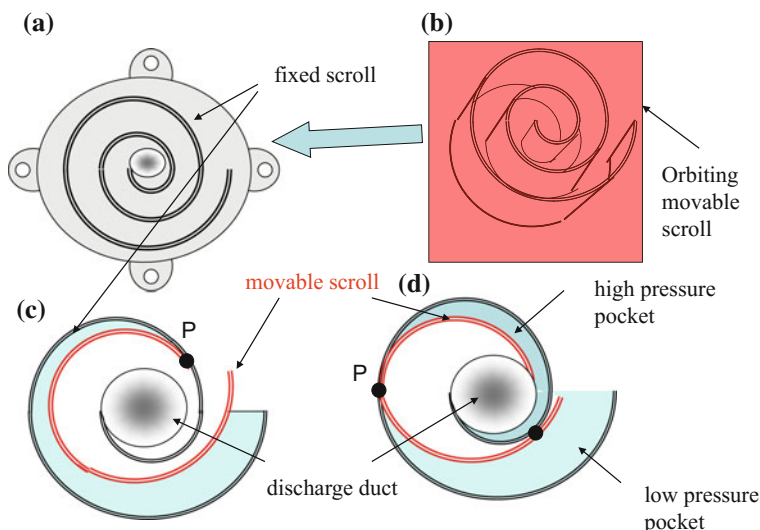
The ratio,  $v_i$ , between the initial (suction) and final (discharge) volumes is the so called “intrinsic volumetric ratio”. Some typical values are 2.2; 2.6; 3.2; 4.4. A given compression ratio corresponds to each  $v_i$ , depending on refrigerant properties. For a given fluid an optimal (top isentropic efficiency) compression ratio can be realized, by using an appropriate intrinsic volumetric ratio. What has been said above about over and sub compression holds also for this type of compressors.

### 2.2.3 Vane and Scroll Compressors

Vane and scroll compressors are mainly employed at the lowest power values. Figure 2.12 shows the scheme of a vane compressor. The rotor is eccentrically placed with respect to the casing. On this, suction and discharge ports are located,



**Fig. 2.12** Vane compressor working scheme



Moving spiral, by orbiting on fixed scroll, figures (a) and (b), forms progressively smaller chambers (pockets), as in figures (c) and (d).

**Fig. 2.13** Working scheme of scroll compressor

without any valve. Sliding vanes are located on the rotor and pushed against the cylindrical casing by centrifugal forces produced by rotation. They originate chambers with a progressively decreasing volume from suction to discharge. A good continuity of the refrigerant flow is guaranteed, in this case too.

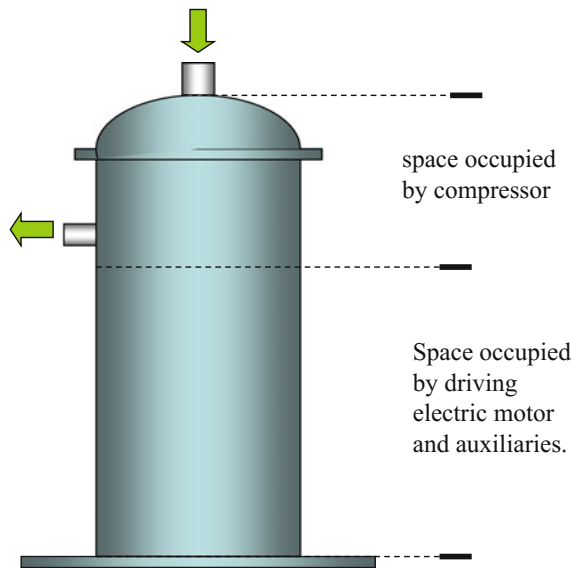
Scroll compressors are basically constituted by two scrolls (spirals), a fixed and a movable one, sketched in Fig. 2.13. The latter is driven by a shaft that makes it orbit (not rotate) about the shaft axis. So, a chamber is formed, compression starts once the suction port is sealed off, progressively reducing the gas volume between the two scrolls.

Seal between fixed and movable scroll is guaranteed by a lubricating oil film. As above said the chamber is in contact with suction, and fluid flows in. After a  $90^\circ$ -rotation, the scroll movement closes the suction port, refrigerant stays confined within the two scrolls and gradually compressed until it is released to the discharge duct.<sup>4</sup>

As all compressors without suction and discharge valves,<sup>5</sup> they have larger isentropic and volumetric efficiencies than the reciprocating ones. They are commonly inserted in a hermetic shell together with the driving electric motor. A typical configuration is shown in Fig. 2.14.

<sup>4</sup>Videos existing on You-tube may help clarify scroll compressor operation.

<sup>5</sup>Actually, a dynamic discharge valve can be adopted in particular in high pressure ratio applications typical of refrigeration. It is located at the scroll discharge port to prevent entry of high pressure gas into the scroll set during the unloaded state.



**Fig. 2.14** Typical external shape of a scroll compressor

By axially separating the two spirals (lifting the movable scroll) capacity reduces to zero. In the discharge phase the movable scroll moves 1 mm apart from the fixed one, annulling the gas flow rate (see: Copeland Scroll Digital™ Compressors).

Generally a scroll compression has its own optimal compression ratio. When the actual compression ratio is lower than this one, over compression losses occur (e.g., half load condition [1]). At high compression ratios sub compression occurs, that can be prevented introducing a dynamic discharge valve, similar to those of reciprocating compressors.

Compression is smooth and silent as very few moving part are involved. It makes this compressor very reliable. Bearing have to be carefully lubricated, while no oil injection in the compression process is needed. The compressor capacity is commonly controlled by an inverter.

The application ranges of compressors can be briefly summarized in Table 2.2. We remark that for the use of ammonia open compressors are used (generally reciprocating and screw), due to its chemical aggressivity. Thus the driving motor is outside the compressor casing.

Furthermore we recall the following definitions.

- Open compressor—the driving motor is separated from the compressor, independently air-cooled and connected by a mechanical coupling.
- Hermetic compressor—motor and compressor are inserted in the same casing and the motor is cooled by the same fluid circulating in the compressor.
- Semi-hermetic compressor—a compressor directly coupled to the driving motor, in the same casing, but with a direct access, separate by the motor.

**Table 2.2** Main types of compressors

| Type          | Model                                   | Capacity (kW)              | Refrigerant                                      | Application   |
|---------------|---|----------------------------|--|---|
| Reciprocating | – Hermetic<br>– Semi hermetic<br>– Open | 0.1/30<br>30/250<br>250/50 | R134a<br>R404A<br>R407A<br>R407C<br>R717<br>R744 | Industrial and commercial refrigerators, low temperature industrial refrigeration |
| Vane          | Hermetic                                | 0.75/3                     | R407C<br>R410A<br>R744                           | Small refrigerators, portable air-conditioning, split systems                     |
| Scroll        | Hermetic                                | 3.5/90                     | R407C<br>R410A                                   | Low and medium size air-conditioning  |
| Screws        | – Semi hermetic<br>– open               | 80/8000                    | R407C<br>R134a<br>R717                           | Medium and large power air conditioning. Industrial refrigeration                 |
| Single screw  | – Semi hermetic<br>– Open               | 100/500                    | R134a<br>R410A                                   | Medium and large power chillers for commercial and industrial climatization       |

### 2.2.4 Control of Compressors' Operation

Compressors must be enabled to work off their nominal load. The on-off control is the simplest way to reach this goal, but it is the most energy consuming. In this way, the reference signal is a set point temperature,  $T_{SP}$ , fixed by a thermostat. When this value is exceeded by  $\Delta T_{SP}$  (depending on thermostat accuracy) the compressor turns off. It starts again once the temperature achieves the value  $T_{SP} - \Delta T_{SP}$ .

In order to keep comfort conditions within the internal environment,  $\Delta T_{SP}$  should be as little as possible, but it could cause too many on- offs, thus stressing too much both compressor and driving motor during start-up phases. Besides, this would produce a COP decrease.

Therefore some techniques have been implemented to work at partial load.

In multicylinder<sup>6</sup> reciprocating compressor one or more cylinders are made ineffective. This is done by bypassing fluid from suction to discharge of the cylinder we want to deactivate, by signals sent to a solenoid valve. So we obtain a step reduction of the active cylinders number as shown in Table 2.3.

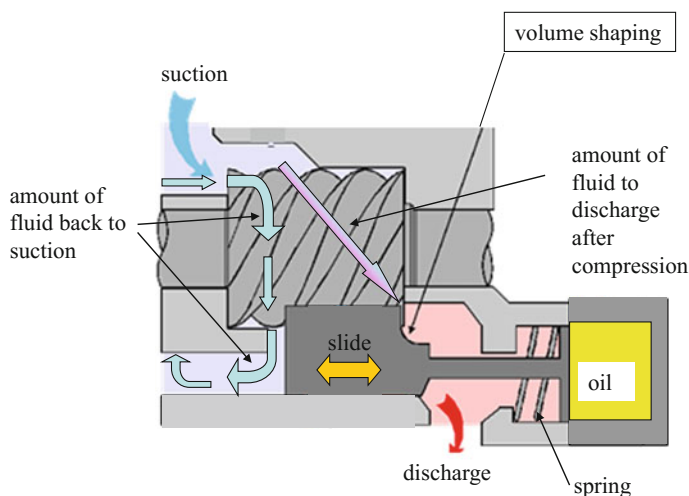
Therefore the number of on-off is reduced. The load to be supplied to the compressor does not proportionally decrease with the percent of reduction (e.g., a 33% reduction may correspond to 40% of the nominal load), as the ineffective cylinders are anyway operated by the crankshaft, consuming power.

<sup>6</sup>The multiple cylinder compressor has also the advantage to keep the fluid flow smoother.



**Table 2.3** Active cylinder reduction in a reciprocating compressor

| Total number of cylinders | Active cylinders | Capacity (%) | Reduction (%) |
|---------------------------|------------------|--------------|---------------|
| 4                         | 4                | 100          | no            |
|                           | 2                | 50           | 50            |
| 6                         | 6                | 100          | no            |
|                           | 4                | 67           | 1/3           |
|                           | 2                | 33           | 2/3           |



**Fig. 2.15** Scheme of a sliding valve for screw compressors. A piston activated by pressurized oil moves the slide towards suction or away from it, modifying the screw length engaged in compression

A method to obtain a continuous modulation (at least in a given range) consists in changing the rotation speed of the driving motor (being it an electric motor or an internal combustion engine).

Electric heat pumps often use an inverter to control the electric motor. Such a device changes the feeding frequency from lower values than the mains one (50 or 60 Hz) to much higher frequencies. The main advantages are: a better achievable comfort, smoother start up, but, above all, an increase of the instantaneous and seasonal COP. It is even possible to gain a COP increase, respect to the nominal value, at a reduced flow rate. This is due to the use of oversized (in this case) heat transfer surfaces in comparison with the design nominal conditions. Thanks to this the temperature differences among heat exchangers and thermal sources shrink.

**Screw compressors.** A typical control employed in screw compressors, using a slide valve, is outlined in Fig. 2.15.<sup>7</sup> A slot, parallel to the screws axis, can be

<sup>7</sup>A duct can also be inserted to allow the fluid flow toward the economizer.

gradually opened (or closed) by a sliding device. This is activated by the pressure exerted by an oil piston, depending on the actual operation requirements.

The intake pressure acts on the left of the valve and the discharge pressure on the right. The right side contour of the slide is properly shaped to keep the intrinsic volumetric ratio practically constant within a given range (e.g., 70% of the full load). So a pretty much constant isentropic efficiency is obtained in the above range.

When the slide is totally shifted to the intake side, the suction volume has its minimum value and compression takes place all along the screws length. By moving rightward (to discharge), the slide increases the suction volume, reducing the screw length got involved in compression. Thus the flow recirculating back to suction increases, while the one discharging decreases. As the consequence of this the actual sucked volume to be compressed lowers and the compression ratio increases. Viceversa, if the slide moves to the opposite side.

The slide shift can be either stepwise or continuous, depending on the application requirements. The use of a continuous shift control is more convenient in the presence of fully variable loads.

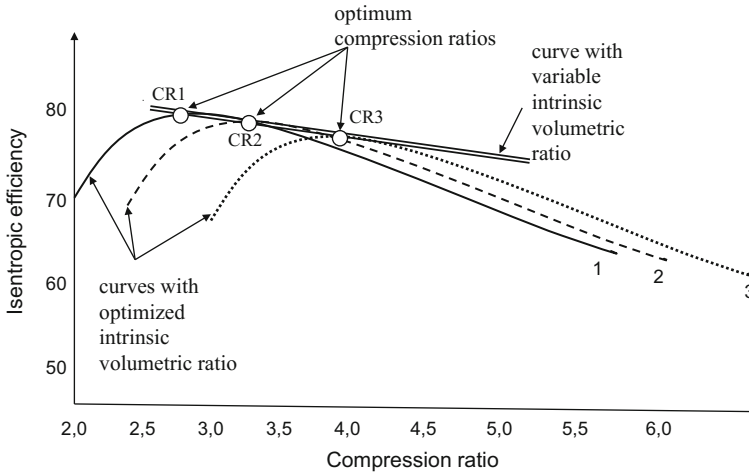
The stepwise configuration generally has four levels (% of the full load):

- 10% minimum level determined by the oil injected in the compressor and commonly used only for start-up;
- 50%
- 75%
- 100%, full load.

In some cases both stepwise and continuous controls are feasible on the same compressor. Several types of the mentioned control method exist. For example: an additional flow rate bypass at partial loads can be adopted as well as a variable volumetric ratio, so that the compressor could always operate with the top isentropic ratio in correspondence to the required loads.

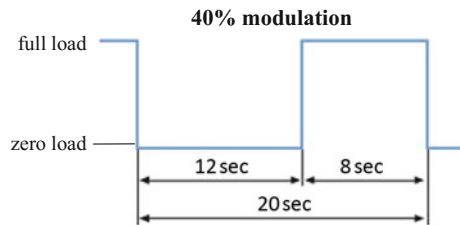
Referring to Fig. 2.16, let us suppose to have a compressor characterized by curve 2 (isentropic efficiency vs. compression ratio), with an optimal compression ratio CR2 and constant intrinsic volumetric ratio. CR2 is the most frequently occurring value during daily operation. Anyway load variations can lead to different compression ratios, for instance CR1 or CR3. In this case a decrease of isentropic efficiency would occur on curve 2, i.e., if we work at constant volumetric ratio. Thus, in the case of rather frequent load changes, a compressor with a variable intrinsic volumetric ratio is suitable, where the trend of isentropic efficiency versus compression ratio passes trough CR1 and CR3. Anyway it is always recommended to contact the manufacturers.

**Scroll compressors.** Capacity modulation, except for the on-off and variable speed methods, can be also obtained by axially distancing the movable scroll from the fixed one. Meanwhile the compressor keeps on rotating, with no significant power losses, at list down to a certain degree of modulation. The procedure is the following (at least the one adopted in Digital Scrolls by Copeland): in nominal conditions the movable scroll (lower scroll) is kept in place, keeping the nominal axial position. The scroll movement is activated by oil pressurized, or depressurized



**Fig. 2.16** Trend of isentropic ratio of a screw compressor with optimized intrinsic volumetric ratio (curves 1, 2, 3) and with variable intrinsic volume ratio. CR1,2,3 are the points corresponding to top isentropic efficiencies

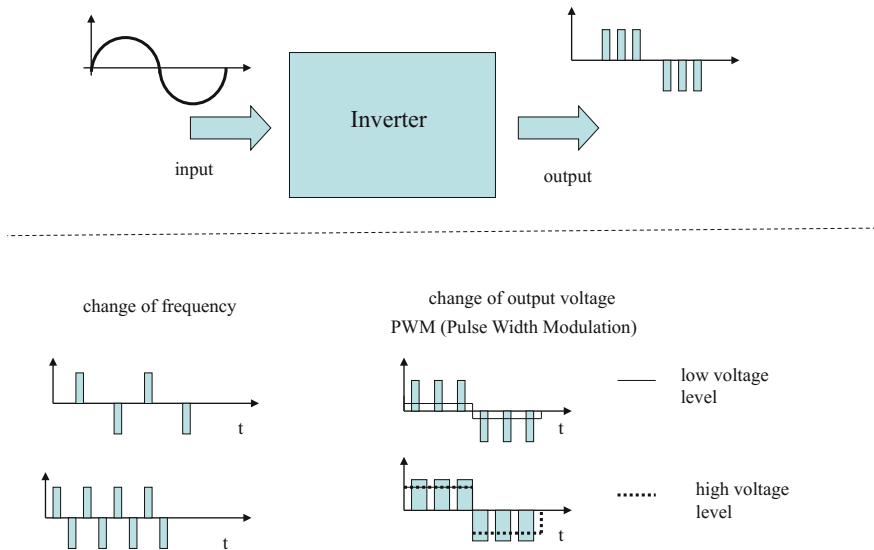
**Fig. 2.17** Load modulation cycle of a scroll compressor



through an electric control valve. This valve, opens and closes, driven by a digital signal, separating the two scrolls axially by one millimeter or restoring the nominal axial position. When the two scrolls are in the nominal position the compressor works at full capacity. When they are separated it works at zero capacity. Modulation is achieved by varying the time of full and zero capacities. The cycles generally last from 10 to 30 s. Figure 2.17 shows a digital scroll modulation in a 20 s cycle where the compressor operates at full capacity for 8 s and 12 s at zero capacity.

### 2.2.5 Inverter Control

Inverter allows for varying the compressor rotation frequency in order to change flow rate at constant compression ratio and, therefore, at constant volumetric and isentropic efficiencies. The inverter used for a.c. electric motors transforms grid a.c.



**Fig. 2.18** Inverter operation

into d.c. voltage. As an output it generates electric pulses, with different amplitude and frequency, simulating an a.c. voltage. The value of this latter is modulated by changing the signal amplitude, PMW (Pulse Width Modification), at a fixed frequency. The change of frequency of simulated voltage is obtained by varying pulses frequency, and, thus, the number of rounds per second of compressors, see Fig. 2.18.

As the input a.c. voltage is converted in a d.c. voltage, at first, also a three-phase load can be fed by a single-phase voltage input.

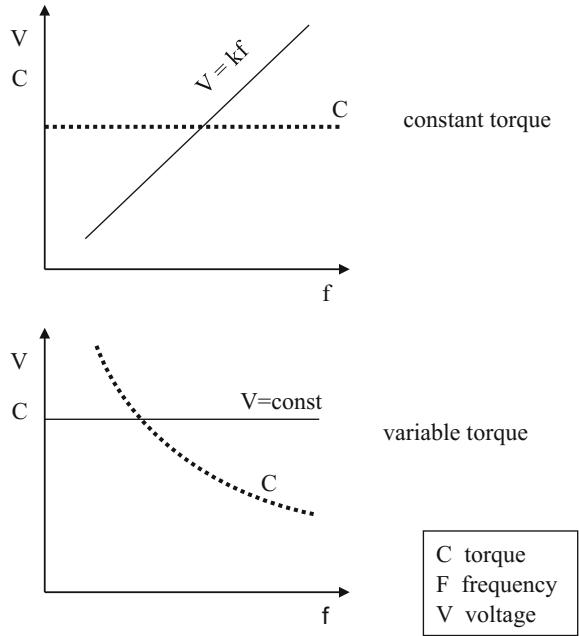
The output signal has a harmonic residual, causing electromagnetic noise, which may propagate in the surrounding environment.

If  $V$  is the voltage applied to the motor,  $\Phi$  the magnetic flux,  $f$  the frequency and  $C$  the torque applied to the rotor, the following relations hold:

$$\begin{aligned}
 V &\propto \Phi \omega \\
 P &= C \omega \\
 C &\propto \frac{V^2}{\omega} \propto \frac{(\Phi \omega)^2}{\omega^2} \\
 \omega &= 2\pi f
 \end{aligned}$$

Usually the magnetic flux is kept constant to avoid magnetic saturation of the iron nucleus with an increase of parasitic currents and consequent overheating (in hermetic compressors it would cause refrigerant overheating). To this purpose, a voltage proportional to frequency has to be applied and power grows up linearly

**Fig. 2.19** Voltage versus frequency trends and torque behavior



with increasing frequency. The top achievable voltage is the one provided by the electric grid.

Nevertheless frequency can be further increased, but doing so the relation between voltage and frequency is no longer linear and the torque decreases.

In other cases, voltage is kept constant making the magnetic flux diminish, to compensate iron losses that increase with the frequency squared. Consequently power decreases with increasing frequency.

The above two cases are sketched in Fig. 2.19.

A better control can be achieved by using the so called “vector inverter, which can control both active (in phase with voltage) and reactive (90° out of phase) current components. For a better control, device, named “encoder”, may further be adopted. It tracks the turning of motor shafts to generate digital position and motion information.

Dynamic power losses depend on the square of feeding voltage and on commutation frequency. As an example we report, in Table 2.4, some data related to an

**Table 2.4** Some inverter data

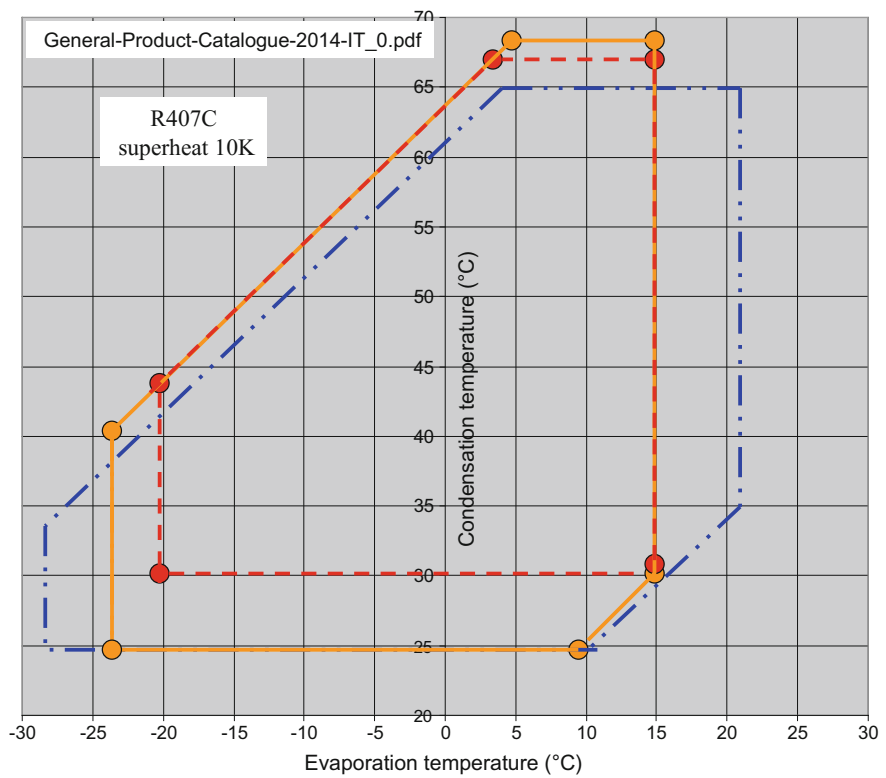
|  |      |      |      |      |      |     |      |
|--|------|------|------|------|------|-----|------|
| Typical useful mechanical power (kW)       | 5.5  | 7.5  | 11   | 15   | 18   | ... | 45   |
| Estimated power losses at nominal load (W) | 269  | 310  | 447  | 602  | 737  | ... | 1636 |
| Efficiency                                 | 0.96 | 0.96 | 0.96 | 0.96 | 0.96 | ... | 0.96 |

electric motor inverter. For a more complete overview of existing products reader can refer to [2] by Danfoss. The inverter efficiency is commonly above 92%.

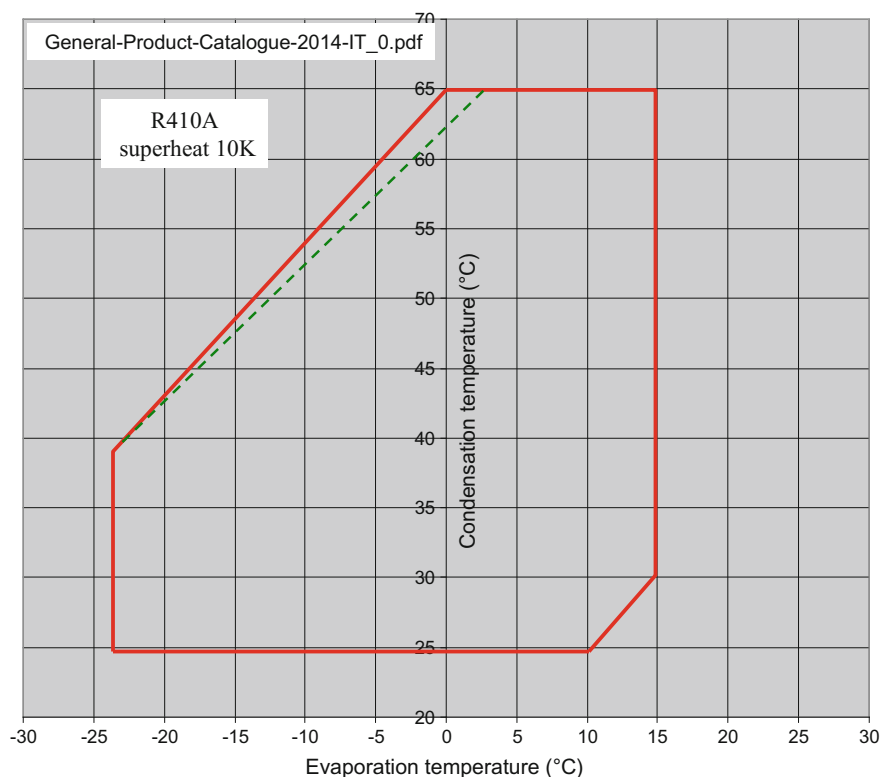
2.2.6 The Compressor Operation Range

A dedicated region on the plane evaporation versus condensation temperature is provided for any compressor, for each given refrigerant. This is the compressor operating range. Out of this range manufacturers do not guarantee its performances. In Figs. 2.20, 2.21, 2.22, 2.23, 2.24 some of the above ranges are shown, obtained by elaborating the data supplied by Copeland (General-Product-Catalogue-2014-IT\_0.pdf). For more detailed information refer to [3].

Suction superheat is also specified in addition to the type of fluid used. The first three figures refer to scroll compressors with 10 °C superheat. The fourth (2.23) refers to reciprocating compressors, with four or six cylinders, provided with a



**Fig. 2.20** Operation ranges of some scroll compressors by Copeland, each marked by a different line, using R407. Power of driving motor from 1.1 to 22 kW, capacities from 3.7 to 81.7 kW



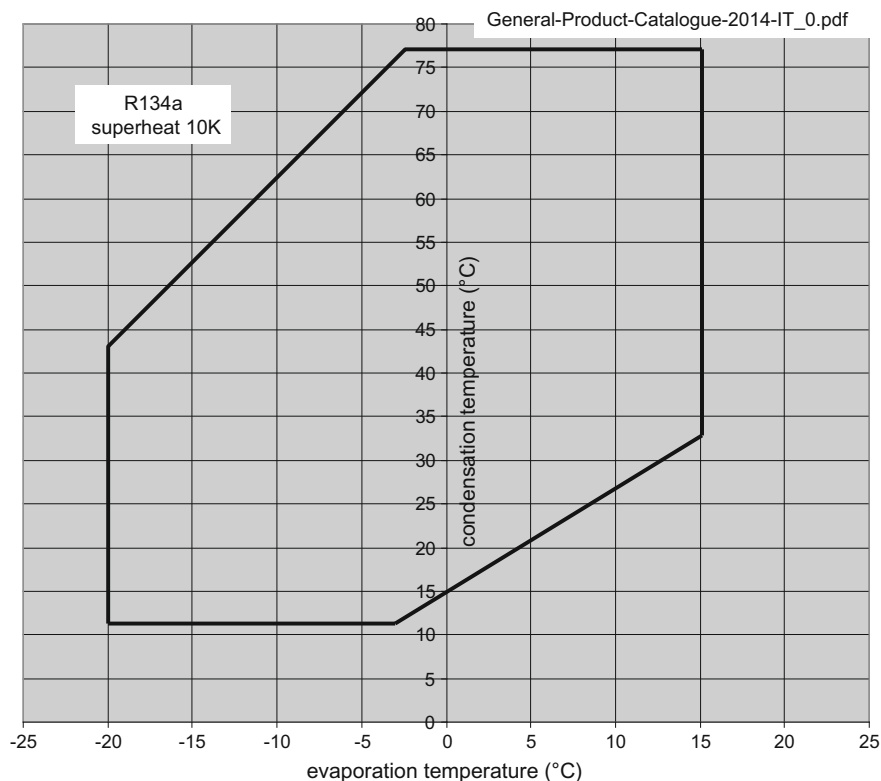
**Fig. 2.21** Operation ranges of Copeland scroll compressors using R410 A as refrigerant. Motor powers from 1.4 to 44 kW, capacities from 5 to 160 kW

continuous inverter modulation. These figures give an idea of which are the main key factors influencing compressor performances. It is the case to stress some more that the cooling of electric motors is performed by the same refrigerant in hermetic compressors.

As already said, Fig. 2.23 concerns reciprocating compressors with an external air cooled driving motor. With regard to this figure we remark:

- The fan is the motor cooling fan.
- The SGRT (Suction Gas Return Temperature) is the vapor temperature at compressor suction. The degree of superheat is given by the difference between this temperature and the evaporation one.
- SH (Superheat) is the suction superheat.

In the end we show the changes of operation range due to injection of vapor, taken from condenser exit, into the compression, with refrigerant R410A. Such a procedure widens this range, increasing heating potentiality and lowering discharge vapor temperature (improving isentropic efficiency) (referring to figs. 2.24 and 2.25).



**Fig. 2.22** Operation ranges of Copeland scroll compressors using R134a. Motor powers 1.5 to 22 kW, capacity 3.3 to 53.2 kW

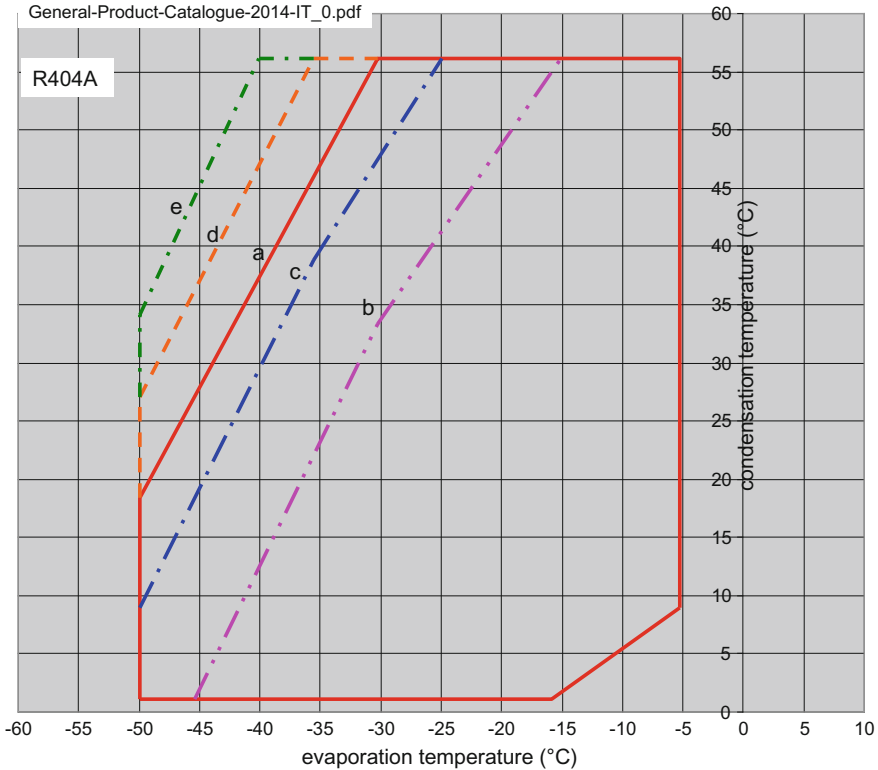
## 2.3 Expansion Valve

As above said it is a metering device that feeds refrigerant to the evaporator, lowering its pressure from the condenser value to that of the evaporator, in order to keep suitable transformations temperatures for heat sources. In simplest applications it is obtained by a fixed bore capillary tube where the total-system charge flows in any operating condition. It has to be long enough to supply the total pressure drop at full flow rate and is generally helically coiled.

Of course, such a device is not able to face load variations. Therefore several systems allowing for varying the discharge area of the valve according with the actual required load have been employed.

The rationale is quite simple. As a consequence of a reduction of the heating power requested by the internal environment and at a constant flow rate, the condensation phase shifts the outlet point towards larger subcooling in the liquid. At the same time the superheat at evaporator exit increases. Both subcooling and



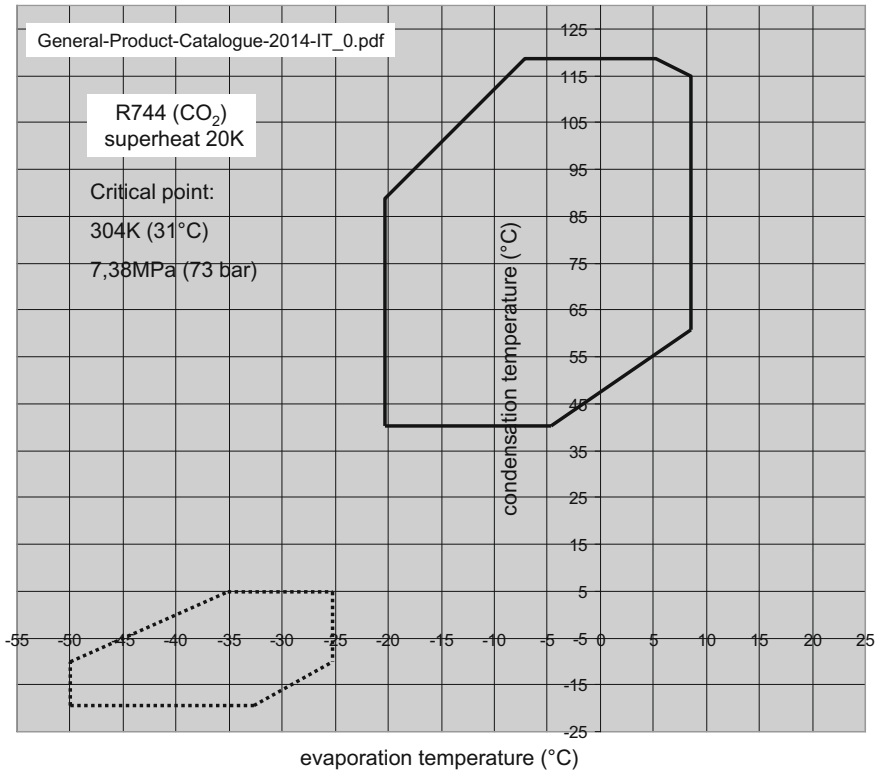


**Fig. 2.23** Operation ranges of Copeland™ Stream Digital with Core Sense™ Diagnostics reciprocating compressors (4–6), with refrigerant R404 A. They use continuous modulation by an inverter from 50 to 100% (4 cylinders) and from 30 to 100% (6 cylinders), with the following characteristics (letters *a*, *b*, *c*, *d*, *e* refer to each graph): **a** 25 °C SGRT at 100% load or 0 °C SGRT + cooling fan, driving motor modulation at 33% (6 cylinders) and 50% (4 cylinders); **b** 25 °C SGRT with cooling fan and modulation at 33% (6 cylinders) or 50% (4 cylinders); **c** 0 °C SGRT with cooling fan and modulation at 33% (6 cylinders) or 50% (4 cylinders); **d** 25 °C SGRT at 100%; **e** SH > 20 °C at 100%

superheat increase as larger is the unbalance between the requested and the available power.

It is, therefore, necessary to lower the flow rate in such a case. The valve discharge area has to be reduced. In many cases the actuating control signal comes from a sensor measuring vapor temperature at evaporator exit. This to keep vapor superheat at compressor suction fixed at a set-point. In this case we speak of thermostatic valve and this method is applied in dry evaporators (those where the exiting vapor is superheated).

If  $\Delta p_v$  is the valve pressure drop with a mass flow rate  $m$ , it can be set forth as  $Km^2$ , where  $K$  is the corresponding flow coefficient. At a reduced flow rate,  $m'$ , the valve partially closes, keeping the pressure drop constant. The new flow coefficient



**Fig. 2.24** Refers to reciprocating compressors used with carbon dioxide. The graph (*continuous line*) on the top right regards compression in the hypercritical region, while the lower graph (*dotted line*) concerns a subcritical compression

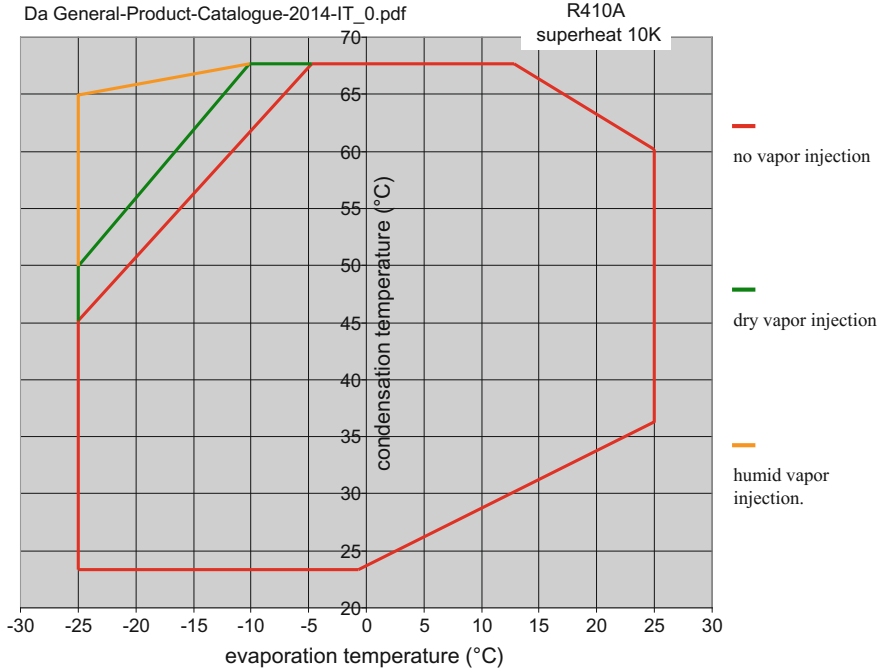
has to become  $K' = \Delta p_v / m'^2$ . Both  $K$  and  $K'$  are two values of the flow characteristic of the installed valve.

*Example 2.2* A heat pump, working with R134a<sup>8</sup> supplies a nominal heating load of 10 kW at 44 °C.<sup>9</sup> Let us suppose liquid inlet into the expansion valve to be saturated. With data provided in the following Table 2.5 the mass flow rate is:

$$m = \frac{Q_c}{h_v - h_l} = \frac{10}{158.7} = 0.063 \text{ kg/s}$$

<sup>8</sup>The data are taken from NIST Chemistry Web Book.

<sup>9</sup>We suppose the de-superheating is used for sanitary hot water production. This may occur in offices, where sanitary water requirements are usually low.



**Fig. 2.25** Change of operation range due to vapor injection

If the required power has a 10% decrease, the mass flow rate has to be reduced of the same % to keep the same conditions. If the lower temperature is  $-2^{\circ}\text{C}$ , lamination goes from 1130 to 272 kPa, with a pressure drop equal to 858 kPa. The relation between the new,  $K'$ , and the old,  $K$ , flow coefficients is:

$$\frac{K'}{K} = \left(\frac{m}{m'}\right)^2 = 1.23$$

First of all, we remark that the refrigerant flow rate cannot be reduced at one's choice. A first constraint is imposed by the need of preventing liquid to flow into the compressor. Therefore vapor has to leave the evaporator (dry evaporator) with a certain degree of superheat (not less than  $3\text{--}5^{\circ}\text{C}$  of static superheating). Thus the expansion valve receives a temperature signal from the evaporator exit, so that the flow rate be reduced if the detected value is lower than the fixed set-point, or increased if it is larger.

The following definitions of superheating are given:

- Static superheating: corresponding to this value the valve starts opening and the flow rate increase begins.
- Opening superheating: is the value, larger than the static superheating, necessary to produce a given valve potentiality.
- Operating superheating: is the sum of the two previous ones.

**Table 2.5** R134a data

| °C | kPa    | $v_l$ (m <sup>3</sup> /kg) | $v_v$ (m <sup>3</sup> /kg) | $h_l$ (kJ/kg) | $h_v - h_l$ (kJ/kg) | $h_v$ (kJ/kg) |
|----|--------|----------------------------|----------------------------|---------------|---------------------|---------------|
| -2 | 272.2  | 0.0007684                  | 0.0744                     | 49.17         | 200.12              | 249.29        |
| 0  | 292.8  | 0.0007723                  | 0.0693                     | 51.86         | 198.60              | 250.46        |
| 2  | 314.6  | 0.0007763                  | 0.0647                     | 54.55         | 197.07              | 251.62        |
| 4  | 337.7  | 0.0007804                  | 0.0604                     | 57.25         | 195.53              | 252.78        |
| 6  | 362.0  | 0.0007845                  | 0.0564                     | 59.97         | 193.95              | 253.92        |
| 8  | 387.6  | 0.0007887                  | 0.0528                     | 62.69         | 192.36              | 255.05        |
| 12 | 443.0  | 0.0007975                  | 0.0463                     | 68.19         | 189.11              | 257.29        |
| 16 | 504.3  | 0.0008066                  | 0.0408                     | 73.73         | 185.74              | 259.47        |
| 20 | 571.7  | 0.0008161                  | 0.0360                     | 79.32         | 182.28              | 261.60        |
| 24 | 645.8  | 0.0008261                  | 0.0319                     | 84.98         | 178.70              | 263.68        |
| 26 | 685.4  | 0.0008313                  | 0.0300                     | 87.83         | 176.87              | 264.70        |
| 28 | 726.9  | 0.0008367                  | 0.0283                     | 90.70         | 175.00              | 265.69        |
| 30 | 770.2  | 0.0008421                  | 0.0266                     | 93.58         | 173.09              | 266.67        |
| 32 | 815.4  | 0.0008478                  | 0.0251                     | 96.48         | 171.16              | 267.64        |
| 34 | 862.6  | 0.0008536                  | 0.0237                     | 99.40         | 169.18              | 268.58        |
| 36 | 911.9  | 0.0008595                  | 0.0224                     | 102.33        | 167.17              | 269.50        |
| 38 | 963.2  | 0.0008657                  | 0.0211                     | 105.29        | 165.12              | 270.41        |
| 40 | 1016.6 | 0.0008720                  | 0.0200                     | 108.27        | 163.01              | 271.28        |
| 42 | 1072.2 | 0.0008786                  | 0.0189                     | 111.26        | 160.88              | 272.14        |
| 44 | 1130.1 | 0.0008854                  | 0.0178                     | 114.28        | 158.69              | 272.97        |
| 48 | 1252.9 | 0.0008997                  | 0.0160                     | 120.39        | 154.16              | 274.55        |

A liquid separator can be inserted immediately after the evaporator, just to be sure and to have a low superheating. Actually these separators are used in the case of wet evaporators, where no superheating is required to increase efficiency.

A further important parameter is the liquid subcooling at the expansion valve inlet. This is necessary to avoid vapor bubbles formation in the liquid line leading to the valve that would reduce its performances. A typical minimum subcooling value is 4 °C.

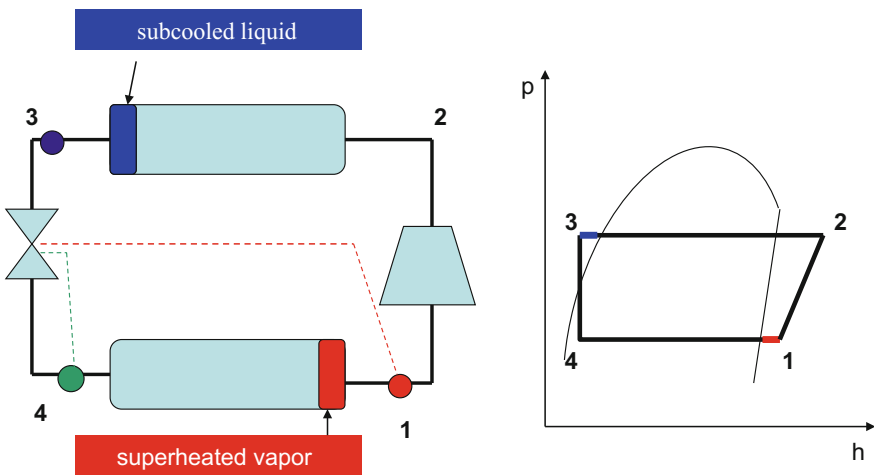
The expansion valves are generally classified as:

- Constant pressure expansion valve—also improperly called automatic expansion valve, it keeps the pressure inside the evaporator constant, no matter what the load inside the evaporator is. It does not allow the control of flow of refrigerant and, thus, this type of valve is not used when this control is needed.
- Thermal (thermostatic) expansion valve—it controls the amount of refrigerant flow thereby controlling superheating at evaporator outlet. Thermal expansion valves are often generically referred to as “metering devices”. They are employed with variable thermal load.

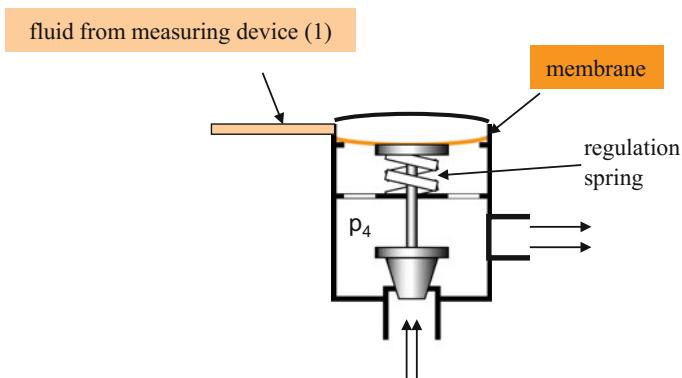
They operate according to the superheating at evaporator exit and to its pressure. This pressure has to be kept below a fixed threshold called MOP (Maximum

Operating Pressure) to avoid any abnormal operation of the compressor. In normal conditions (pressure below MOP) the expansion valve works according to the superheat, but once MOP is reached the valve orifice reduces preventing any further pressure increase.

Figure 2.26 schematically shows the location (in the cycle) of the temperature sensor controlling the expansion valve. The related signal is collected by a bulb connected to the valve through a capillary tube. The fluid in the bulb contracts and expands according to the refrigerant superheat and cause changes in volume of a chamber of the valve, provided with an elastic membrane. It is attached to the valve stem moving up and down to increase or decrease the flow rate, as depicted in Fig. 2.27.



**Fig. 2.26** Significant physical quantities and signals controlling operation of an expansion valve



**Fig. 2.27** Scheme of a thermostatic valve

On the top of the figure there is the membrane connected to the stem. The valve shutter is located on the other end of the stem and controls the valve orifice opening (in some types the orifice is interchangeable). One more chamber is placed underneath, where the evaporator pressure acts and a regulation spring is contained.

The pressures below act on the membrane:

- $p_{\text{sup}}$ —corresponding to the overheating temperature. Its values grow up with the superheat.
- $p_{\text{ev}}$ —evaporation pressure ( $p_4$  in the figure).
- $p_{\text{spr}}$ —spring pressure, settle at an appropriate value according to the desired static superheat.

The resulting pressure acting on the membrane is  $p_{\text{sup}} - (p_{\text{ev}} + p_{\text{spr}})$ . For a given refrigerant, to size and/or choose a thermostatic valve we need to know:

- Temperature and pressure,  $T_{\text{ev}}$  and  $p_{\text{ev}}$ , of evaporator.
- Evaporator capacity.
- Condensation temperature and pressure,  $T_{\text{cond}}$  and  $p_{\text{cond}}$ .
- Liquid temperature,  $T_l$ , at the valve inlet.
- Sum of pressure losses in the liquid line, distributor and evaporator  $\Delta p_l$ .

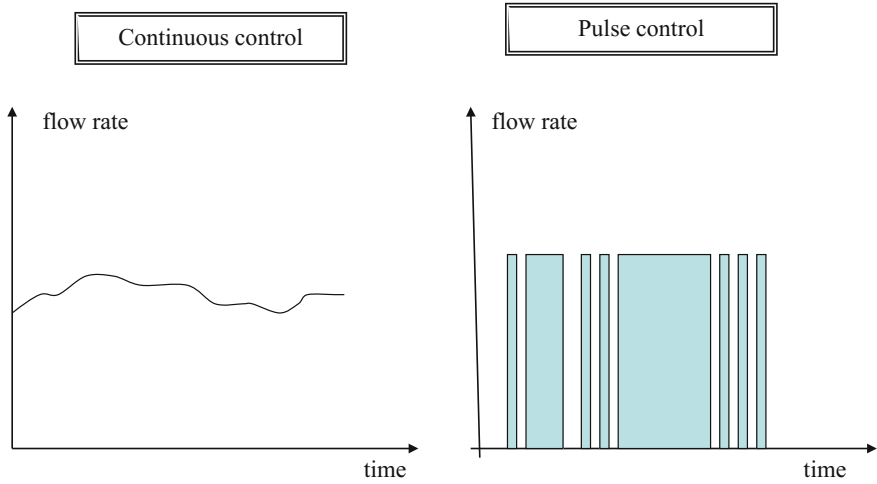
The use of electronic expansion valves is becoming ever more common nowadays. In this type of valves the stem is controlled either by an electric motor (flow continuous modulation) or by a pulse controller, modulating the pulses duration (pulse flow modulation). They allow for a better flexibility than the traditional thermostatic ones with regard to:

- MOP and, therefore, to the evaporator temperature.
- Superheating, so reducing its value.
- Possible injection into the evaporator of the optimum vapor flow at partial loads. This way, it is possible to keep instantaneously superheat at its minimum optimal value, thanks to the precision provided by the electronic control.

In the case of continuous modulation, controller supplies a low voltage signal to the motor, capable of making rotor move either clockwise or anticlockwise. Pulse modulation provides proper windings with voltage pulses, axially moving a magnet connected with the valve stem. The valve can only work fully open or fully closed. Flow is regulated through pulses duration (see Fig. 2.28).

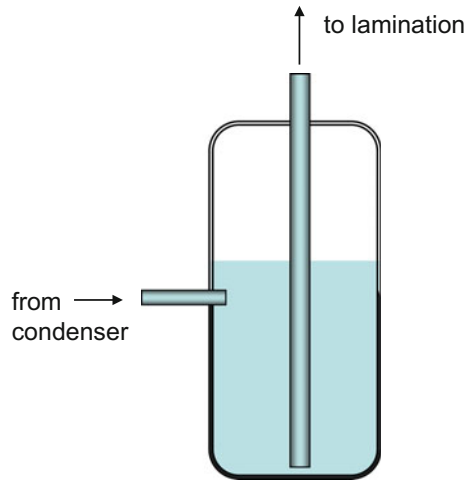
## 2.4 The Liquid Receiver

Generally a tank is placed at condenser outlet and upstream the expansion valve to store high pressure liquid leaving the condenser. It is sized to contain the whole refrigerant charge during the off-duty periods. Its purpose is to collect fluid when load fluctuations occur, thus allowing for flow rate modulation by the lamination valve.



**Fig. 2.28** Control methods of electronic expansion valves

**Fig. 2.29** Liquid receiver



A scheme of the receiver is illustrated in Fig. 2.29. It is a cylindrical steel tank with a pipe introducing the refrigerant coming from the condenser and an internal dip tube, ensuring that only 100% of liquid leaves the receiver.

Of course this type of device is not used when a capillary tube is used instead an expansion valve, as no flow modulation is possible.

## 2.5 Evaporator and Condenser

They can exchange heat with several types of indoor and outdoor sources. There is a widespread use of rather small systems (split systems), where both indoor and outdoor heat exchangers are cooled by air blown by properly sized axial fans. In this case we say we are using an air/air heat pump, in the sense that the refrigerant exchanges heat directly with air on both the heat exchangers. The internal source can also be water of a hydraulic system or sanitary water, while the external source can be water and even ground.

With regard to the type of thermal sources heat pumps can be synthetically classified as:

- Air/Air heat pumps, if both the sources are air.
- Air/Water heat pumps, if the outer source is air and the inner one is water, as in water heating systems.
- Water/Water heat pumps, if both the sources are water.

In any case the first word refers to the outer source and the second one to the inner source.

When the heat source/sink is air (air/air or air/water heat pumps) the air cooled heat exchanger mainly consists of a finned tube bundle with rectangular box headers on both ends of the tubes. Cooling air is provided by one or more fan.

If refrigerant exchanges heat with water, plate and frame heat exchangers are used. They have corrugated metal plates to transfer heat between the fluids (Fig. 2.30). They may be welded, semiwelded and brazed (most commonly adopted in heat pumps).

They are high heat transfer efficiency and compact<sup>10</sup> heat exchangers.<sup>11</sup> The plates are generally spaced by rubber sealing gaskets (Gasketed Plate Heat Exchangers GPHE) and are pressed to form troughs at right angles to the main direction of flow. Each fluid flows in gaps, each formed by two consecutive plates, 1.3–1.5 mm wide.

Plates are compressed together in a rigid frame and form a set of parallel channels with alternating hot and cold fluids. They can easily be disassembled for cleaning and maintenance purposes as well as for inserting further elements.

The plates can be also brazed (e.g., copper brazed) instead of welded, thus named Brazed Plate Heat Exchangers (BPHE). Plates are shaped to promote high levels of turbulence in order to increase heat transfer efficiency and self-cleaning.

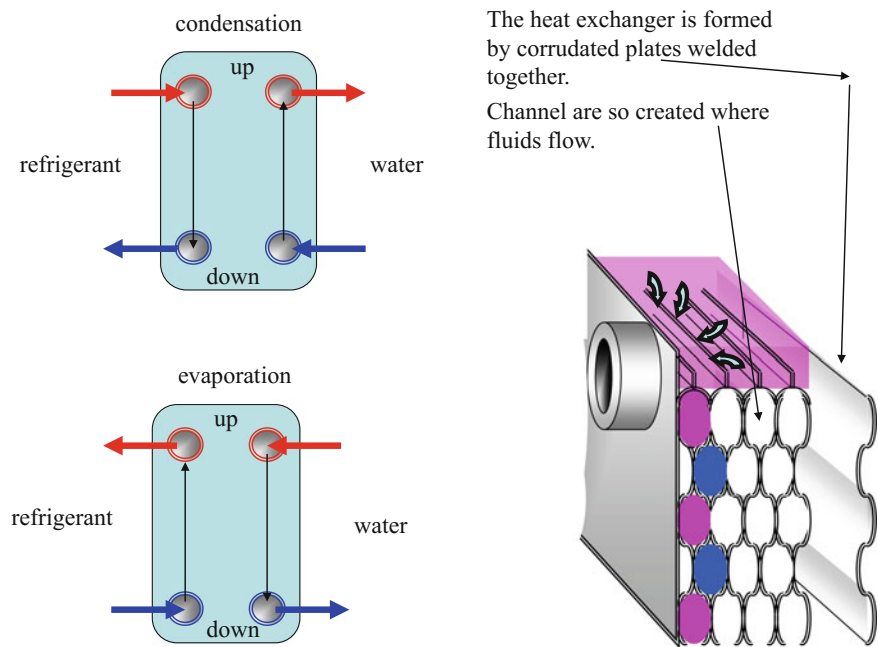
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<sup>10</sup>A heat exchanger compactness is usually based on the value of two parameters: the hydraulic diameter  $D_H$  (the lowest of those employed for the two fluids) and the ratio between the heat transfer area and the volume where fluids flow  $S/V$ . The following definitions are given:

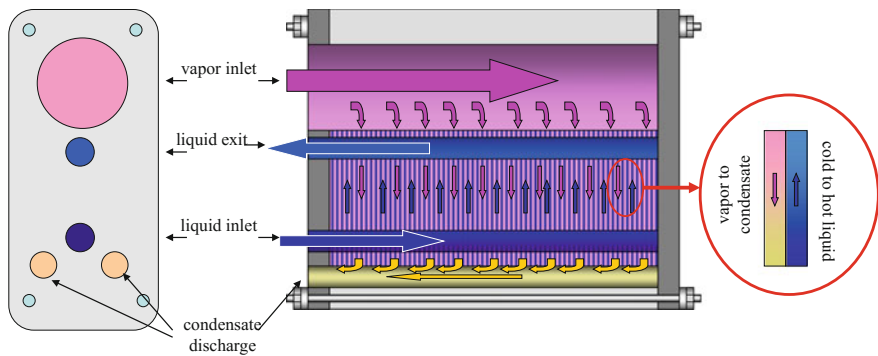
- Conventional heat exchangers for  $D_H > 5 \text{ mm}$  or  $S/V < 400 \text{ m}^2/\text{m}^3$ .
- Compact heat exchangers for  $1 < D_H(\text{mm}) < 5$  or  $S/V > 400$ .

<sup>11</sup>For a more detailed description of these heat exchanger the interested reader can also refer to [4].





**Fig. 2.30** Scheme of a plate heat exchanger



**Fig. 2.31** Scheme of a plate condenser. Vapor enters the larger duct then flowing through the gaps formed by the plates. It is collected by the two lower ducts. Cooling liquid flows in thermal contact with vapor

Some changes are adopted in two-phase applications to account for the difference between liquid and vapor specific volumes. Figure 2.31 shows a sketch of an Alfa-Laval condenser. Vapor enters a wider channel (to take into accounts its larger specific volume) and drops down through the plates toward two channels were condensed liquid flows.

**Table 2.6** Size and weight of plate exchangers

| Length (mm) | Width (mm) | Weight (kg) |
|-------------|------------|-------------|
| 194         | 33–101     | 1.3–2.8     |
| 306         | 34–298     | 2.5–16.6    |
| 613         | 62–470     | 15.3–84.8   |
| ...         | ...        | ...         |

Some features of BPHE for air/water heat pumps from HYDAC International are reported herein just to give some order of magnitude of the main parameters [5]:

Some operating data:

“Operating data Plate material Stainless steel 1.4401 (AISI 316).

Braze material Copper (standard), Nickel.

Pressures Copper brazed: max. 30 bar (test pressure 45 bar).

Nickel braze: max. 10 bar.

Use nickel-brazed plate heat exchangers with corrosive fluids: e.g., ammonia, sulphides and sulphates, deionizer or dematerialized water and other fluids on request.

Temperature range up to +200 °C (freezing point and boiling point must be taken into consideration).

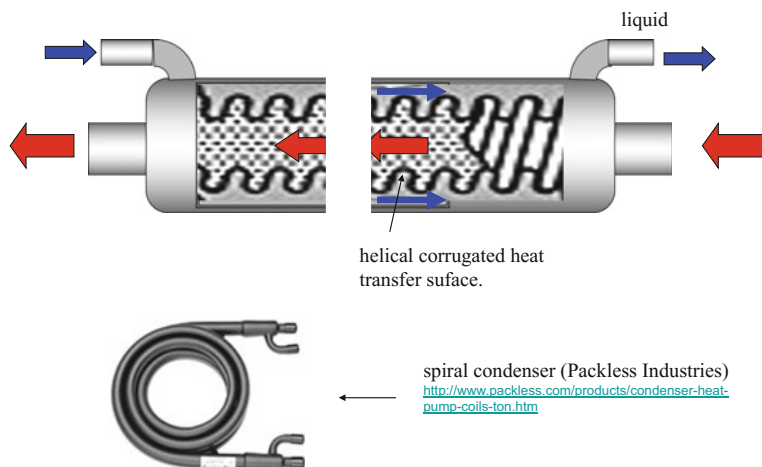
Contamination: the quantity of particles in suspension should be less than 10 mg/l...” (Table 2.6).

Capacities ranging from 0.7 to 186 kW are, for example, available for water cooled BPHE condensers and from 0.7 to 141 kW for evaporator, both using R410A, and from 0.7 to 176 kW and from 0.7 to 141 kW respectively for condensers and evaporators using R134a.

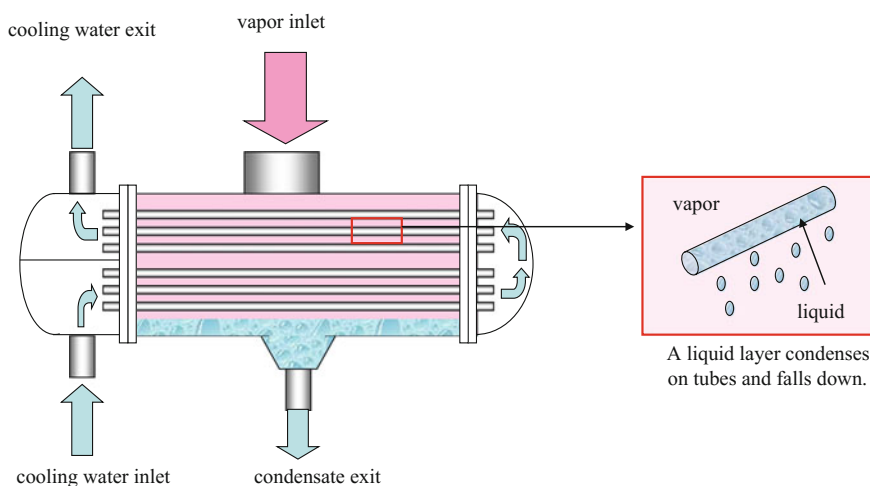
Tube in tube heat exchangers are also employed, constituted by two coaxial tubes with the inner one corrugated, as shown in Fig. 2.32, to increase the heat transfer area and to promote turbulence. Both heat transfer coefficient and self-cleaning capability increase in so doing. Different heat transfer capacities are available. These heat exchangers are often spiral windings shaped to reduce the occupied room.

Still to give some order of magnitude, few data related to this type of heat exchangers are supplied. For example Packless Industries [6] provides heat exchanger capacities in the range 1.76 kW (1/2 ton<sup>12</sup>)–105.5 kW(30 tons). The outer and inner tubes are, in general, respectively made by steel and copper. The volume of these spirally wound exchanger is evaluated as the one of a parallelepiped (like a box containing them) with sides a and b and height c. For lower capacities they may be a = 25 cm, b = 17 cm and c ranging from 8 to 10 cm. For higher capacities a = 64 cm, b = 51 cm, c = 56 cm or a = 90 cm, b = 31 cm, c = 31 cm. There are also the so called “trombone” heat exchangers, just shaped as

<sup>12</sup>Ton (ton of refrigeration) is an Anglo-Saxon unit: 1 ton = 3.517 kW.



**Fig. 2.32** Tube in tube heat exchanger with corrugated surface. On the bottom a spiral condenser (coil)



**Fig. 2.33** Shell and tube condenser

the musical instrument. The sides of the basis are larger than before, but  $c$  is in the order of 5 or 6 cm.

Shell and tube heat exchangers are employed as well. Figure 2.33 shows the scheme of a flooded condenser. Cooling water flows in tubes and refrigerant in the shell. Vapor enters the shell and condenses, in contact with cold tubes,. As an example Alfa Laval, [6], supplies heat exchangers for R407C and R134a, cooled by

**Table 2.7** Evaporation data for R134a

| Mass flow rate<br>(kg/m <sup>2</sup> s) | Temperature<br>(°C) | Heat transfer coefficient<br>(kW/m <sup>2</sup> K) | Vapor<br>quality |
|---|---------------------|--|------------------|
| 200–900                                 | –17.8               | 5–15   | –                |
| 400                                     | –18–5               | 8.5–9.0  | –                |
| 400                                     | /                   | 6.0–9.0  | 0.05–0.2         |
| 400                                     | –                   | 9.0  | >0.2             |

water coming from cooling towers, wells, rivers and lakes as well as from industrial processes, with condensing power between 60 and 1680 kW.

Heat exchangers are sized for nominal requirements. So that, for instance, vapor exiting the compressor is taken to subcooled liquid to the expansion valve. When refrigerant flow rate is lower than the nominal value, with the same cooling water flow rate, the region of subcooled liquid refrigerant widens.

As above mentioned, a liquid receiver is placed at condenser exit, aimed at collecting refrigerant in the case of maintenance and to allow for flow rate modulation, so that only liquid could flow into the lamination valve. Another tank can be located at evaporator exit (wet evaporators). As stressed before in dry evaporators only superheated vapor flows out of the heat exchangers. Thus, there is no danger for the compressor, but we lose heat transfer efficiency as variable temperature region of refrigerant exist in the final part of the evaporator. As a remedy to this a flooded evaporator is used, where the change of phase ends with no superheating. A gravity vapor separator is then placed before compressor suction.

Once more we can mention some technical data for the related heat transfer coefficient ranging from 2 to 7 kW/m<sup>2</sup>K for a corrugated plate evaporator. Similar value for condensing micro finned tube with tube diameters up to 9.40 mm [7].

The following data (Table 2.7) for R134a can be roughly obtained by diagrams [8].

Air cooled heat exchangers are basically composed by finned tube bundles. Refrigerant flows in the tubes, often made of copper, and air is blown by fans. Air flow rate can be varied both stepwise (usually three steps) or continuously, by an inverter, according to the requested load.

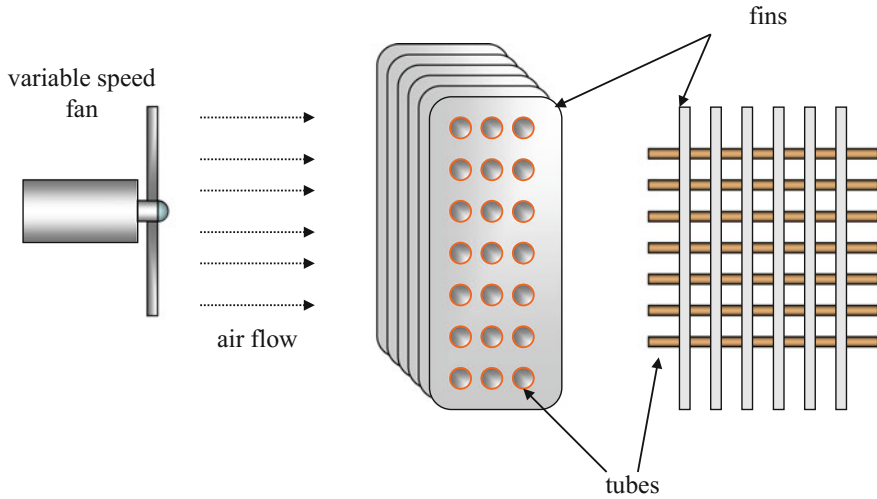
A typical basic scheme is shown Fig. 2.34.

This is the most commonly adopted type for low capacity heat exchangers. Much more complex devices are employed for larger machines, anyhow working with the same operation principle.

Whatever the size, these exchangers may undergo frost formation.

### 2.5.1 The Effect of Outside Air Humidity and Frosting

The external heat exchanger is generally sized referring to summer conditions, roughly with a temperature difference of 12–15 °C between the flowing refrigerant



**Fig. 2.34** Heat exchange coil

and the outdoor air. Therefore if the air temperature is 35 °C, condensation takes place at around 50 °C, and only sensible heat is exchanged.

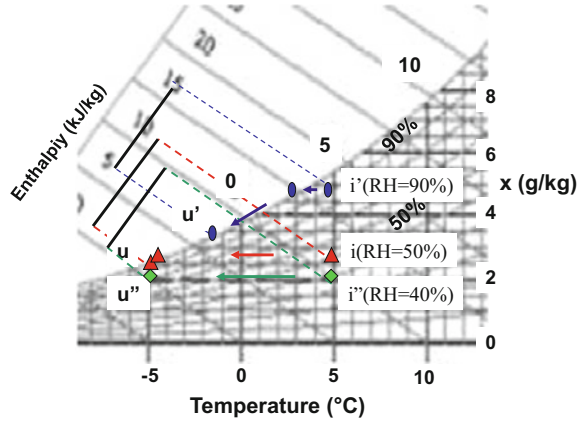
In winter air flowing to the fins, with an inlet temperature  $T_i$  and specific heat  $c_{pA}$ , transfers a heat  $q$  (J/kg), equal to the difference between the inlet enthalpy,  $h_i$ , and the outlet one,  $h_u$ . Then we have:

$$\begin{aligned}
 q &= h_i - h_u = c_{pA}(T_i - T_u) + r'(x_i - x_u) \\
 T_u &= T_i - \frac{[q - r'(x_i - x_u)]}{c_{pA}} \\
 \text{se } x_i &= x_u \\
 T_u &= T_i - \frac{q}{c_{pA}}
 \end{aligned}$$

From the above equations we infer,  $T_i$  and  $q$  being the same, the outlet air temperature  $T_u$  is lower for a pure sensible heat exchange ( $x_u = x_i$ ) than for a transformation with a latent heat exchange ( $x_u \neq x_i$ ). Figure 2.35 shows, for a given  $T_i = 5$  °C and  $q = 10$  kJ/kg, three transformations starting from different inlet relative humidities (RH): 90% (blue circles), 50% (red circles) and 40% (green circles). In the usual operating conditions we can affirm that no condensation occurs for a relative humidity lower than 50%, with such air temperature (5 °C).

*Example 2.3* To go into more details consider the case with an outdoor air temperature  $T_i = 5$  °C. By extracting the numerical values from the graph of Fig. 3.32a, we can roughly say that  $x_i = 2.8$  g/kg and a dew temperature  $T_d = -4.6$  °C at a relative humidity RH = 50% and  $x_i = 4.8$  g/kg and  $T_d = 3.0$  °C with RH = 90%.

**Fig. 2.35** Transformations on an evaporating coil with different values of outdoor air humidity. Symbols triangle  
 RH = 50%, ellipse  
 RH = 90%, rhombus  
 RH = 40%



Thus we get ( $c_{p,A} = 1.0 \text{ kJ/kgK}$  and  $r = 2500 \text{ kJ/kg}$ )

RH = 90%;  $h_i' = 17.0 \text{ kJ/kg}$

RH = 50%;  $h_i = 13.0 \text{ kJ/kg}$

RH = 90%;  $h_u' = 7.0 \text{ kJ/kg}$

RH = 50%;  $h_u = 3.0 \text{ kJ/kg}$ .

With RH = 90% we respectively have a sensible and latent heat exchange equal to 2.0 and 8.0 kJ/kg, while at RH = 50% the heat exchange is essentially sensible.

The evaporation temperature is normally assumed 4 °C lower the outflowing cooling air so that the relative humidity improves COP at temperatures above that of defrost cycles' start up (e.g., just 5 °C we referred to in the previous example). At lower temperatures the air humidity causes more demanding defrost cycles, decreasing the performance of a heat pump as much as higher its value is.

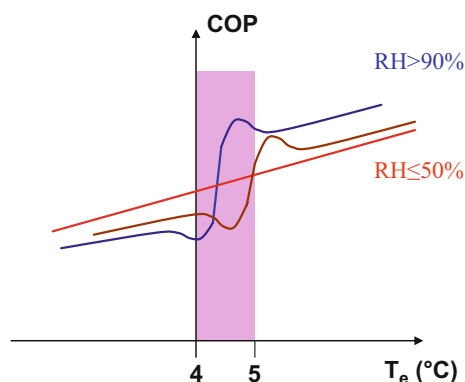
Therefore a COP trend versus outside air temperature similar to the one sketched in Fig. 2.36 has to be expected. The red and the blue curves respectively refer to RH = 50% and RH = 90%, while the brown curve refers to an intermediate RH value.

As it is clear from the above discussion, the knowledge of outlet air temperature is basic to know how far we are from frosting.

The formula below can be used to this aim:

$$T_u = T_i - 0.8 \frac{P_t - P_c}{V}$$

**Fig. 2.36** COP versus outdoor temperature with different relative humidities (RH)



With the following meaning of symbols:

- $P_t$  thermal power kW, according to Eurovent,<sup>13</sup>
- $P_c$  compressor(s) power in kW, according to Eurovent. Eurovent certifies the total absorbed power  $P_A$ ; to obtain the compressor power we need to subtract the power of fan(s).
- $V$  volumetric cooling air flow rate in  $\text{m}^3/\text{s}$ .

*Example 2.4* Let us refer to a heat pump with the data below:

$P_t = 5.28 \text{ kW}$ ;

$P_A = 1.64 \text{ kW}$ ;

$V = 2350 \text{ l/s}$ ;

Fan power  $0.12 \text{ kW}$ .

Consequently the cooling air temperature difference (outlet minus inlet temperatures) is  $1.28 \text{ K}$ .

The normal reference conditions are: air temperature  $7^\circ\text{C}$  and  $\text{RH} = 87\%$ .

With the given values  $T_u = 5.72^\circ\text{C}$  and the dew temperature can be evaluated as  $4.6^\circ\text{C}$ .

In winter ice can freeze over, both on tube-fins and on tubes themselves, owing to outside air relative humidity and low temperature. This phenomenon takes place with an outdoor temperature even higher than  $5$  or  $6^\circ\text{C}$  and a humidity exceeding  $60\%$ . At the very beginning a thin ice layer forms. At this stage the formed ice is a good thermal conductor, increases the heat exchange area, and lowers the flow

<sup>13</sup>Eurovent is the Europe's Industry Association for Indoor Climate (HVAC), Process Cooling, and Food Cold Chain Technologies. Its associates (more than a thousand companies) belong to Europe, the Middle East and Africa.

**Table 2.8** Defrost typical data

| Outdoor temperature (°C) | Relative humidity (%) | Duration (minutes) |
|--------------------------|-----------------------|--------------------|
| 0                        | 70                    | 220                |
|                          | 80                    | 100                |
|                          | 90                    | 50                 |
|                          | 100                   | 30                 |
| 5                        | 70                    | 220                |
|                          | 80                    | 100                |
|                          | 90                    | 50                 |
|                          | 100                   | 30                 |

section (gaps between fins) increasing air velocity, then, it enhances heat transfer rate. The additional ice layers forming afterwards are porous and contain air. Therefore they are insulating and deteriorate the heat exchange. Consequently evaporator efficiency increases at first, but diminishes afterwards.

The insulating ice, thus, reduces heat pump performances. It is fundamental to take action at the right time and for a proper period to remove all the grown ice.

The best way to set the time when starting defrost is to provide the heat pump with sensors detecting: air temperature, its flow rate through the finned tubes and, at the same time, the pressure of the refrigerant. This way, defrost starts at the right moment and lasts just the suitable time, as defrost cycles effects on heat pump performances are far from being negligible.

Just to give an idea, some data are provided, concerning the time interval between two consecutive defrosts, depending on outdoor temperature and relative humidity in Table 2.8. The table reports just indicative values of this time intervals, and the actual ones should be set up on a case-by-case basis.

The defrosting technique may consist either in an electric resistance which switches on when fin temperature approaches 0 °C<sup>14</sup> or, more commonly, thermal cycle reversal. This means that the unit switches over to the cooling mode and the outdoor coil (evaporator) becomes the hot condenser. In doing so, some discomfort to users is caused.

The process takes place according to the following stages:

- switch off of the outdoor coil fan, through a dedicated relay.
- Cycle reversing valve switching to the cooling mode.
- Switch on of an auxiliary heating source for the indoor environment, if available.

In any case an amount of ice that reduces the cooling air flow rate more than 50% of the nominal value is not acceptable, as it might impair the compressor. On one hand equipment safety would suggest frequent defrosts, but economy and machine viability require performing few defrosting cycles.

<sup>14</sup>Or at a certain level of obstruction of the fin gap, due to frost formation.



Several aspects have to be accounted for to optimize defrost start up.

- A first control can be performed on air pressure within the coil, (differential pressure between inlet and outlet). When this value exceeds a given set point value, the process starts. This type of systems reacts to low pressure difference, so that they might be activated by a wind burst. Therefore a time delay has to be introduced to verify the permanency of such a pressure drop increase, before starting the defrost. Even debris and leaves can cause an improper system action.
- It is also possible to refer to temperature differences. This method is based on the fact that the usual temperature difference between outdoor and evaporator temperatures varies in the range 3–9 °C. As ice builds up, this difference increases. Defrost starts when a set threshold is exceeded.
- As both the afore said methods revealed not to be always reliable, and also somehow costly, at first many manufacturers decided to use a timer in residential applications. So the process started at given time intervals. This is a very simple method and was the most widespread at least as long as electronics was introduced. The timer was coupled to a thermostat measuring air temperature at coil exit generally set at 3 °C. If air is cooler than this for a given period, say 30 min, defrost starts, otherwise it does not. Defrost ends when evaporator temperature achieves a preset value by manufacturers. This method, timer + thermostat, is the most used in residential buildings also because of its low cost. A further control based on the pressure difference mentioned above may be added so that defrost startup is also influenced by this, when the related increase is about 100 Pa.

An additional method consists in injecting superheated vapor from the compressor into the evaporator through a dedicated defrosting valve. This is aimed at preventing the indoor environment from being cooled, even if some power is anyway subtracted from it. Other solutions use tanks where thermal energy can be stored and then released to coils, as those using melting salts or ethylene glycol exchanging heat with the condenser de-superheating stage.

In general defrost can affect energy consumption by more than 10%, depending on the adopted solution. In case of absorption heat pumps some hot fluid (e.g., ammonia), coming from the generator, can be diverted to the outdoor coil, without any cycle reversal. Such a reversal does not even occur in endothermic engine driven heat pumps. Before ending this paragraph, some features of this type of coils are provided, see Table 2.9.

## 2.6 Economizer and Vapor Injection

A way to save energy in the case of a large temperature difference between thermal sources, i.e., large pressure ratio, is injecting vapor into the compressor at an intermediate pressure.

**Table 2.9** Heat pump coils (R410A) for residential use

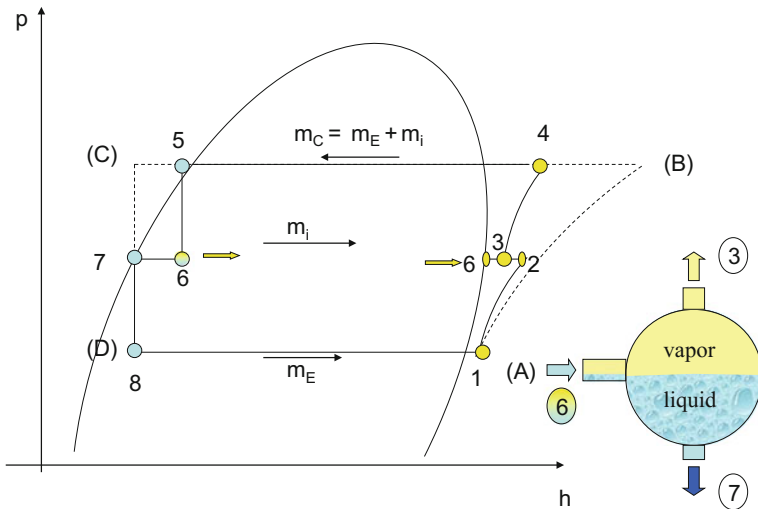
|   |   |                     |                       |                       |
|---|---|---------------------|-----------------------|-----------------------|
| Nominal cooling power<br>(1) (min/max) kW | 4.13<br>(1.80/5.00)   | 6.49<br>(3.00/8.20) | 8.20<br>(3.70/10.0)   | 10.51<br>(4.0/13.10)  |
| Nominal power input (1) kW                | 1.33  | 2.08                | 2.65                  | 3.39                  |
| E.E.R. (1) W/W                            | 3.11  | 3.12                | 3.10                  | 3.10                  |
| E.S.E.E.R. W/W                            | 3.43  | 3.49                | 3.41                  | 3.48                  |
| Nominal cooling power<br>(2) (min/max) kW | 5.72<br>(2.30/0.20)   | 8.93<br>(3.70/0.90) | 12.36<br>(4.60/13.20) | 14.00<br>(6.00/16.00) |
| Nominal power input (3) kW                | 1.44  | 2.27                | 2.98                  | 3.64                  |
| E.E.R. (2) W/W                            | 3.98  | 3.93                | 4.15                  | 3.85                  |
| Nominal heating power<br>(3) (min/max) kW | 5.48<br>(2.10/0.80)   | 8.43<br>(3.50/9.30) | 11.81<br>(4.40/12.60) | 13.38<br>(5.60/14.80) |
| Nominal power input (3) kW                | 1.65  | 2.55                | 3.45                  | 4.13                  |
| C.O.P. (3) W/W                            | 3.32  | 3.30                | 3.42                  | 3.24                  |
| Nominal heating power<br>(4) (min/max) kW | 5.77<br>(2.40/6.50)   | 9.06<br>(4.00/10.0) | 12.40<br>(4.70/13.40) | 14.16<br>(6.30/16.40) |
| Nominal power input (4) kW                | 1.39  | 2.21                | 2.95                  | 3.45                  |
| C.O.P. (4) W/W                            | 4.15  | 4.11                | 4.21                  | 4.15                  |
| 1.  | Cooling: outdoor air temperature 35 °C; inlet/outlet water temperature 12/7 °C                |                     |                       |                       |
| 2.  | Cooling: outdoor air temperature 35 °C; inlet/outlet water temperature 23/18 °CC              |                     |                       |                       |
| 3.  | Heating: outdoor air temperature 7 °C d.b. 6 °C w.b.; inlet/outlet water temperature 40/45 °C |                     |                       |                       |
| 4.  | Heating: outdoor air temperature 7 °C d.b. 6 °C w.b.; inlet/outlet water temperature 30/35 °C |                     |                       |                       |

*Note* Data declared according to UNI EN 14511:2011. The performance data shown in the table refer to units without options and/or accessories and could be subject to change. Attention: for antifreeze unit version, for lowest ambient temperature 5 °C, you must add a suitable quantity of antifreeze additives

In Fig. 2.37, the classical cycle is drawn in the pressure enthalpy plane, with dotted lines (points marked by capital letters), while that with continuous lines (point marked by numbers) represents a cycle with vapor injection.

At the end of condensation (4→5), the refrigerant is sent to a first expansion valve (5→6), at its exit vapor<sup>15</sup> is separated from liquid in a separator, at pressure  $p_7$ , and forwarded to the compressor (6→3) at the beginning of the second stage of compression. Liquid goes to a second expansion valve to enter the evaporator.

<sup>15</sup>Dry vapor from separator mixes with vapor coming from the first compression stage, point 3 in the figure.



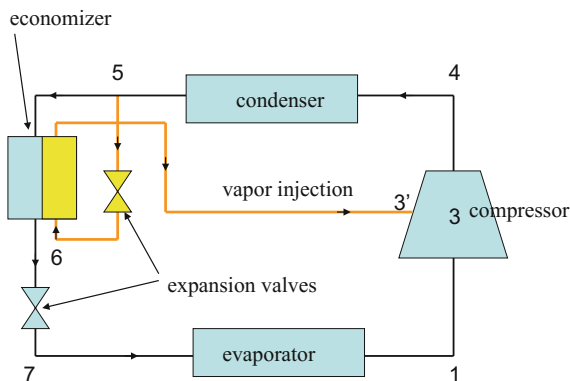
**Fig. 2.37** Cycle with vapor injection

The following relations hold:

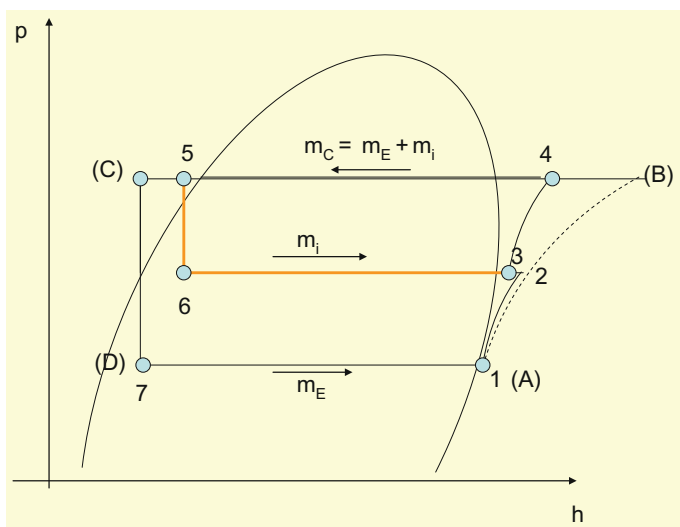
$$\begin{aligned}
 Q_C &= m_C(h_5 - h_4) & Q_E &= m_E(h_8 - h_1) \\
 m_C &= m_E + m_i \\
 L_{1,3} &= m_E(h_1 - h_2) \\
 L_{3,4} &= m_C(h_3 - h_4) \\
 COP &= \frac{Q_C}{L_{1,3} + L_{3,4}} = \frac{(h_5 - h_4)}{(h_3 - h_4) + \frac{m_E}{m_C}(h_1 - h_2)} \\
 h_3 &= \frac{m_i}{m_C}h_6 + \frac{m_E}{m_C}h_2
 \end{aligned}$$

where subscripts C, E and i respectively indicate: condenser, evaporator and injection. Vapor injection produces an intermediate cooling that lowers the work of compression. The reduction of compression work can be easily evaluated by comparing the single compression stage cycle with the one obtained by vapor injection.

Fluid exiting from the condenser can also undergo a double expansion, a first one between the whole cycle pressure difference (between condenser and evaporator) and a second one between the condenser pressure and an intermediate value, as sketched in Fig. 2.38. A given amount of refrigerant (primary fluid) flows to a first lamination valve (path 5-7) after passing through a heat exchanger, economizer, and goes to evaporator. In this heat exchanger primary fluid transfers heat to another amount of fluid flowing (secondary fluid) along path 5-6-3'. This secondary fluid is laminated to an intermediate pressure in a second valve and reaches the



**Fig. 2.38** Cycle with economizer



**Fig. 2.39** Cycle with economizer in the pressure-enthalpy plane

compressor as superheated vapor. This way, primary fluid is further subcooled and secondary fluid heats up, then mixing with vapor of the first compression stage (point 3).

Figure 2.39 shows the related cycle in the pressure-enthalpy plane.

Still to make an example: for a 8 kW heating power and 6 kW cooling power heat pump, using R407C, with condensation temperature and pressure of 50 °C and 22 bar and evaporation at -7 °C and 4 bar Copeland [9] provides a scroll compressor (Model ZH09KVE-TFD) with a total flow rate of 29.7 g/s and a vapor injection flow rate of 9.70 g/s, at an intermediate pressure of 5.97 bar. Vapor

injection is also employed in screw compressor with a COP claimed increase around 20%.

This technology is also implemented for sanitary water supply at about 50 °C, even with outdoor temperatures below 0 °C.

In addition scroll compressors exist on the market which can bear liquid, thus allowing for saturated vapor injection, named wet vapor injection, instead of dry or superheated vapor. In so doing the operation range of compressors can be widened, but it is generally fixed a top value of time duration of such a type of injection (e.g., 2000 h). The use of wet vapor injection is aimed at limiting the discharge temperature, so that it does not exceed a safety value, say around 140 °C. The effect of vapor injection on compressor operating range has been already shown in Fig. 2.25.

## 2.7 The Four Way Reversing Valve

The scheme of a cycle reversing valve for reversible heat pumps is drawn in Figs. 2.40 and 2.41. Four ports are placed on it. On top we have the port where compressor discharge fluid flows in, high pressure port. Of the three ports placed on

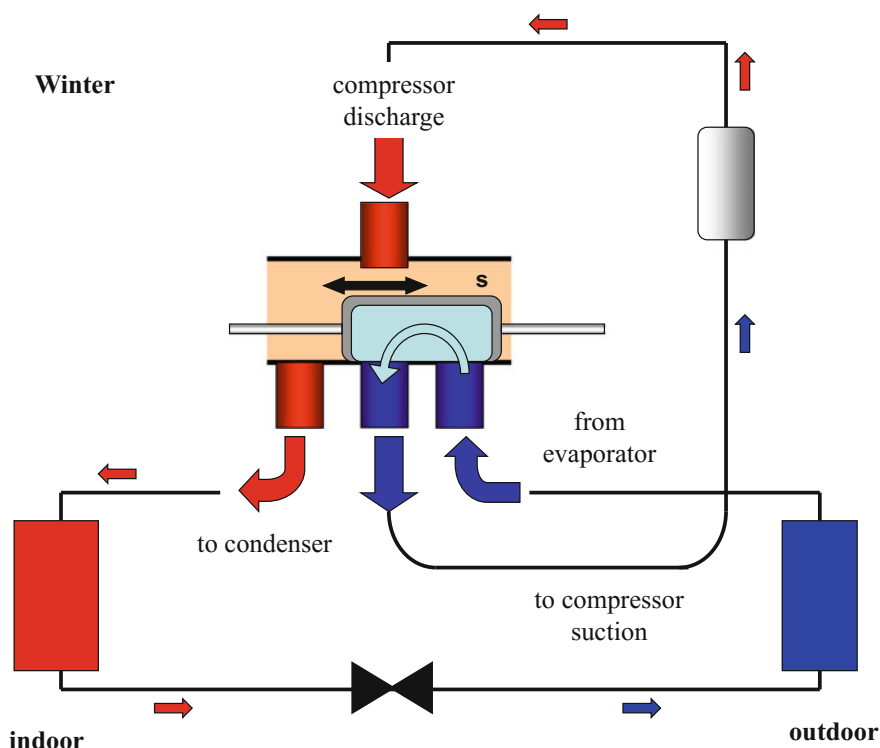
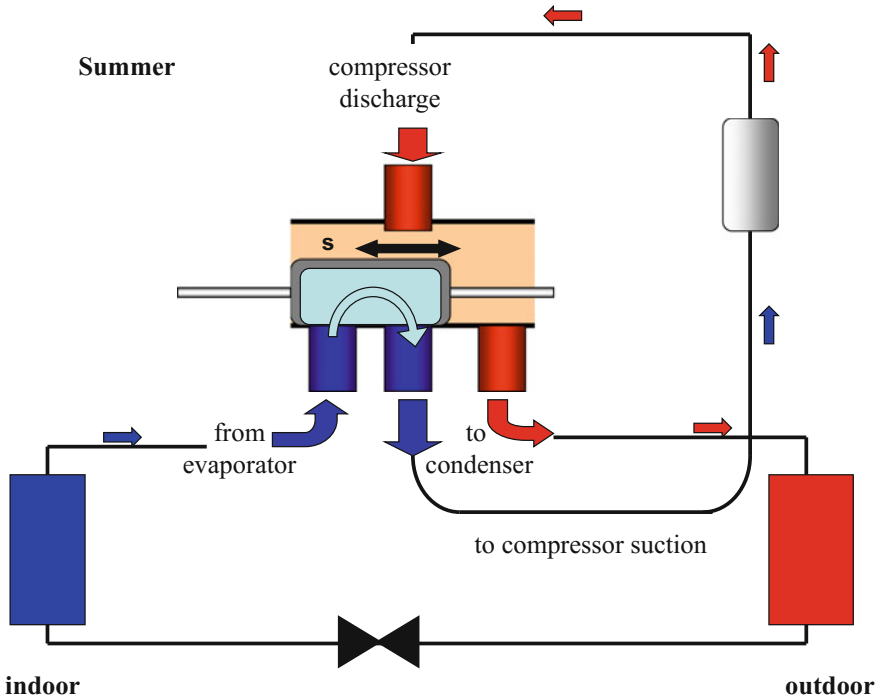


Fig. 2.40 Reversing valve configuration in winter operation mode



**Fig. 2.41** Reversing valve configuration in summer operation mode

the lower side the central one send fluid to the compressor suction. Cycle inversion is obtained by a slide S that puts into contact these ports by twos, moving right and left. Its movement is caused by the refrigerant itself flowing through dedicated capillary tubes. This flow is controlled by a valve activated by an electric coil. If the valve coil is fed, the winter mode of operation is active (Fig. 2.40), while, when it is not, the summer mode takes place. This is done both for seasonal operation change and for defrosting.

## 2.8 Engine Driven Heat Pumps (GHP)

In this type of heat pumps, usually addressed as GHP (Gas Heat Pumps), the compressor is driven by a gas engine, instead of the more commonly used electric motor. Beyond the mechanical work delivered to the compressor these machines recover the engine exhaust heat according two different ways: direct and indirect heat recovering.

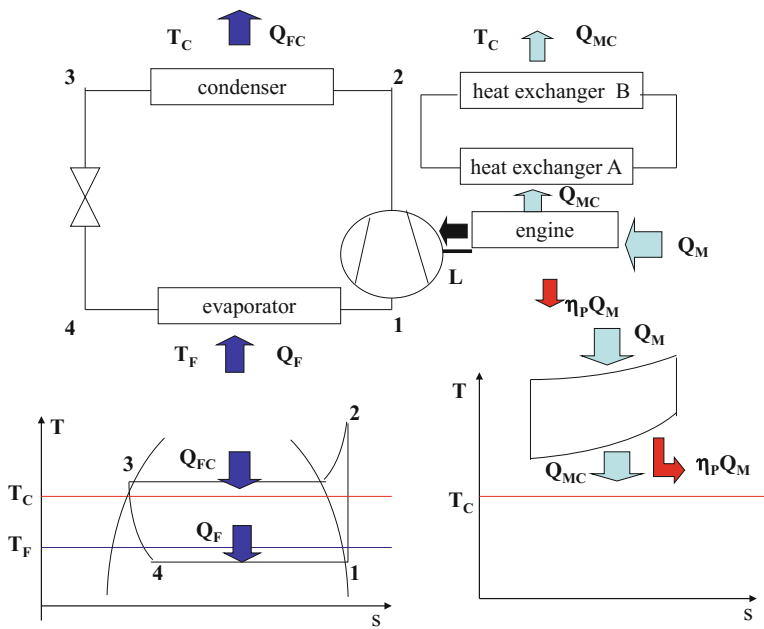


Fig. 2.42 Engine driven heat pump (GHP)

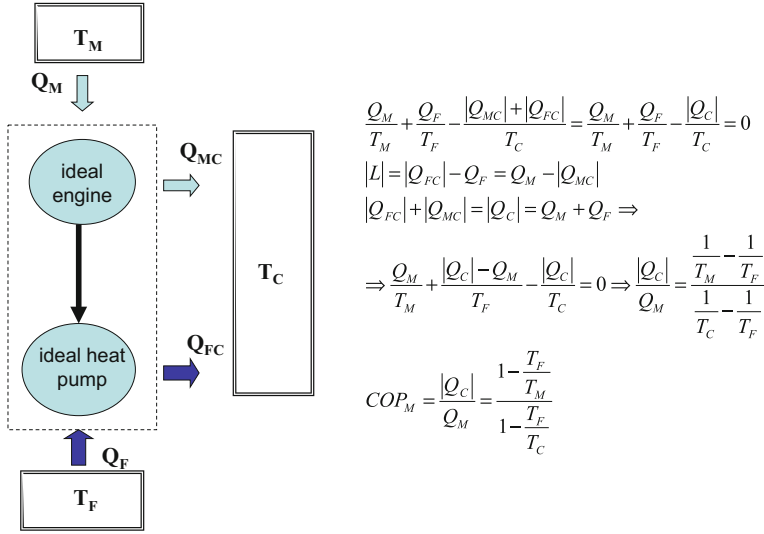
The direct heat recovering uses the engine cooling water either for indoor environment heating or for sanitary water production. Figure 2.42 shows a typical scheme with a diesel engine. The used symbols have the following meaning:

|          |  |
|----------|--|
| $T_C$    | Hot heat source temperature                          |
| $T_F$    | Cold source temperature                              |
| $T_M$    | Equivalent temperature of “engine source”            |
| $Q_{FC}$ | Amount of heat released by heat pump to cold source  |
| $Q_F$    | Heat exchanged with the cold heat source             |
| $Q_M$    | Heat supplied to engine by combustion process, $T_M$ |
| $Q_{MC}$ | Heat supplied to hot source by regenerator           |

The system, heat pump plus engine, interacts with three thermal sources. In fact, at its boundary, it only exchanges heat, while work  $L$  is an internal mechanical exchange between engine and compressor. This work is related to the supplied combustion heat, through the engine thermodynamic efficiency  $\eta_M$ . In other words:

$$|L| = |Q_{FC}| - Q_F = \eta_M Q_M$$

If  $e_R$  is the efficiency of exhaust heat recovery system, i.e., the amount,  $(1 - \eta_M) Q_M$ , of heat that can be recovered by cooling the engine we obtain that



**Fig. 2.43** Basic scheme of a GHP with direct heat recovery system

$Q_{MC} = e_R(1 - \eta_M)Q_M$ , while the heat released to outdoor environment is  $(1 - e_R)(1 - \eta_M)Q_M$ . Thus the heat exchanged with the hot source is:

$$Q_C = Q_{FC} + Q_{MC} = Q_{FC} + e_R(1 - \eta_M)Q_M$$

where the second term on the right hand side of the above equation can be used to produce hot sanitary water, in summer.

Therefore  $Q_C$  is the “useful” heat we can obtain from an engine driven heat pump in winter.

By applying the first and second Principle of Thermodynamic for ideal conditions<sup>16</sup> (Carnot cycle), we obtain:

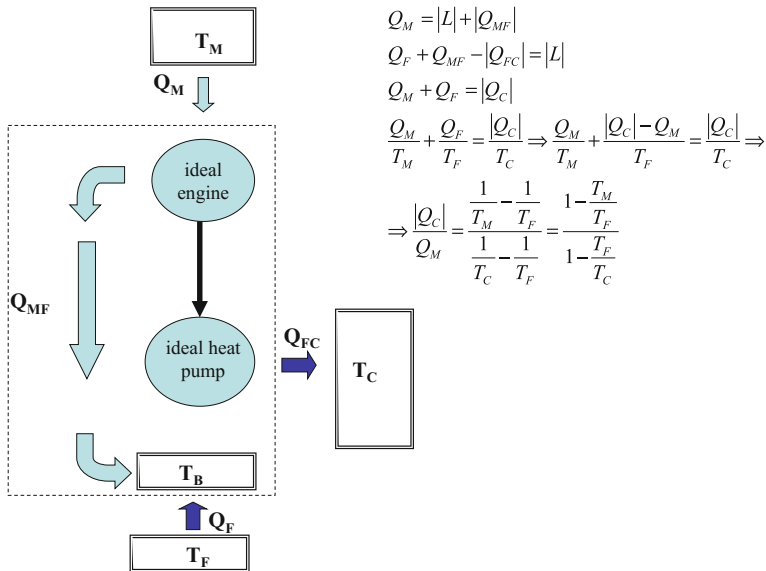
$$\begin{cases} -\frac{|Q_C|}{T_C} + \frac{Q_F}{T_F} + \frac{Q_M}{T_M} = 0 \\ -|Q_C| + Q_F + Q_M = 0 \end{cases}$$

The coefficient of performance (see Fig. 2.43),  $COP_{MI}$ , in winter is:

$$COP_{MI} = \frac{|Q_C|}{Q_M} = \frac{\frac{1}{T_M} - \frac{1}{T_F}}{\frac{1}{T_C} - \frac{1}{T_F}} = \left(1 - \frac{T_F}{T_M}\right) \frac{1}{1 - \frac{T_F}{T_C}} = \left(1 - \frac{T_F}{T_M}\right) COP_{EI}$$

<sup>16</sup>In this case we can also suppose that the whole exhaust heat,  $Q_{CM} = (1 - \eta_{Carnot})Q_M$ , could be recovered.





**Fig. 2.44** GHP with indirect heat recovery

where  $COP_{EI}$  is an electric heat pump (EHP), working between the same heat sources.

$T_M$  is usually assumed 1000 K and, therefore, it comes out that  $COP_M < COP_E$ .

Another way we can follow with a GHP is the so called indirect recovery. In this configuration, the engine exhaust heat is employed to increase the temperature of the source exchanging with the evaporator, in winter. So COP increases. The related scheme is reported in Fig. 2.44. The evaporator thermal source has a temperature  $T_B$  instead of  $T_F$  ( $T_B > T_F$ ) and refrigerant follows the cycle 1234 instead of the initial 1'234', increasing the heat exchanged by the evaporator and lowering the compression power.

In a fully reversible condition (referring to Fig. 2.44) we have:

$$\begin{aligned}
 Q_M &= L + |Q_{MF}| \\
 Q_{ev} &= Q_F + Q_{MF}
 \end{aligned}$$

where  $Q_{MF}$  is the power recovered by cooling the engine ( $>0$ ),  $Q_F$  the one obtained by the cold source ( $>0$ ) and  $Q_{ev}$  the power supplied to the evaporator ( $<0$ ).

Where  $Q_{MF}$  is the heat recovered by engine cooling ( $>0$ ),  $Q_F$  the g from the cold source ( $>0$ ) and  $Q_{ev}$  the supplied to the evaporator ( $<0$ ).

The coefficient of performance is:

$$COP'_M = \frac{Q_C}{Q_M} = \frac{Q_F + Q_{MF} + L}{Q_M}$$

$$\frac{Q_F + Q_{MF}}{T_B} - \frac{|Q_C|}{T_C} = 0$$

Still, with reference to a winter ideal condition, where engine and heat pump respectively follow a direct and an inverse Carnot cycle and with an ideal regenerator, the results illustrated below hold.

The mechanical work engine provides to the heat pump is  $L = \eta_M Q_M$ , where  $\eta_M$  is the Carnot efficiency:

$$\eta_M = 1 - \frac{T_C}{T_M}$$

$$Q_{MC} = (1 - \eta_M) Q_M$$

While the heat supplied to indoor environment is:

$$Q_{FC} = \left( \frac{1}{1 - \frac{T_F}{T_C}} \right) L = \left( \frac{1}{1 - \frac{T_F}{T_C}} \right) \eta_M Q_M$$

It clearly comes out that heat provided to indoor environment is made up of an amount,  $Q_{FC}$ , depending on outdoor temperature,  $T_F$ , and another one,  $Q_{MC}$ , independent of this temperature. Therefore:

$$Q_C = Q_{FC} + Q_{MC} = \left( \frac{1}{1 - \frac{T_F}{T_C}} \right) \eta_P Q_M + (1 - \eta_P) Q_M$$

$$= [(COP_{\text{Carnot,inv}} - 1) \eta_{\text{Carnot,dir}} + 1] Q_M$$

This relation, once again, stresses that part of this heat does not depend on the outdoor climate. This part is poor in ideal conditions, due to the high efficiency of a Carnot engine, but this is not in actual cases. Let us refer to the example below.

*Example 2.5* Refer to the ideal condition, first. Suppose that  $T_F = 5^\circ \text{C}$  (278 K),  $T_C = 20^\circ \text{C}$  (303 K) and  $T_M = 1000 \text{ K}$ . The Carnot cycle efficiency is:

$$\eta_M = 0.697$$

and

$$COP = 12.12$$

Therefore:

$$Q_C = 8.411Q_M + 0.303Q_M$$

And the percent of recovered energy is 3.60%.

In a real case, with the same source temperatures, consider  $\eta_M = 30\%$  and a recovery efficiency equal to 35% of the power supplied to the engine ( $Q_M$ ), i.e., 50% of the heat dispersed by the engine. With  $COP = 3.5$  we obtain:  
 $Q_C = 1.05Q_M + 0.35Q_M$ .

In conclusion the amount of recovered energy is around 35%.

It is easy to see that, heat supplied to indoor environment being equal, this heat pump draws a lower power from the outdoor thermal source, than an electric heat pump does. Besides, GHP is less sensitive to temperature changes of this latter source. In summer the heat balance is as well influenced by what has been previously said. Anyway the exhaust heat can be used for producing sanitary hot water.

To have some more data about machines actually available in the market, the interested reader can refer to [10].

## 2.9 Carbon Dioxide Heat Pumps

This type of compression heat pumps is dealt with separately owing to the peculiar refrigerant used. In fact, carbon dioxide critical point is characterized by a temperature  $T_{cr} = 30,978\text{ }^\circ\text{C}$  and a pressure  $p_{cr} = 73,773\text{ MPa}$  and its boiling point at atmospheric pressure by  $-45.56\text{ }^\circ\text{C}$ .

These features allows for their adoption in cold climates ( $-25\text{ }^\circ\text{C}$ ), but, at the same time, they imply the use of a hyper-critical cycle in most applications. A gas cooler is placed downstream the compressor, instead of the usual condenser, where vapor is isobarically cooled with large temperature changes. These cycles typically work with pressure values ranging from 30 bar at the evaporator up to 130 bar of the gas-cooler. This latter pressure has to be fixed by a dedicated controller, as it is no longer connected to temperature, as it occurs in phase change processes. As a general rule the optimal value,  $p_{opt}$  (bar) is connected to the exit temperature,  $T_{ex}$  ( $^\circ\text{C}$ ), from the gas cooler by the equation:

$$p_{opt.} = 2.6T_{usc.} + 8$$

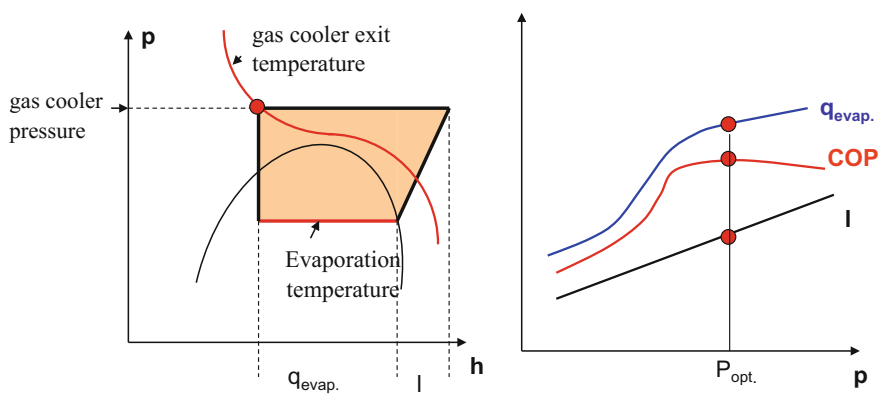
Holding in the range  $38 \leq T_{ex.} \leq 53\text{ }^\circ\text{C}$ .

As a consequence of this, there are some basic difference between the use of organic refrigerants, employed with sub-critical cycles and carbon dioxide working in super-critical cycles. They are stressed in Table 2.10 (Fig. 2.45):

These machines are often used for sanitary water production and/or for high temperature water tanks. They can easily apply to retrofit “old” heating systems

**Table 2.10** Differences between organic refrigerant and CO<sub>2</sub> cycles

| Cycle differences                | Usual refrigerant                              | CO <sub>2</sub>  |
|----------------------------------|--|--|
| High pressure transformation     | Condenser at constant temperature and pressure | Gas cooler at constant pressure and large temperature change |
| High pressure                    | 10–40 bar                                      | 90–130 bar   |
| Low pressure                     | 2–9 bar  | 25–50 bar  |
| Compressor discharge temperature | <95 °C   | Up to 140 °C   |
| Lamination                       | Super heating control                          | Gas cooler pressure control                                  |
| High pressure (with shut down)   | Depending on temperature                       | Pressure control device                                      |



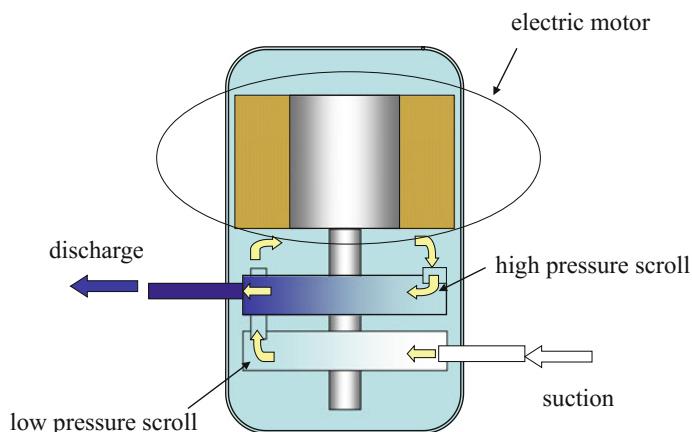
**Fig. 2.45** To the *left* scheme of a hyper critical cycle. To the *right* qualitative trends of thermal power ,  $q_{\text{evap}}$ , mechanical power,  $I$ , and COP versus compressor discharge pressure

using high temperature radiators. This is the case of inner cities to which not enough attention is devoted, instead focusing mainly on new buildings.

Now let us briefly describe the main differences introduced by the use of carbon dioxide.

**2.9.1    Compressor**

The compression ratio is lower, and the clearance volume re-expansion is reduced. Therefore, suction valve opening can be hastened and volumetric efficiency improved. CO<sub>2</sub> leakages constitute the major cause of efficiency degradation and



**Fig. 2.46** Two-stage scroll compressor

must be carefully addressed. Rotary compressors have been added, for small loads,<sup>17</sup> to reciprocating compressors, generally used for large loads,

For commercial, sea and railways refrigeration, Dorin [11] provides a vast range of semi-hermetical reciprocating compressors.

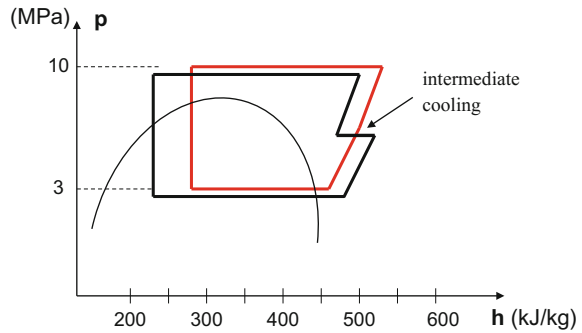
Copeland [12] produces scroll compressors with sub critical cycles for low temperatures (evaporation between  $-50$  and  $-25$  °C and condensation between  $-20$  and  $5$  °C) coupled with an upper HFC stage. Double stage scroll compressors have been designed for high pressures. The related scheme is shown in Fig. 2.46 and works as follows: fluid exiting the evaporator flows to a first stage, where it is compressed up to an intermediate pressure value, then it is discharged into compressor shell and reaches the second stage, undergoing a further compression. Both weight and size are lowered in doing so, as the compressor shell stands “only” an intermediate pressure value. In addition leakage losses are reduced as the compression ratio is splitted in two stages. Efficiency increases of 15 and 30% are reported respectively for compression ratio of 2 and 4, by choosing a double stage compressor instead of a single stage one. The related cycle is shown in Fig. 2.47.

## 2.9.2 Gas Cooler

Tube in tube, plate, micro channel, or plate with micro channels heat exchangers are used as gas coolers. Plate heat exchangers are in particular employed for sanitary water production. While cooling, carbon dioxide undergoes a large temperature

<sup>17</sup>A brochure by Sanyo (SANYO CO<sub>2</sub> Heat Pump Water Heater pdf) reports a scroll compressor with a total length of 217 mm for a 4.5 kW heat pump, dedicated to domestic use.

**Fig. 2.47** CO<sub>2</sub> sub-critical cycles

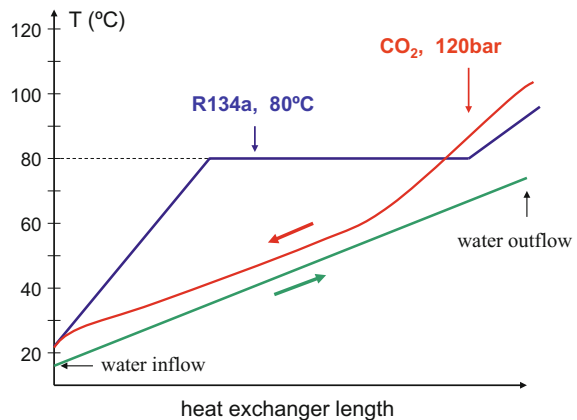


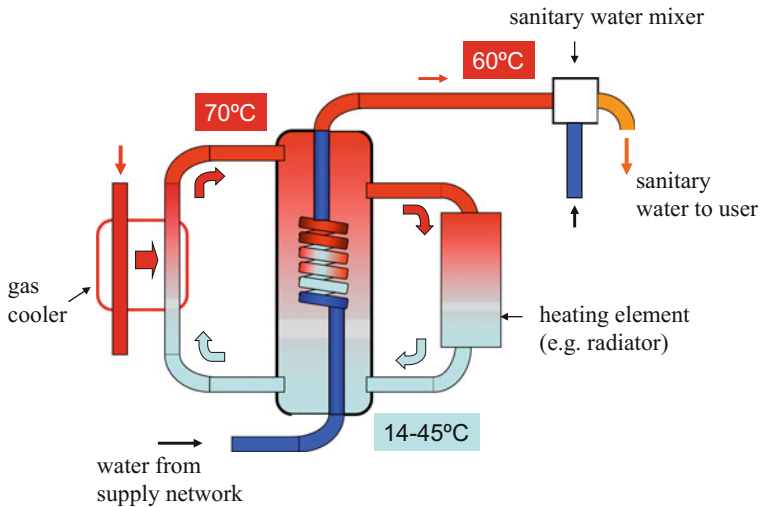
change with a significant change of thermo-physical properties. As a result the heat transfer coefficient varies within a rather wide range. For example it is 2000 W/m<sup>2</sup>K at compressor discharge with a pressure of 80 bar and a temperature of 80–85 °C, then increasing up to 13,500 W/m<sup>2</sup>K [13]. This increase reduces with increasing pressure (e.g., at 100 bar it “only” triples).

As a consequence the log mean temperature difference (LMTD) method is no longer reliable to size the heat exchanger, and some more accurate technique must be used. The temperature difference between exiting (from heat exchanger) carbon dioxide and entering cooling fluid is minimized to maximize heat exchange efficiency.

Figure 2.48 shows the temperature trends achievable producing sanitary water by a gas cooler or a traditional R134a condenser. It clearly comes out that the temperature difference between heating fluid and water is much smoother, thus more effective, in the case of carbon dioxide. Incidentally we recall that the percent of energy consumption in residential buildings to produce sanitary water is around 20% of the total heating consumptions, and that a water temperature up to 90 °C can be obtained by CO<sub>2</sub> heat pumps. A scheme of a Sanyo heat pump for combined

**Fig. 2.48** Comparison among CO<sub>2</sub>, R134a and sanitary water temperature trends versus heat exchanger length





**Fig. 2.49** Ambient heating and instantaneous production of sanitary water

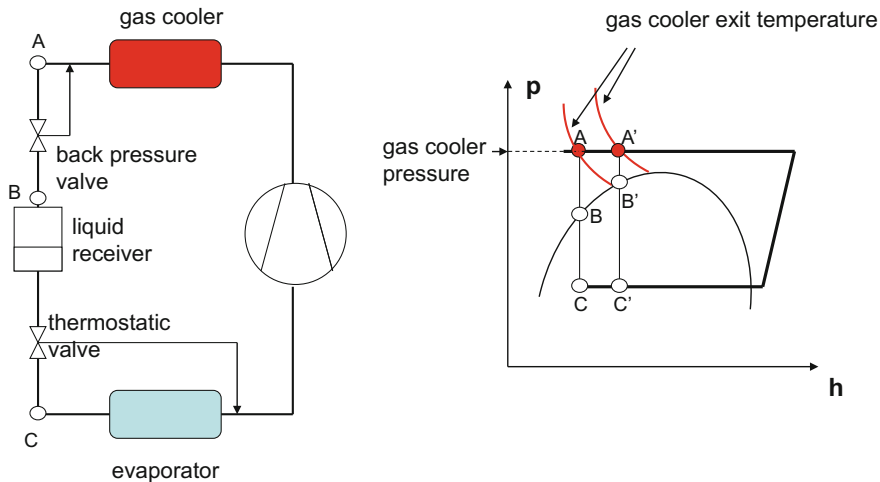
heating and sanitary water production is sketched in Fig. 2.49. A water tank heated up by the gas cooler is proposed. Water is taken away from this tank for house heating, so using the water mass of this accumulator to smooth the effect of occurring transients. Water from supply network is heated by the tank fluid and mixed with cold water to provide sanitary water.

### 2.9.3 Expansion Valve

In this case the expansion valve is not used to control the fluid flow to the evaporator, as in the usual heat pumps. It is, instead, necessary to keep gas cooler pressure at its optimum value. The following types of valves are used in trans-critical cycles.

**Back pressure valve.** The stem position is controlled by the upstream pressure, i.e., at gas cooler exit, in counter-action with a properly calibrated spring. If gas cooler pressure grows up, the stem travels to increase valve opening in order to increase flow rate; the opposite occurs if pressure decreases. In so doing, the valve is able to control the upstream pressure, but cannot control the flow rate and, consequently, the superheat at compressor's suction. Due to this a liquid separator is placed at the evaporator exit. Therefore we consider the evaporator as flooded.

**Back pressure valve coupled with a thermostatic valve.** This coupling allows for a "dry" evaporation: the back pressure valve is controlled by gas cooler pressure, while the thermostatic valve keeps vapor superheat constant at compressor's suction. Referring to Fig. 2.50 lines AB (or A'B' depending on the exit temperature



**Fig. 2.50** Back pressure valve coupled to a thermostatic valve

from gas cooler) and BC (or B'C') respectively represent the transformation occurring in the back pressure valve and in the thermostatic valve. In between the two valves a liquid receiver is located at the pressure of point B, on the saturation curve.

**Differential valve coupled with a thermostatic valve.** Back pressure valve can be replaced by a differential valve, where the stem position is controlled by the pressure difference,  $\Delta p$ , between the two sides of the valve. This difference is kept substantially constant and, as the thermostatic valve inlet is fixed at the point of saturated liquid, pressure is determined by the temperature at gas cooler exit. Still referring to Fig. 2.50, segment AB has a fixed length and point B moves on the saturation curve as a function of point A, that is the intersection of the isenthalpic passing through B with the constant pressure curve of the gas cooler.

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