

1

Space Heating

The primary function of a heating installation is to maintain ambient temperature in a building. Heating represents a significant portion of energy consumption: around 70% in the case of old residential buildings, and under 30% in modern buildings which comply with France's 2020 Building Regulation (known as RE 2020). The heating depends on the use of the building, the thermal characteristics, the local climate and the quality of the indoor temperature control systems.

1.1. Ambient temperature in buildings

In the wake of the first oil crisis in 1973, a number of legislative texts were introduced, setting thresholds to be respected. France's Construction and Housing Code (articles R131-20 to R131-22) stipulates the ambient temperature for the heating system based on building type and occupation schedule. For example, in residential, office and education buildings, the operative temperature in winter must not be below:

- 19°C when occupied;
- 16°C when unoccupied for 24–48 h;
- 8°C when unoccupied for longer than 48 h.

Note that indoor temperatures will only ever fall as low as these set points, when the building is unoccupied, if the outdoor temperature is low enough. To ensure the

For a color version of all figures in this chapter, see www.iste.co.uk/salagnac/energyefficiency2.zip.

temperature does not fall below these points, heating systems must include efficient control systems which, in all cases, require input from a professional heating engineer. In France, 77% of households say that they are most comfortable when the temperature is at least 20°C, with 63% specifying that they prefer it between 20 and 22°C (Association Qualitel 2015).

In buildings that are intermittently occupied, such as schools (which are occupied about 30% of the time), it is crucial to carefully manage periods when the ambient temperature is allowed to drop. This can save considerable amounts of energy on heating. Two strategies may be used to lower the temperature when the building is unoccupied (evenings, weekends and holidays) using a hot-water heating system:

- lowering the temperature of the water before it is sent to the emitters;
- shutting down the circulators so that no water is sent to the emitters.

The second solution is more effective. The temperature falls more quickly if no hot water reaches the radiators than if water at a lower temperature is still circulating through the radiators. In addition, shutting down the circulators saves electricity. It is essential to have a suitable control system for the circulators (see section 1.3.2.1).

1.2. Types of installations

Heating installations (but also domestic hot water (DHW) and air conditioning), whose performance is directly related to that of the building envelope, may be:

– Decentralized: local emitters are used, such as electric radiators, gas heaters and heat pumps (HPs), etc. (see Figure 1.1).

– Centralized with a dedicated heating network: an example of this would be a large heating system that supplies hot water to a district heating network serving multiple buildings, or even an entire neighborhood (see Figure 1.4). Paris' district heating company (CPCU, from the French name) uses eight heat generators and three recovery units, distributing steam through a 500-km network to supply over 6,000 delivery points. On a smaller scale, heating for priority urbanization areas (ZUP) on the city's outskirts distributes the hot water from a heater to a few dozen buildings. The same principle is sometimes applied for cold water distribution for cooling in urban zones with a high density of tertiary buildings (e.g. the network Climespace, for cold production in La Défense in Paris).

– Centralized and collective: a heating or cooling system (or thermodynamic system) serves an entire building (see Figure 1.3).

– Centralized and individual: a central heating (or cooling) system feeds the emitters by means of a distribution network, but that network serves only a single dwelling or workspace. An example is a private central heating system for an apartment which has its own boilers, radiators and distribution piping (see Figure 1.2).

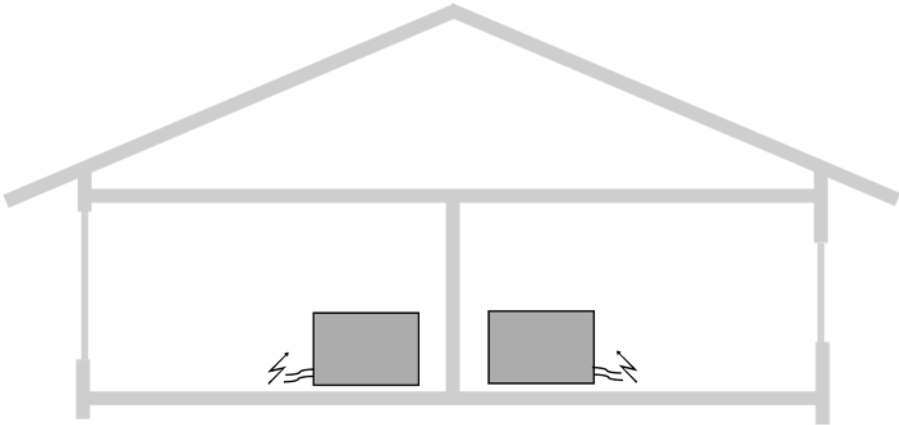


Figure 1.1. *Decentralized heating in a house (electric radiators)*

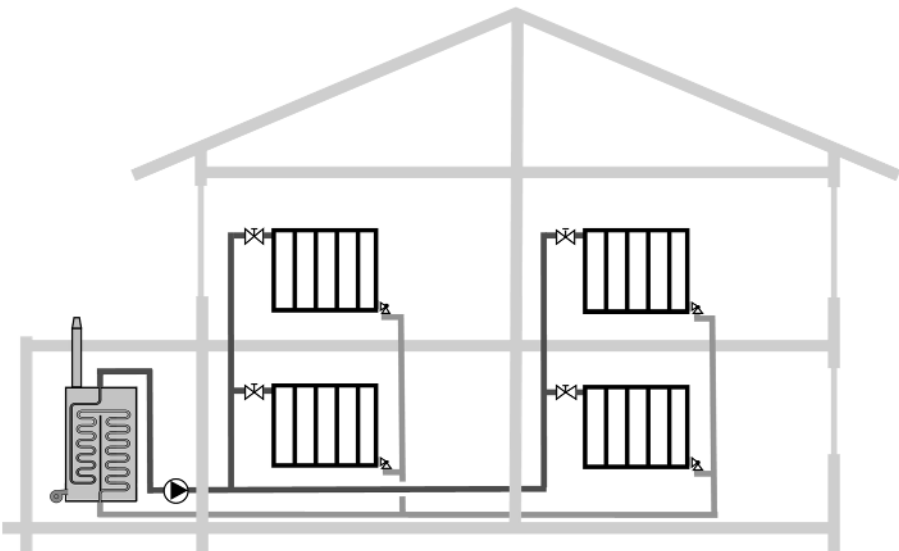


Figure 1.2. *Centralized and individual heating for a house (boiler + radiators)*

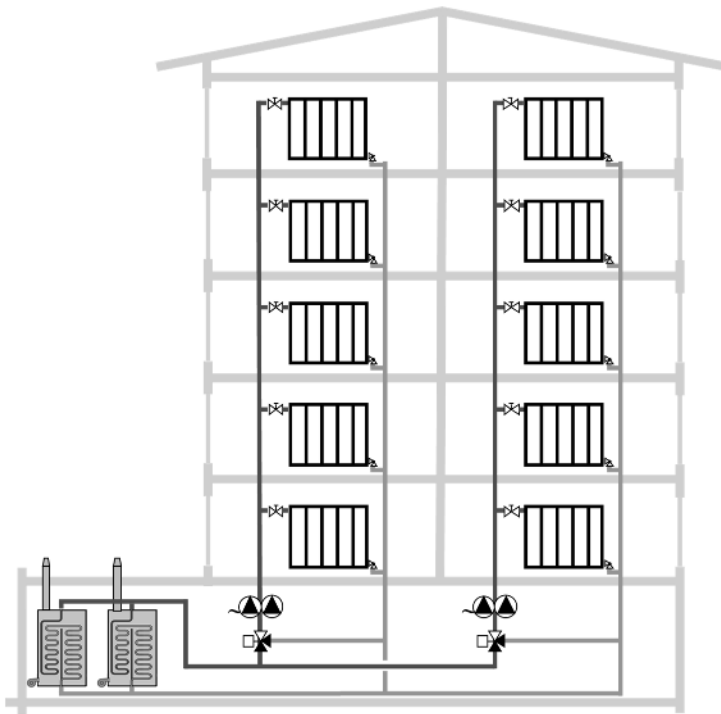


Figure 1.3. *Collective centralized heating system for a group of dwellings (boilers + radiators)*

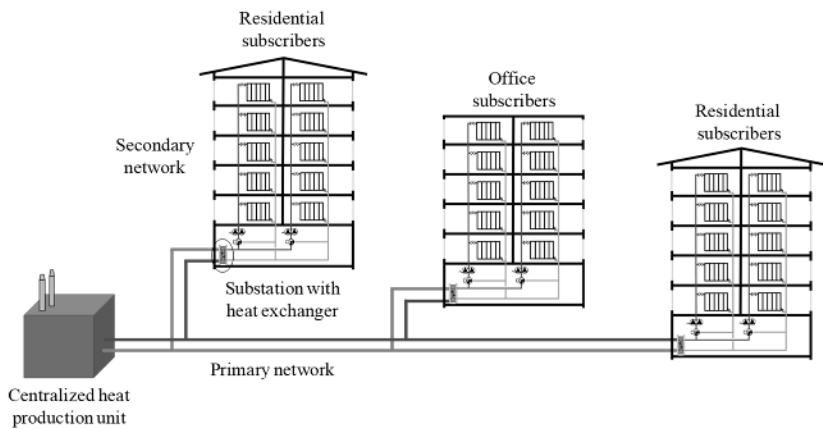


Figure 1.4. *Centralized heating system with a heating network serving multiple buildings*

1.3. Layout of an installation

In heating installations (and also DHW and air conditioning), we can always distinguish the following six functional subsets (see Figure 1.5):

- Emission: the terminal units (radiators, underfloor heating or cooling, air inlet vents) emit or absorb heat to or from the occupied spaces.
- Distribution: the heat-carrying fluid (water, air or steam) is carried to the terminal units by a network.
- Storage: the heat may be stored in insulated hot water tanks (this form of storage is necessary when using intermittent types of energy such as solar, or wood-fired boilers or HPs).
- Safety: safety equipment must be included to protect both the installation and its users – safety valve, expansion tank, disconnector, safety pump, low-water pressure switch, flow controller and safety thermostat.
- Energy production: to produce the required energy, it is possible to use boilers (fuel oil, gas, biomass: wood, waste materials, straw, etc.), combined heat and power (CHP), thermodynamic systems (HPs, refrigeration pumps, hot/cold pumps) or other types of energy (solar, geothermal, etc.) to heat or cool the fluid.
- Control system: all production, distribution and emission systems are governed by control protocols that enable the installation to operate automatically, while maintaining safety protections.

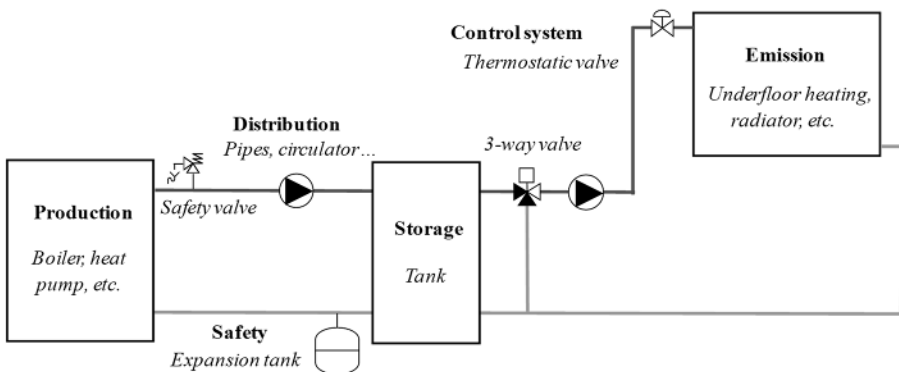


Figure 1.5. *Simplified schematic diagram of a hot-water heating system*

1.3.1. Emission

Heat emitters are designed to satisfy the heating demand of the spaces in which they are installed. These may be:

- Independent: plug-in electric heaters, gas radiators, liquid fuel heaters and wood-burning stoves. Some of these devices are stationary, while others are portable. Most combustion units have an exhaust duct to remove the combustion products.

- Gas-fired units connected to an internal fuel distribution system (gas radiators). They employ the same technologies as the most efficient standalone devices (both fixed and connected), although the presence of a gas supply network means they are classified as connected devices.

- Warm air insufflation devices, relying on mixing or displacement.

- Terminal units supplied by a network. These emitters are supplied by:

- a specific electrical network (convector, panel heater, electric underfloor system);

- a refrigerant-based system (multi-split, variable refrigerant volume (VRV) system);

- or a heat-carrying fluid distribution network using steam or hot water (smooth tube, winged tube, panel heater, radiator, underfloor heating, overceiling heating, radiant panels, heated wall, convector, fan-assisted convector, exhaust convector, chilled beam, air heater) (Figure 1.6).

An emitter is made of surfaces that are raised to a high temperature. It emits thermal power to compensate for the losses from the space:

- by convection;

- by longwave (infrared) radiation.

Depending on their design, some devices emit mostly by convection (natural or forced), while others emit mostly by radiation. A radiator normally emits more heat through radiation than convection, whereas a convector primarily relies on convective heat transfer. In practice, however, this is not always the case, as the heat distribution depends on the geometry and average temperature of the emitter (see Figure 1.7).

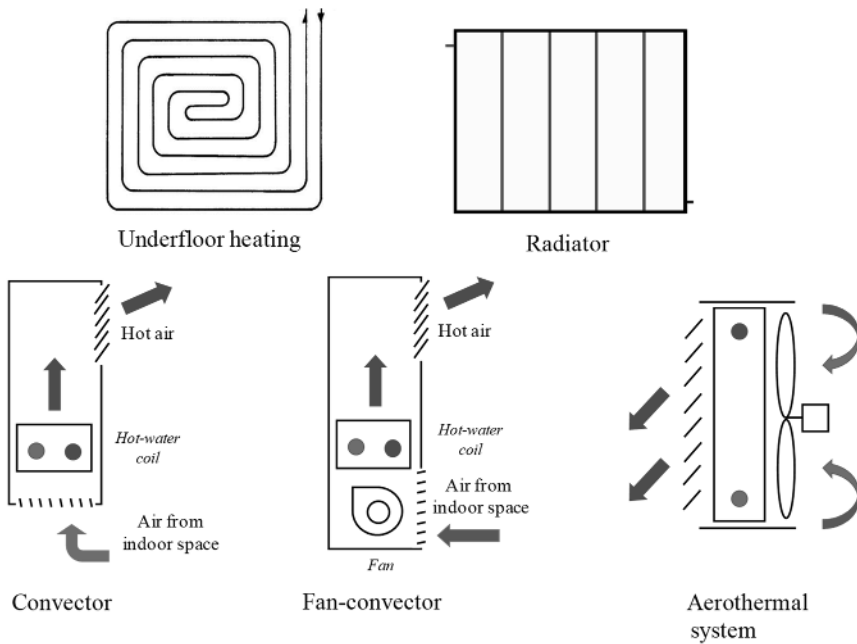


Figure 1.6. Some hot-water heat emission systems

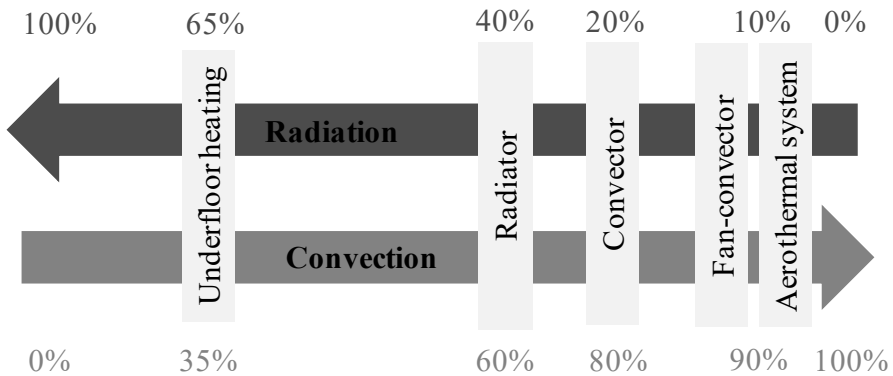


Figure 1.7. Convective and radiant share of some emitters (indicative values)

For heating large spaces such as atria or halls, it is advisable to opt for a radiant system (such as underfloor heating) to limit the need to heat the air mass, minimize

vertical temperature stratification and thus save on energy. With this method, optimal comfort is achieved.

When choosing the type of emitter, considerations should include the use of the building, the heating/ventilation/cooling demand, the volume and overhead height of the space, the installation constraints, the diffuser noise levels, intermittency and thermal inertia. The vertical temperature profile in a space will differ depending on the share of convective and radiant heat transfer, as shown in Figure 1.8.

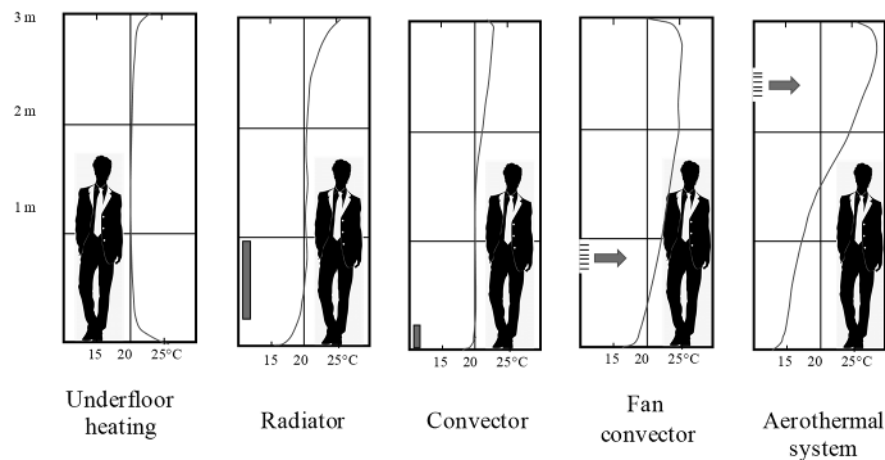


Figure 1.8. Vertical temperature gradient for different emitters

The energy efficiency of an emitter can be characterized by its spatial variation (SV) class (see Table 1.1). This parameter represents the difference between the temperature in the occupied zone and the average temperature of the surroundings. In essence, it is linked to floor-to-ceiling stratification and depends on the convective and radiant heat transfer from the emitter. Consider the example of a space with a ceiling height of 5 m. If underfloor heating is used (class A, and $SV = 0$), there is no vertical stratification of the ambient temperature. If, on the other hand, the space is heated by a radiator (class C, and $SV = 1.2$), the vertical temperature gradient is $5 \times 1.2 = 6$ K. In this case, with the same level of comfort for the occupants, the vertical stratification causes the average temperature in the room to rise. The air is warmer at the ceiling. Losses increase, as does energy consumption for space heating.

Emitter	Class	$H < 4 \text{ m}$	$4 \leq H < 6 \text{ m}$	$6 \leq H < 8 \text{ m}$	$H \geq 8 \text{ m}$
Underfloor heating	A	0	0	0	0
High-induction emitters: vertical jet diffusers, etc.	B1	0	0	0.5	1
Air diffusion, chilled beams, fan-convectors	B2	0	0.6	1.7	2.8
Radiative emitters (other than underfloor heating), ceiling- mounted heaters	B3	0.2	0.8	1.2	1.6
Other cases: radiators...	C	0.4	1.2	2	2.8

Table 1.1. *Spatial variation (SV) values in (K/m) for different emitters as a function of the ceiling height H in (m) (Th-BCE regulation 2012)*

1.3.1.1. Radiator technology

Many hot-water radiator models are available from manufacturers (see Figure 1.9):

- single or multi-panel radiators made of steel or aluminum. Some models have fins to increase their heat exchange surface area. They are available in a variety of geometric configurations;
- steel radiators with nerves, lamellae or folds;
- radiators with transverse sections or columns. These are composed of steel elements, or often cast iron, with high thermal inertia;
- tube radiators. These are made of steel or aluminum alloys and can therefore be made into elegant shapes.

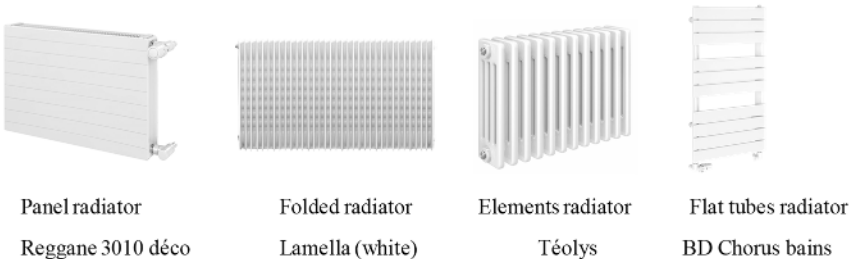


Figure 1.9. *Examples of radiators (credit: Finimetal)*

The position of heaters in a room is partially necessitated by the room's intended use (i.e. which areas must remain accessible) and by the pipe connection (piping layout, optimization of length). Typically, radiators are installed beneath windowsills. This helps to prevent the cold wall sensation, though this phenomenon is less pronounced in well-insulated buildings. To optimize convective and radiant heat transfer, a few rules should be observed: the emitter should be placed 5–6 cm from the wall and 8–10 cm above the floor, with no furniture or curtains in front of it. The fittings (horizontal and vertical panels, or vertical screens) limit the emissive power.

A radiator can be connected in a number of ways. For optimal heat distribution and to prevent temperature stratification within the radiator sections, it is recommended to supply hot water at the top and return it at the bottom on the opposite side (see Figure 1.10).

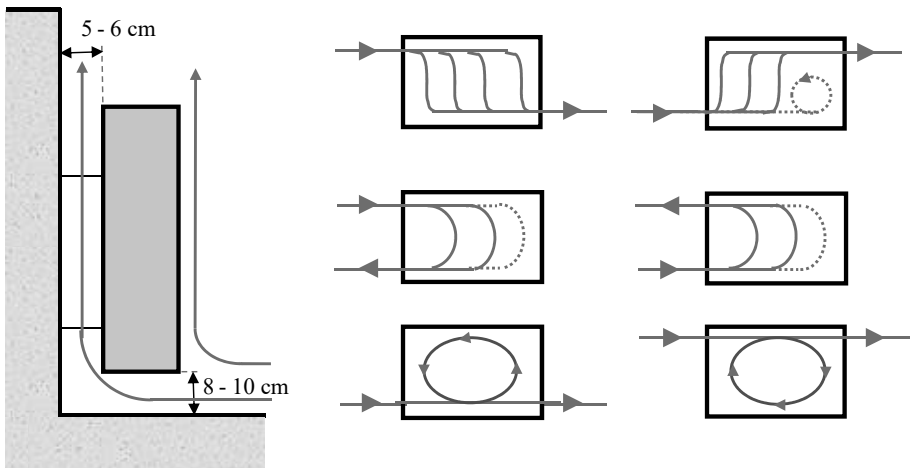


Figure 1.10. *Installation and connections of a radiator*

1.3.1.2. Heat emission from radiators

The heat output of an emitter can be expressed in two different ways, depending on whether it is calculated on the basis of the energy supplied by the heat-carrying fluid or the energy emitted into the surrounding space (see Figure 1.11). Under steady-state operating conditions, the power supplied by the fluid is equal to the power emitted into the surrounding space.

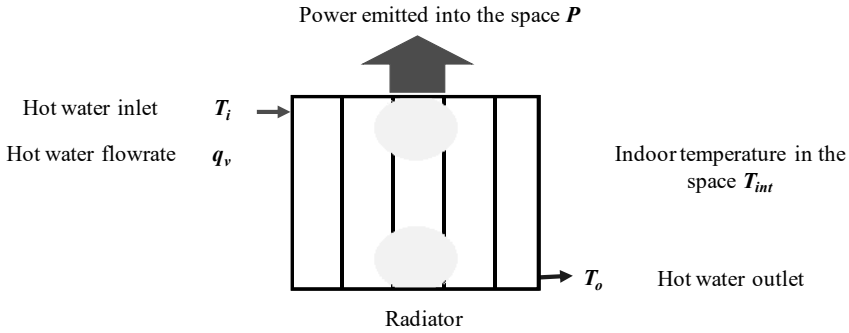


Figure 1.11. Heat balance of a radiator

The power P in (kW) delivered by the heating fluid is expressed by the following formula:

$$P = 1.16 q_v (T_i - T_o) \quad [1.1]$$

where 1.16 represents the specific heat capacity of water ($\text{kWh}/(\text{m}^3 \cdot ^\circ\text{C})$), q_v is the volumetric flow rate (m^3/h), and T_i and T_o are the water inlet and outlet temperatures ($^\circ\text{C}$), respectively.

The power emitted into the space is set out in EN 442-2, and is expressed by:

$$P = B \Delta T^n \quad [1.2]$$

If $\frac{T_o - T_{int}}{T_i - T_{int}} < 0.7$, we use the logarithmic mean temperature difference:

$$\Delta T = \Delta T_{lm} = \frac{(T_i - T_{int}) - (T_o - T_{int})}{\ln \frac{T_i - T_{int}}{T_o - T_{int}}}$$

If $\frac{T_o - T_{int}}{T_i - T_{int}} \geq 0.7$, we use the arithmetic mean temperature difference:

$$\Delta T = \Delta T_{arit} = \frac{T_i + T_o}{2} - T_{int}$$

where B is a characteristic of the radiator determined experimentally, and T_{int} is the interior temperature of the space ($^\circ\text{C}$). n is known as the “radiator exponent” or the “temperature coefficient” and is determined by means of standardized tests. For a

radiator, n lies between 1.1 and 1.4. The power of an emitter varies as a function of the water inlet temperature. Its emission law is not a linear function, except in the case of underfloor heating (see Figure 1.12).

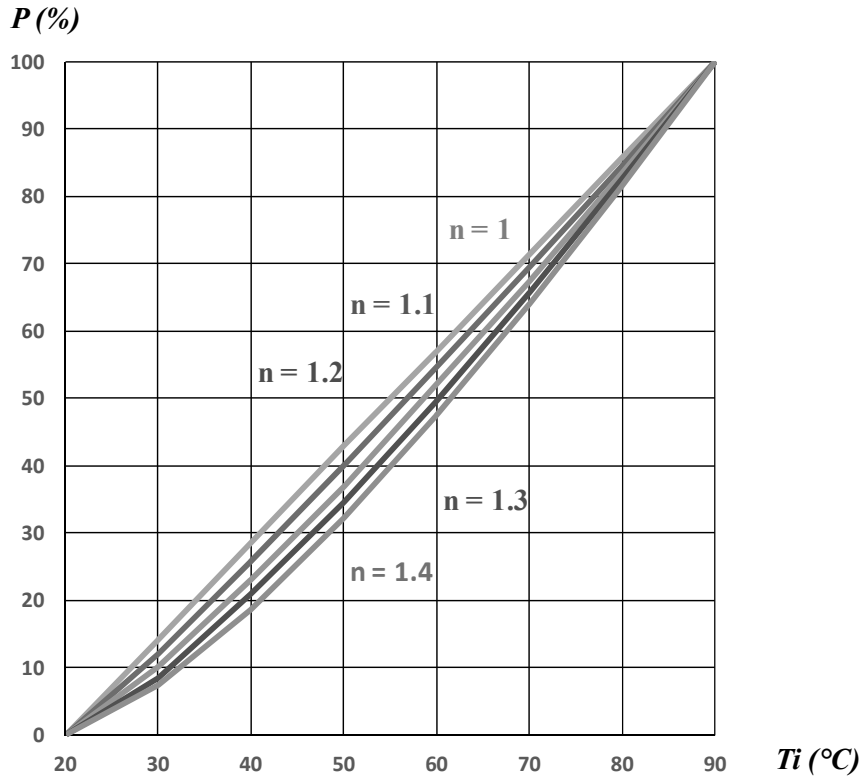


Figure 1.12. Power/temperature characteristics of emitters ($n = 1$ for underfloor heating; $1.1 \leq n \leq 1.4$ for a convector; $n = 1.3$ for a radiator)

Thermal power is determined by standardized tests carried out in an environmental chamber. Manufacturer catalogs quote reference powers of P_{50} for a ΔT of 50°C ($T_i = 75^\circ\text{C}$, $T_o = 65^\circ\text{C}$ and $T_{int} = 20^\circ\text{C}$).

Radiators are available in a wide range of geometries to suit diverse installation layouts (see Table 1.2). For example, a type-21 panel heater means that it is composed of two panels (the first digit, 2) and one row of fins (the second digit, 1).

Type 10	Type 11	Type 20	Type 21	Type 22	Type 32	Type 33
						

Table 1.2. *Characteristics of panel heaters*

A radiator is sized to compensate for the heat losses of a space, calculated for standardized outdoor conditions. When selecting a radiator from the manufacturer's catalog, we choose the nominal power closest to but *higher than* the required power. Some oversizing is therefore inevitable. However, this surplus must not be excessive, as it will needlessly increase the cost of the installation. Also, if the surplus power is too high, the control will become more complex. In a detailed calculation, the heat output from the piping carrying the heating fluid must be taken into account and subtracted from the total power required from the emitters. In the case of intermittently occupied buildings (which might have a temperature of 15°C while unoccupied, for example), the emitter has a “natural” overpower of around 15% upon restart. This overpower can be further increased by raising the temperature of the water supplied to the emitter.

To select the appropriate radiator, it is necessary to set a suitable operating condition, defined by the water inlet and outlet temperature. The operating condition influences the radiator sizing, water flow rate, hydraulic balance and the overall control system.

In old and poorly insulated buildings, the heating demand is high and the radiators are supplied with “high-temperature” water at around 80–90°C (with $T_i - T_o \approx 0^\circ\text{C}$). For insulated buildings, “medium-temperature” water at around 65–80°C is suitable (with $T_i - T_o \approx 15^\circ\text{C}$). New buildings (compliant with RE 2020) or buildings renovated for low energy consumption have “low-temperature” water at around 50–65°C, or “very low-temperature” water at 30–50°C (with $T_i - T_o \approx 10^\circ\text{C}$). Low and very low temperatures favor low-potential energy sources (heat pumps, solar power, etc.), ensure high efficiency in condensing boilers, reduce distribution and storage losses, and improve thermal comfort by preventing the formation of significant “hot spots” within buildings.

The flow rate of water through the emitter is directly linked to the temperature difference between the inlet and outlet ($T_i - T_o$) in nominal conditions. To ensure satisfactory thermal and hydraulic operating conditions, a temperature difference must be selected to suit the type of emitter (with minimum flow rate). The

temperature difference also needs to be optimized for both technical and economic reasons. If the temperature difference increases:

- the flow rate decreases, which reduces the cost of the piping network, the pump and the associated electrical consumption;
- the radiator's average temperature decreases, requiring a larger surface to achieve the same heat output, and the cost goes up.

The flow rate can immediately be deduced:

$$q_v = \frac{P}{1.16 (T_i - T_o)} \quad [1.3]$$

1.3.1.2.1. Example of a renovation including a radiator

The design heat losses of an existing space are 1,500 W, assuming an indoor temperature of 20°C and an outdoor temperature of −5°C. The radiator is from the Finimetal brand (model Reggane 3010 Déco type 21) and is 600 mm tall. The operating condition is 80/60°C.

Under these conditions, as $\frac{T_o - T_{int}}{T_i - T_{int}} = \frac{60 - 20}{80 - 20} = 0.66 < 0.7$, then:

$$\Delta T = \Delta T_{lm} = \frac{(T_i - T_{int}) - (T_o - T_{int})}{\ln \frac{T_i - T_{int}}{T_o - T_{int}}} = \frac{(80 - 20) - (60 - 20)}{\ln \frac{80 - 20}{60 - 20}} = 49.3^\circ\text{C}$$

$$\text{Taking } \Delta T = \Delta T_{arit} = \frac{T_i + T_o}{2} - T_{int} = \frac{80 + 60}{2} - 20 = 50^\circ\text{C}$$

The difference between ΔT_{lm} and ΔT_{arit} is 1.4%.

Based on the manufacturer's data, the technical characteristics can be determined, as shown in Figure 1.13 (see Finimetal's power calculator: <https://www.finimetal.fr/fr-fr/outils-et-services/calculateur-de-puissance>, but it is only available in French).

The required water flow rate can be calculated as follows:

$$q_v = \frac{P}{1.16 (T_i - T_o)} = \frac{1.534}{1.16 (80 - 60)} = 0.066 \text{ m}^3 \cdot \text{h}^{-1} = 66 \text{ L} \cdot \text{h}^{-1}$$



	50
Power (W)	1,534
Height (mm)	600
Width (mm)	1,200
Thickness (mm)	105
Mass (kg)	38.6

Figure 1.13. Specifications of the Finimetal Reggane 3010 Déco type 21 radiator

After renovation, the design heat losses become 750 W. Thus, the radiator is too powerful. There are then two options: remove elements from the radiator, or adapt the operating condition.

Under the same operating condition, the radiator only needs to be half as long. Considering a new operating condition, for example, 60/50°C, let us determine the new specifications for the radiator:

$$\text{Under these conditions: } \Delta T_{arit} = \frac{T_i + T_o}{2} - T_{int} = \frac{60 + 50}{2} - 20 = 35^\circ\text{C}$$

The radiator's power is then $P_{(35^\circ\text{C})} = 752 \text{ W}$.

The radiator dimensions are as follows: height 600 mm, width 920 mm and depth 105 mm.

The water circulation rate must be:

$$q_v = \frac{P}{1.16 (T_i - T_o)} = \frac{0.752}{1.16 (60 - 50)} = 0.065 \text{ m}^3 \cdot \text{h}^{-1} = 65 \text{ L} \cdot \text{h}^{-1}$$

It is possible to more accurately determine the new operating condition without altering the existing radiator.

Based on the emission law:

– For the operating condition 80/60°C:

$$P = B \Delta T^n$$

– For the new operating condition at T_i'/T_o' :

$$P' = B \Delta T'^n$$

B and n are constants, which are characteristic of the radiator. The slope coefficient n is given by the manufacturer as $n = 1.3022$, and B can be determined, but as we will see, it is not necessary to determine it.

Based on the emission laws, we can deduce:

$$\frac{P'}{P} = \frac{B \Delta T'^n}{B \Delta T^n} = \left(\frac{\Delta T'}{\Delta T} \right)^n$$

$$\text{Hence, } \Delta T' = \Delta T \left(\frac{P'}{P} \right)^{1/n}, \text{ where } \Delta T' = \frac{T_i' + T_o'}{2} - T_{int}$$

$$T_i' + T_o' = 2 (\Delta T' + T_{int})$$

On the basis of the heat balance on the water side:

– For the operating condition 80/60°C:

$$P = 1.16 q_v (T_i - T_o)$$

– For the new operating condition at T_i'/T_o' :

$$P' = 1.16 q_v (T_i' - T_o')$$

The flow rate is not altered, so:

$$\frac{P'}{P} = \frac{1.16 q_v (T'_i - T'_o)}{1.16 q_v (T_i - T_o)} = \frac{T'_i - T'_o}{T_i - T_o}$$

$$T'_i - T'_o = \frac{P'}{P} (T_i - T_o)$$

NUMERICAL EXAMPLE OF APPLICATION.

$$\Delta T' = \Delta T \left(\frac{P'}{P} \right)^{1/n} = 50 \left(\frac{750}{1,500} \right)^{1/1.3022} = 29.36^\circ\text{C}$$

$$T'_i + T'_o = 2 (\Delta T' + T_{int}) = 98.7^\circ\text{C}$$

$$T'_i - T'_o = \frac{P'}{P} (T_i - T_o) = \frac{750}{1,500} (80 - 60) = 10^\circ\text{C}$$

Thus: $T'_i = 54.4^\circ\text{C}$ and $T'_o = 44.4^\circ\text{C}$

Hence, the new operating condition after renovation is $55/45^\circ\text{C}$. The radiator and the hydraulic network, if they are in good condition, are not modified. Only the heating curve is adapted. This low-temperature solution is favorable for energy efficiency in production systems using a condensing boiler or an HP.

1.3.1.2.2. Control of radiator power

To maintain the comfort temperature and make energy savings, the heat emission of the radiators must adapt to free heat contributions in the spaces.

The emission of a radiator is controlled by adjusting the inflow rate using manual, thermostatic or motorized valves. However, the variation in power is not proportional to the flow rate (Figure 1.14).

It is a function of the efficiency of the heat emitter, defined by:

$$\varepsilon = \frac{T_i - T_o}{T_i - T_{int}} \quad [1.4]$$

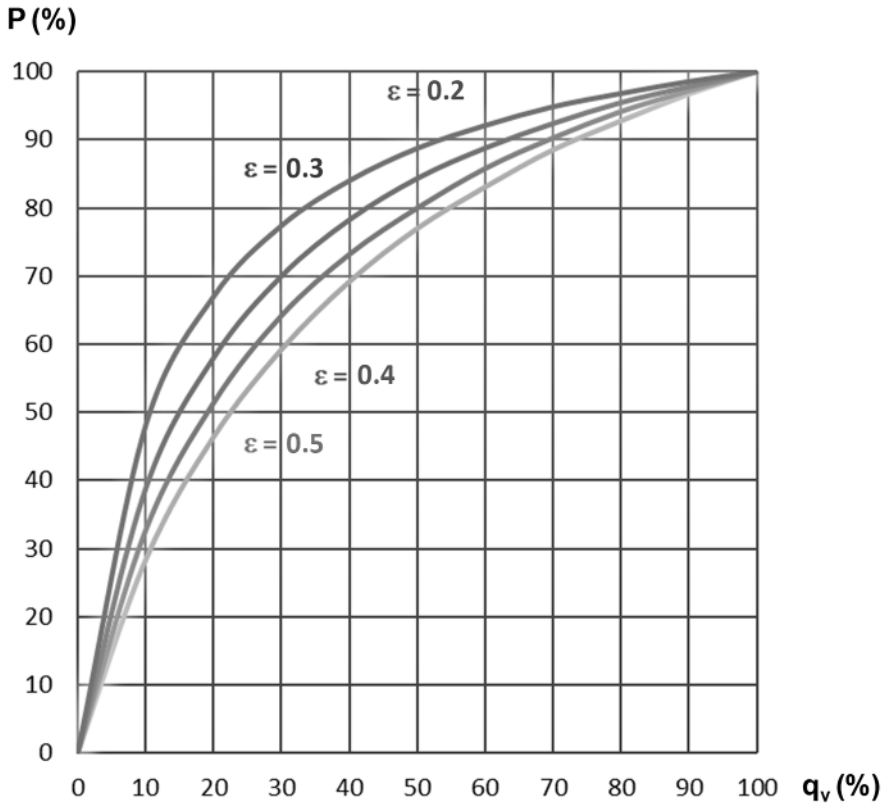


Figure 1.14. Variation of thermal power of a radiator depending on the flow rate and the nominal efficiency

To obtain good control at low and high flow rates, the nominal efficiency must be greater than 0.2. The efficiency of an emitter is better in a low-temperature regime than a high-temperature one:

- low-temperature operating condition at 45/35°C and $T_{int} = 20^\circ\text{C}$, so $\Sigma = 0.40$;
- high-temperature operating condition at 90/70°C and $T_{int} = 20^\circ\text{C}$, so $\Sigma = 0.29$.

The choice of two-way valve is crucial for the quality of the control system. We select a two-way valve whose emission, as a function of flow rate, varies in the opposite direction to the emission law.

The valve must have sufficient pressure drop in relation to the radiator to ensure the flow rate can be controlled with sufficient accuracy. The recommended hydraulic authority must be at least 0.5 (meaning that the pressure drop of the valve when fully open is equal to the pressure drop of the radiator).

Thermostatic valves automatically regulate heat emission with a head that is sensitive to the temperature in the space. This head contains a capsule filled with a liquid, gel or gas, which expands or contracts as the ambient temperature changes. Users can control the set ambient temperature, with graded settings from 1 to 5. Note that the settings are not graded in relation to the ambient temperature ($^{\circ}\text{C}$), because the thermostatic head is inevitably affected by the heat released by the radiator, and by the water temperature, which varies as a function of the outdoor temperature and the cold radiation from the wall.

A thermostatic valve has specific geometric characteristics (diameter, shape, pipe fitting, etc.), hydraulic characteristics (K_v value) and thermal characteristics (hysteresis, thermal conductivity, influence of pressure, response time, variation over time).

In the legislation governing the performance of thermostatic valves, in France, AFNOR certifies the value of the temporal variation (TV) of thermostatic valves, meaning the degree of precision with which the temperature can be controlled. (Figure 1.15). Thus, TV is a crucial parameter relating to a building's energy consumption. The TV of certified thermostatic valves is generally between 0.1 and 1 K. A TV reduced by 0.1 K is equivalent to a 2% decrease in consumption for space heating.

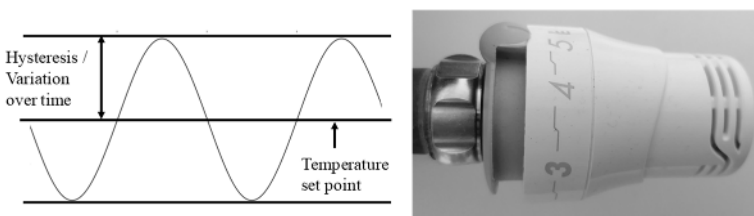


Figure 1.15. *Thermostatic valve and temporal variation (TV)*

The temperature measured by the thermostatic head must be as representative as possible of the ambient temperature. The head needs to be far from parasitic influences: the corner of a room, hot air rising from the pipes or the radiator, a thick radiator, decorative covers, curtains, etc. Manufacturers of thermostatic valves offer

numerous functions: a separate thermostatic probe, a connected thermostatic head, presetting of flow rate, flow rate limiters, limited or prohibited settings and self-balancing thermostatic valves. A thermostatic valve alone cannot control the thermal power emitted by the radiator. In order to operate optimally, the following conditions must be met:

- The first level of control adjusts the inflow temperature as a function of the outdoor temperature. This is the role of the three-way valve (3WV) in the heating system.

- This control system must take account of the effect of solar radiation received (zoning) so that the thermostatic valve only compensates for heat gains within the space itself.

- In private residences, one or two radiators may have no thermostatic valves – in particular, in the room containing the thermostat or the centralized ambience sensor, so as not to influence the central control system.

1.3.2. Distribution

Generally, radiators are connected by a two-pipe system: each radiator has an inlet and an outlet pipe. This is the most commonly used setup (see Figure 1.16). It allows all the radiators on a network to be fed at the same temperature and facilitates hydraulic balancing and control. Today, the use of manifold layouts is becoming increasingly common.

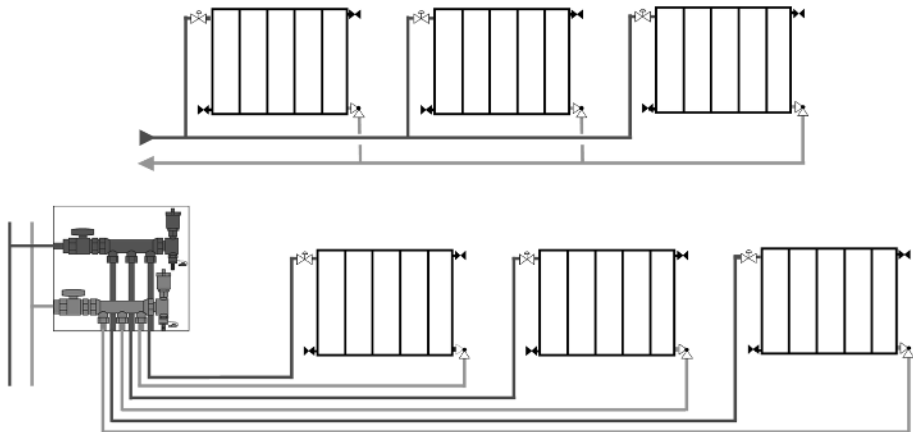


Figure 1.16. *Two-pipe setup (top) and manifold layout (bottom)*

A one-pipe serial system is used to connect two, or a maximum of three, radiators in a single room (Figure 1.17). In this way, we can save on piping and installation. If a valve on the circuit closes, all the radiators are isolated.

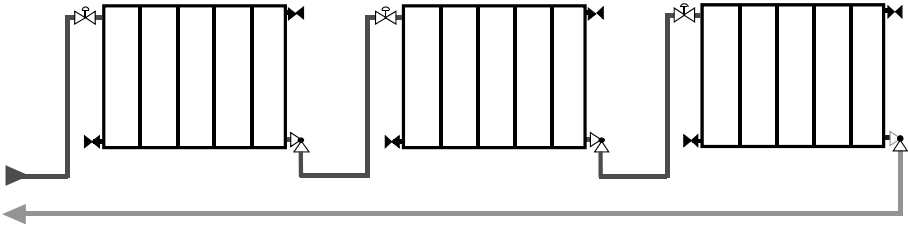


Figure 1.17. *One-pipe serial system*

The one-pipe bypass system is a simple setup (see Figure 1.18) using fewer pipes than a two-pipe system. The temperature of the water supplied to the last radiator in the system is lower than that supplied to the first. Therefore, the later radiators must be larger to have the same heating power. The design and hydraulic balancing are more difficult. Therefore, this system is limited to a small number of radiators.

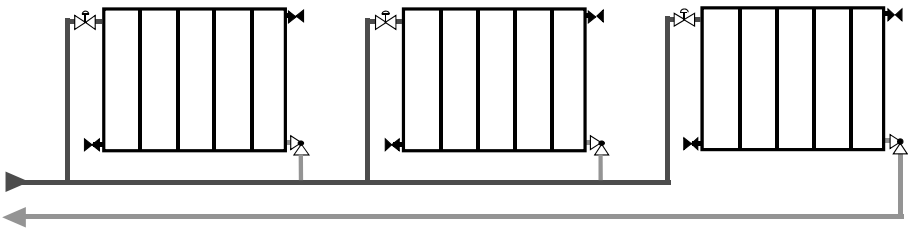


Figure 1.18. *One-pipe bypass system*

Distribution differs depending on the type of building. To control the distribution by thermal zoning of the installation, we create a sub-network with emitters in rooms where the heating demand tends to follow the same pattern. Often, the sub-network may include devices placed in rooms that face the same direction, and therefore have the same exposure to solar heat gains (the southern elevation in one sub-network, and the northern elevation in another). Each network has its own distribution system (circulator) and a centralized control system (3WV), which adjusts the water supply temperature, based on the outdoor and ambient

temperatures. The radiators are fitted with thermostatic valves to control the temperature in each room (see Figure 1.19).

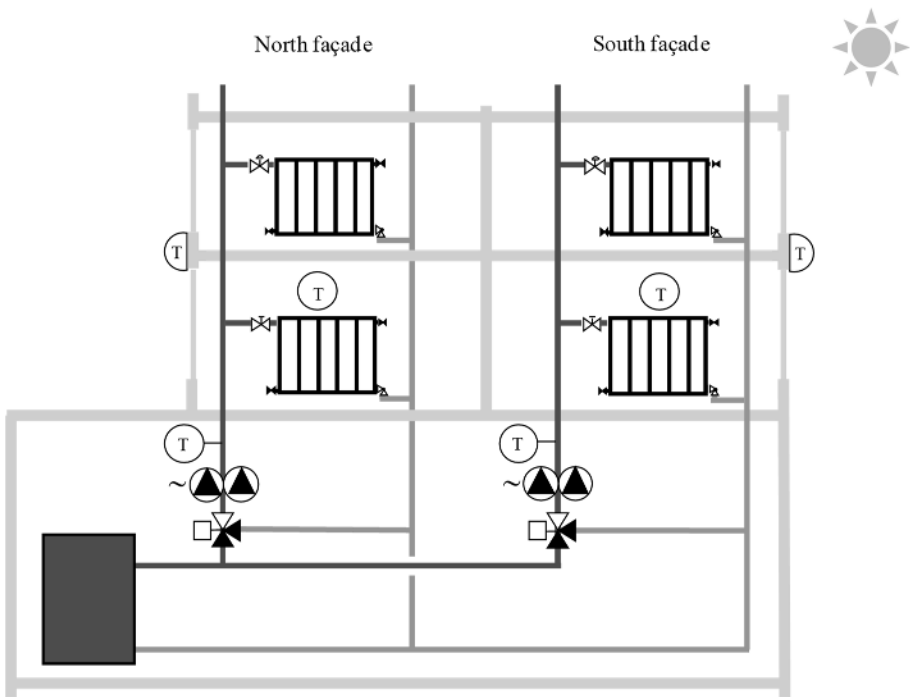


Figure 1.19. *Distribution with northerly and southerly supply columns*

For the conventional distribution with upward columns (see Figure 1.20), the radiator connections branch off from those main columns, serving spaces with similar heat losses, in candelabra or umbrella configurations. This setup is often found in existing buildings, but is not advisable in new buildings. A single control system is used for all the networks and thermal zones, rather than an individual system for each.

Centralized heating systems with individual distribution networks emerged in residential buildings in the early 1990s. One or more vertical risers in a vertical shaft, in the communal areas, carry heat around the building from the boiler room (see Figure 1.21). Each residence has an independent horizontal heating distribution

network, connected to the vertical riser by a thermal module, comprising a hydraulic separator (HS) and a circulator. Each dwelling has its own heating control system with a thermostat to control the ambient temperature, which turns the circulator on or off. A thermal energy meter records the consumption of the individual dwelling, and thus ensures that the building's heating costs are fairly distributed among the residents, in accordance with the Construction and Housing Code (article R131-2). Using a tank avoids any hydraulic impact on the distribution network, and therefore on the other apartments in the building. It requires a single circulator per apartment and reduces the power of the circulator for the distribution network because pressure drop at the apartment level is not accounted for. However, the presence of a tank is not conducive to condensation in the boilers because it reheats the water from apartments which do not require hot water at the time. The flow rate of the main pump does not change.

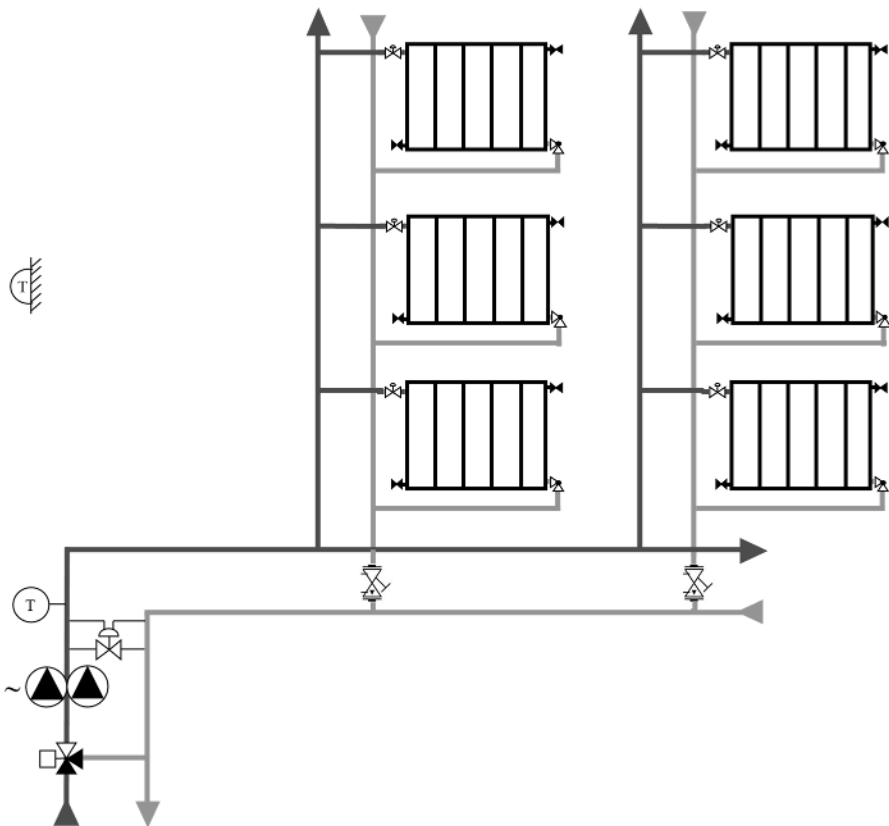


Figure 1.20. Regular distribution with vertical risers

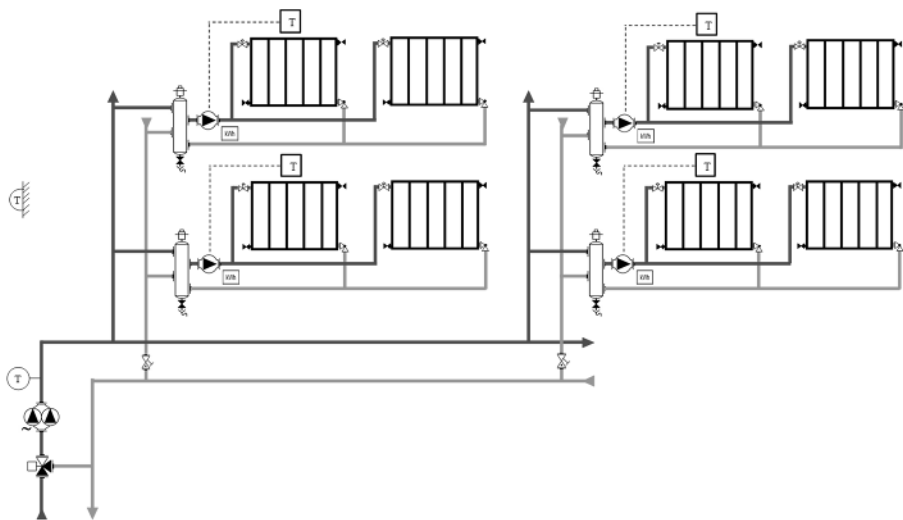


Figure 1.21. *Fixed flow rate, collective heating system with individual connections*

Using a two-way valve, we can introduce a variable-speed circulator, which consumes less energy, into the collective distribution network. The heating in the apartments is regulated by an ambient temperature thermostat, which controls the motorized two-way valve for the heating system (Figure 1.22).

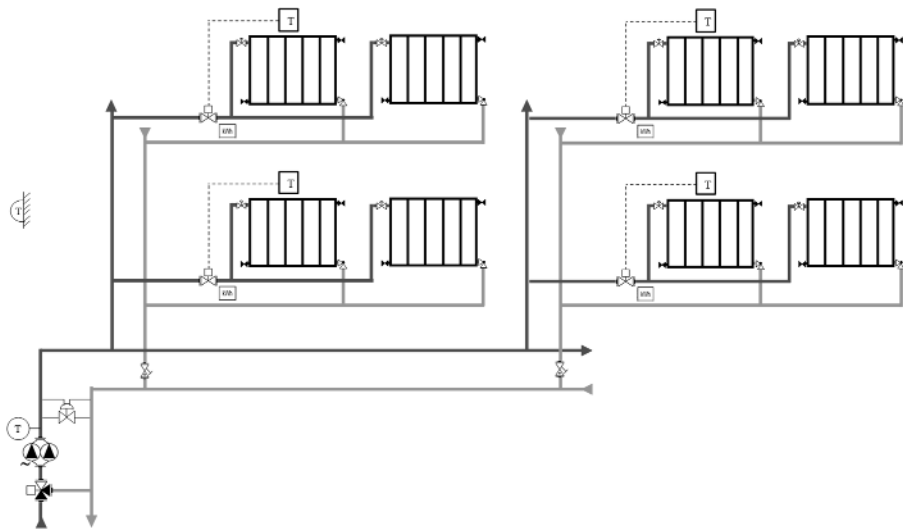


Figure 1.22. *Variable flow rate, collective heating system with individual connections*

For distribution by an Apartment Heat Interface Unit (HIU), an HIU placed in each apartment or technical shaft supplies the various apartments with heating and DHW from a central boiler room.

The arrangement shown in Figure 1.23 includes a flat-plate heat exchanger with a two-way valve, controlled by a thermostat, for instantaneous DHW production. For space heating, radiators are supplied with hot water by a circulator. The independent control system for each dwelling is controlled by a three-way mixing valve. The operating conditions are limited by a bypass. Each HIU has a thermal energy meter, which ensures that heating and DHW costs are accurately distributed among the various apartments. Manufacturers offer numerous types of HIUs with different hydraulic systems and control systems.

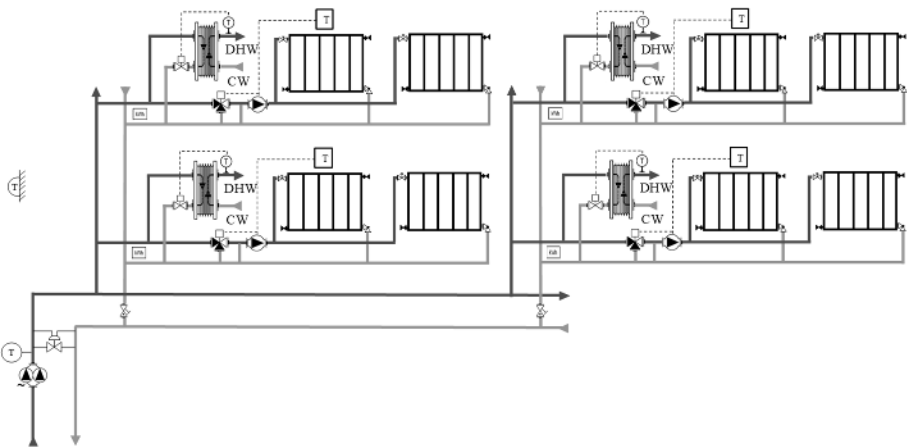


Figure 1.23. *Distribution by a Heat Interface Unit*

1.3.2.1. Heating distribution equipment in the boiler room

The distribution equipment in the boiler room supplies a network of emitters, with the correct flow rate and temperature of water to ensure the occupants' thermal comfort (see Figure 1.24).

Three-way sector mixing valves and three-way plug mixing valves are available from manufacturers. For heating networks, it is advisable to use plug valves. When installing the valve, it is important to follow the setup directions specified by the manufacturer, which are marked on the valve itself by the letters A (direct path), B (bypass path) and AB (mixed path).

For a radiator network, it comprises:

- a 3-way valve;
- a twin pump;
- a differential pressure valve;
- numerous components that are indispensable for the network to operate correctly:
 - outdoor temperature probe;
 - water outlet temperature probe;
 - water return temperature probe;
 - ambient temperature probes;
 - thermometers;
 - isolation valves;
 - drainage valves;
 - check valve;
 - automatic purgers.

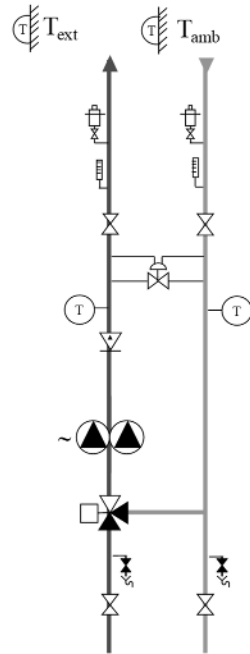


Figure 1.24. *Distribution for a radiator network*

The three-way valve controls the power of a network of emitters by adjusting the water outlet temperature or flow rate (see Figure 1.25).

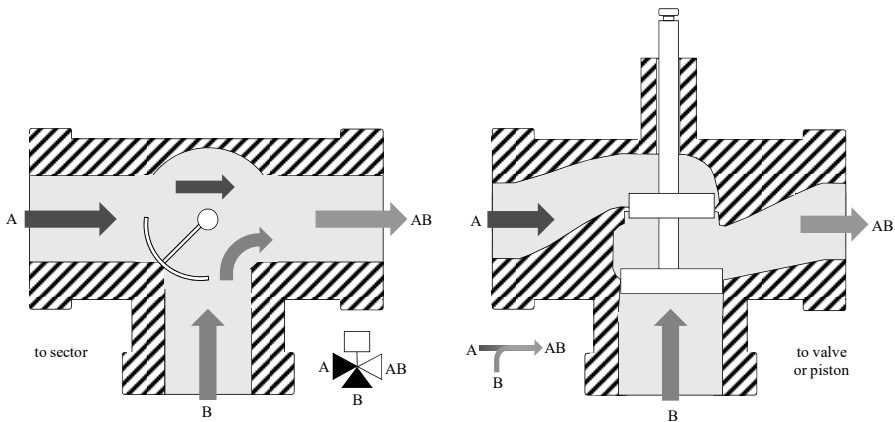


Figure 1.25. *Three-way valve*

There are various ways to set up the three-way valve. In practice, two tend to be implemented: mixing and inverse proportional pressure relief (see Figure 1.26). Indeed, the majority of three-way valves sold are “convergent”, meaning they have two inlets and one outlet.

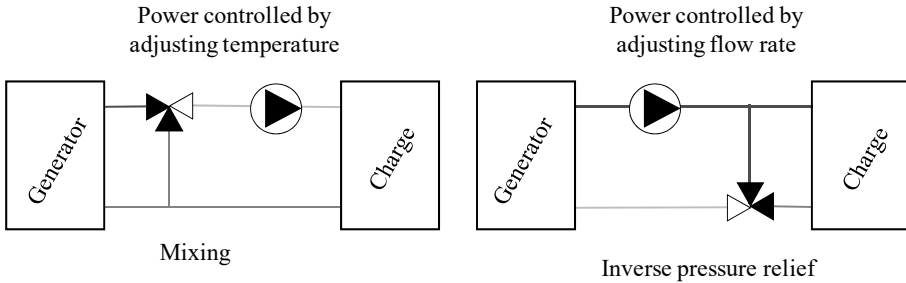


Figure 1.26. *Hydraulic setup of three-way valves*

In a radiator network, the water temperature can be varied by adjusting the three-way mixing valve. The water emanating from the hot water production system is mixed with the flow of water returning from the emitters.

In winter, heating demand is high. The water must be sent to the emitters at a high temperature. When it is less cold, the temperature of the outgoing water can be lowered. This will reduce the efficiency of condensing boilers and also reduce distribution and storage losses.

The set water outlet temperature is calculated in accordance with a heating curve, programmed into the system. Using the heating curve, we can calculate the set water outlet temperature T_{set} in relation to the outdoor temperature T_{ext} . That heating curve is linear for an emitter with a linear power/temperature emission function, such as an underfloor heating system (Figure 1.27). It is defined by two points. For example:

- point A: for an outdoor temperature of -5°C , the water outlet temperature must be 65°C ;
- point B: for an outdoor temperature of 20°C , the heating is no longer needed, and the water outlet temperature must be 20°C .

For convectors and radiators, the power law is slightly curved to compensate for their nonlinear power/temperature emission law.

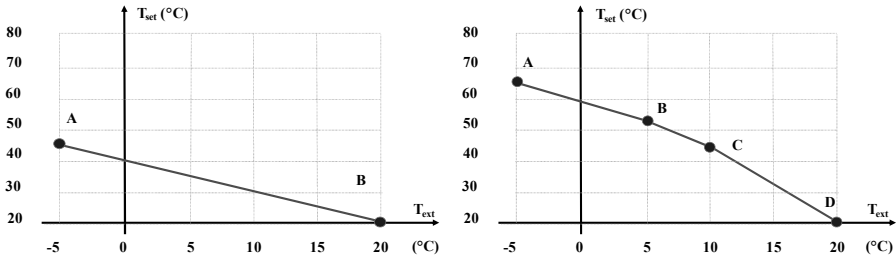


Figure 1.27. Examples of heating curves

The system continuously calculates the difference between the set water outlet temperature T_{set} (outdoor temperature probe + heating curve) and the true value T_d (measured by a water outlet probe). A control signal between 0 and 100%, proportional to that difference, is sent to the motor of the three-way valve.

To optimize the control of the ambient temperature, multiple ambient probes are needed. The average of the readings from the ambient probes will tend to increase the heating curve by 3°C , or reduce it by 5°C (adjustable), per degree Celsius of difference between the set temperature and the average temperature read by the ambient probes.

In order to function properly, the three-way valve must have an equal-percentage characteristic curve (bypass plunger, skirt-guided or cylindrical pierced) to compensate for the incurvation of the emitters' power/flow rate curve (Figure 1.28).

The three-way valve also needs to have sufficient authority a – as close as possible to 0.5. The power/aperture law then exhibits behavior very close to linear, and the power control system shows good progressivity across the whole of its range of variation. An authority of 0.5 is obtained by selecting a three-way valve with a pressure drop equal to that of the variable-flow circuit.

The authority of a control valve, at maximum aperture, is defined as follows:

$$a = \frac{\Delta H_{v100\%}}{\Delta H_{v100\%} + \Delta H_{r100\%}} \quad [1.5]$$

– $\Delta H_{v100\%}$: pressure drop of the three-way valve when 100% open;

– $\Delta H_{r100\%}$: pressure drop of the variable-flow loop when the three-way valve is 100% open.

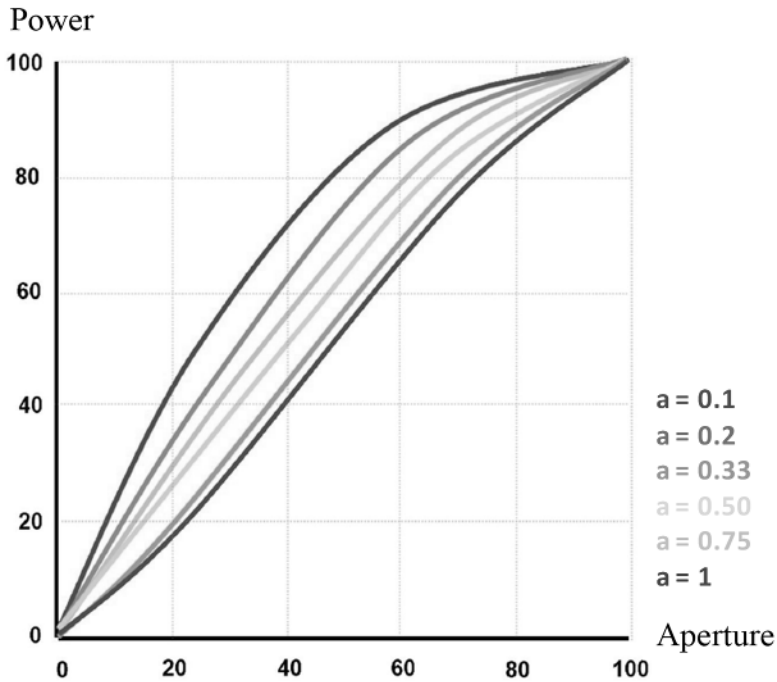


Figure 1.28. *Power/aperture law for an equal-percentage three-way valve*

The circulator of a heating loop moves the fluid (water, for example) from the generator to the emitters. It consumes electricity. Its purpose is to counteract the pressure drops ΔH (friction generated by the water in the closed heating loop) by maintaining a pressure difference between the pump inlet and outlet.

The pressure drop in mCE changes with the square of the flow rate q_v in (m^3/h): $\Delta H = k q_v^2$, where k is a constant characteristic of the network in ($\text{mCE} \cdot \text{m}^{-6} \cdot \text{h}^2$).

By a detailed calculation of the pressure drop at nominal flow rate, we can determine the characteristic constant of the network, k , and select the circulator on the basis of the characteristic curve given by the manufacturer (Figure 1.29).

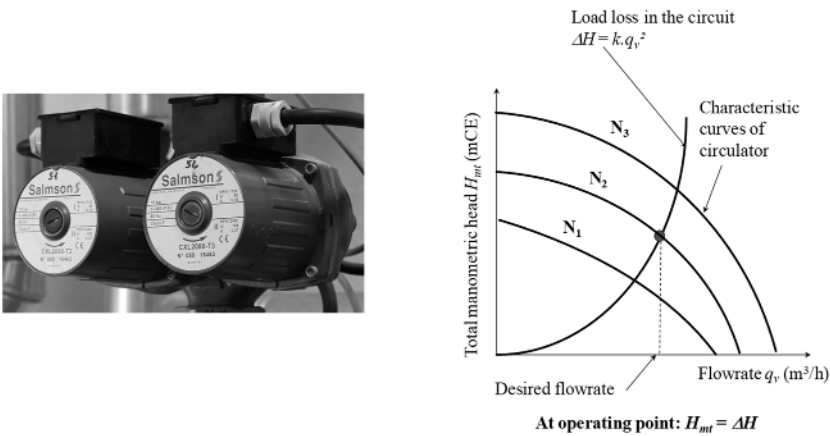


Figure 1.29. Characteristic curves of a circulator with its operating point

The characteristic curve represents the manometric head (H_{mt}) as a function of flow rate for different rotation speeds (N_1 , N_2 , N_3). The intersection between the curve of the network's pressure drop and the circulator's characteristic curve gives us the operating point and the desired flow rate.

To ensure continuity of service, it is advisable to use a twin pump, with the two systems in a parallel arrangement. Generally, only one pump is operational at a time. The second is an emergency backup in case the first fails. A cyclical switchover is scheduled every 168 h to ensure both pumps experience the same degree of wear and tear, and keep them in good working condition. After a prolonged stoppage, the axle of the pump may seize up, and then need to be unstuck by forcing the axis of the rotor with a screwdriver to start it rotating again.

Modern circulators operate at variable flow rates. This enables them to adapt the flow of water to the needs of the network, saving electricity and favoring condensation of boilers made with this technology. When the control valves close at the terminal level (e.g. the thermostatic valves for the radiators), the pressure increases and the circulator compensates for the pressure difference by decreasing its rotation speed. Moreover, it is no longer necessary to have a bypass on the circulator equipped with a manometer because the flow rate is displayed on the screen of the circulator.

The control system for the variable flow rate is generally integrated into the heating circulator by the manufacturer. Four modes of operation may be configured (see Figure 1.30):

- Fixed rate.
- Variable rate at constant differential pressure: the circulator's rotation speed varies to keep the manometric head at a constant set point, which can be adjusted. This mode of operation is suitable for a network with a low pressure drop, enabling the supply to reach the emitters at the far end of the network, while also decreasing electrical consumption.
- Variable rate at variable differential pressure: the circulator's rotation speed varies to keep the manometric head at a variable set point, which decreases in proportion to the flow rate. This solution is suitable for networks with high pressure drop and helps further reduce electrical consumption.
- Variable rate controlled by an automation or control system: in this case, a control law is programmed into the system.

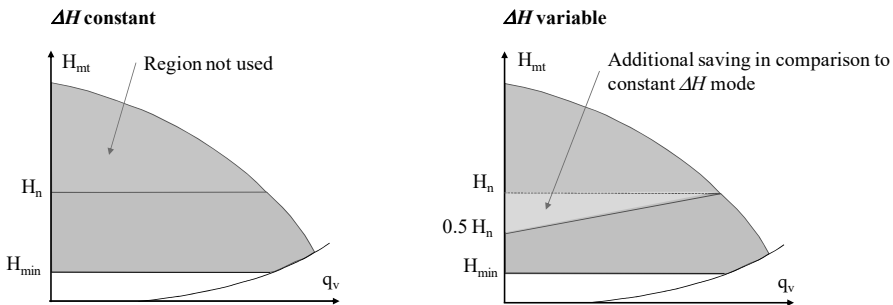


Figure 1.30. Variable-rate circulator at a constant ΔH and variable ΔH

For heating networks, the circulators generally work with a submerged rotor (i.e. the rotor is cooled by the water in the circuit). Electricity consumed varies with the cube of its rotation speed. An energy efficiency ratio (EER) is defined by the EU's Energy-related Products (ErP) Directive. This indicator, which does not have a unit, characterizes the system's efficiency. As of 1 January 2020, the EER of submerged rotor circulators must be less than or equal to 0.23.

During the season when heating is required, in intermittently occupied buildings, by judiciously lowering the ambient temperature when they are unoccupied (e.g. 16°C in the evenings and at weekends, and 8°C during the holidays), considerable

energy savings can be made. To manage these setbacks, the most efficient solution is to shut off the circulators of the heating networks. An optimizer can determine the optimal shut-down and restart times.

When the building is unoccupied, the circulators need to be restarted when:

- the outdoor temperature is lower than 5°C, to prevent the pipes from freezing, as pipes sometimes run outside the building or through unheated spaces;
- the ambient temperature drops below its set point.

On system restart, before the occupants arrive, the circulators restart, and then run for as long as the building is occupied.

1.3.2.2. *Miscellaneous equipment*

For radiator networks with thermostatic valves, a differential pressure valve at the output of the pump is crucial to ensure the circulator works properly when all thermostatic valves are closed. With variable-rate circulators, the flow rates vary between 20 and 100% of the nominal flow rate. For flow rates less than 20%, a differential pressure valve is still needed, or else it is necessary to equip some radiators with manual valves, which are constantly open.

The isolation and drainage valves at the bottom of the system are needed for maintenance. The check valve prevents water from circulating when the circulator of a circuit is off. The two thermometers on the supply and return pipes are used to monitor the network's performance. Automatic purge valves at the top of the system are used to bleed off gases.

1.3.3. *Storage*

To function correctly, certain heating systems, such as HPs, wood-fired boilers, CHP generators and thermal solar installations, require hot water storage tanks. The buffer tank (storage) sits between the production system and the point of delivery. There are two possible configurations for a storage tank: two-pipe tanks (one in and one out) and four-pipe tanks (two ins and two outs).

With HPs, manufacturers require the hydraulic network to contain enough water to allow the compressor to run at its minimum rate for between 5 and 10 min. Therefore, a buffer tank is generally needed. Thus, the number of times that the HP starts up is reduced, which improves its lifespan.

On wood-fired boilers, buffer tanks are recommended because they help:

- store the surplus energy released by combustion of the wood, when the building's needs decrease, and prevent the water temperature in the installation itself from rising too high;
- allow the generator to run at its minimum level for a period;
- store the energy needed to restart the system in the morning for intermittently occupied buildings;
- hydraulically separate the primary and secondary circuits.

The volume of the tank is determined by the amount of the thermal energy that needs to be stored. By increasing the temperature of the buffer tank T_b from $T_{b,initial}$ to $T_{b,final}$ with a volume of water V_b (m^3), the energy stored in (kWh) is calculated as follows:

$$E_{stored} = 1.16 V_b (T_{b,final} - T_{b,initial}) \quad [1.6]$$

Therefore

$$V_b = \frac{E_{stored}}{1.16 (T_{b,final} - T_{b,initial})} \quad [1.7]$$

A buffer tank gives rise to thermal losses. These can be attributed to heat exchanges through the tank walls, the support structure and the hydraulic connections. They are characterized by a cooling constant Cr ($\text{kWh} \cdot \text{day}^{-1} \cdot \text{m}^{-3} \cdot ^\circ\text{C}^{-1}$) or a heat loss coefficient. The cooling constant expresses the daily heat losses per cubic meter, per degree of difference between the water temperature and the temperature of the surrounding environment. The annual losses (kWh) are calculated thus:

$$E_{losses} = Cr N V_b (T_b - T_l) \quad [1.8]$$

where N is the number of days and T_l is the temperature of the space in which the tank is installed ($^\circ\text{C}$).

In French regulation RE 2020, an equivalent formula is used, which brings in the thermal loss coefficient, written as UA , expressed in ($\text{W}/^\circ\text{C}$):

$$E_{losses} = UA N 24 (T_b - T_l) / 1,000 \quad [1.9]$$

1.3.3.1. *Example of how to calculate the losses from a tank*

For a tank at an average temperature of 65°C, with a volume $V = 300$ L (height $H = 1.7$ m and diameter $D = 0.47$ m), in a space at a temperature $T_l = 15^\circ\text{C}$, insulated with a material whose thermal conductivity is $\lambda = 0.03 \text{ W.m}^{-1}.\text{K}^{-1}$, the losses are as follows.

Insulator thickness (mm)	40	100	160
Thermal losses (W)	128	56	33
Consumption due to energy losses (kWh/year)	1,121	490	289
Cost of losses (€/year) – EDF’s Tarif bleu, option based on 6 kVA, October 2023: €0.2276, inc. taxes, excluding subscription	255	111.5	65.8

Table 1.3. *Thermal losses from a DHW tank and corresponding cost*

In a buffer tank, the water temperature is not homogeneous. It is hotter at the top than at the bottom because its density decreases as its temperature increases. This physical phenomenon is characterized by vertical stratification. We must take account of this when connecting the tank to the hydraulic network.

The water outlet temperature for the secondary circuit (loads) is lower than the primary water inlet temperature (production). The water flow leaving the production system at 60°C is mixed with a small amount of the secondary returning stream at 55°C. The secondary supply temperature is cooled slightly to 59°C (Figure 1.31).

To limit such recirculation and sweeping, and encourage stratification that is conducive to proper operation, we must abide by certain rules:

- connect the flows requiring hotter water near the top of the tank;
- keep connections on the same level;
- ensure the height-to-diameter ratio of the tank is around 3;
- inject water at a slow rate;
- on the side going to the production system, the flow rate should be between 10 and 30% (four-pipe tank);
- the connections for production and loads should not cross.

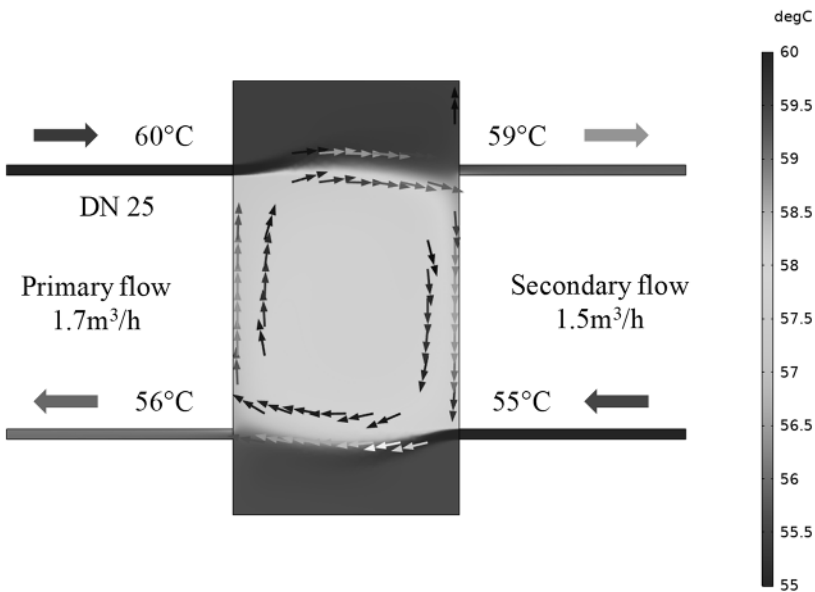


Figure 1.31. Temperature distribution in a four-pipe tank ($V = 300 \text{ L}$; $H = 1.15 \text{ m}$; $D = 0.60 \text{ m}$ /on primary flow 10%)

1.3.3.2. Calculating the necessary tank dimensions

A water-to-water heat pump provides heating to a set of apartments by means of underfloor heating. Its nominal thermal power is 70 kW. The heat pump is controlled, in on/off mode, by a thermostat on the water supply. The thermostat's target temperature is set at 45°C, with a hysteresis of 5°C. The manufacturer recommends that the heat pump be allowed to run for at least 10 min (see Figure 1.32).

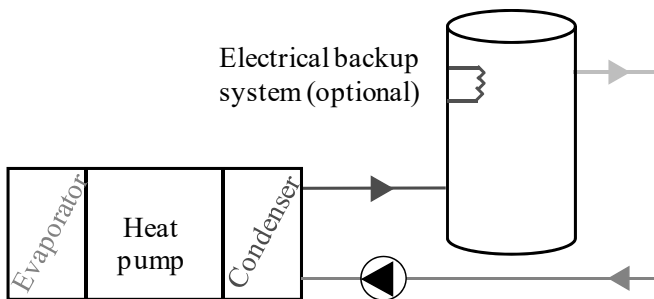


Figure 1.32. Water-to-water heat pump, coupled with a buffer tank

The thermal energy produced by the heat pump over 10 min is:

$$E_{Heat\ pump} = \frac{P_{nominal, heat\ pump}}{\Delta t} = \frac{70 \times 10}{60} = 11.67 \text{ kWh}$$

The heat pump starts when the water outlet temperature falls below 40°C and switches off again when it reaches 45°C. The volume of water contained in the installation must be sufficient to store this quantity of energy.

$$V_{water} = \frac{11.67}{1.16 (45 - 40)} = 2 \text{ m}^3$$

To determine the water contents of the installation as an initial estimation, we use the following values as a function of power:

- steel panel radiators: 11 L/kW;
- cast-iron radiators: 14 L/kW;
- underfloor heating: 23 L/kW;
- convectors: 6 L/kW;
- air-based systems (air conditioners, coils, etc.): 8 L/kW;
- heating systems in large-scale buildings: 20 L/kW.

In the scenario under consideration, the water contents amount to:
 $C_{water} = 70 \times 23 = 1.610 \text{ m}^3$.

The volume of the tank required, therefore, is:
 $V_{tank} = V_{water} - C_{water} = 2 - 1.61 = 0.39 \text{ m}^3$.

The tank, which has a volume of 400 L, has a cooling constant of 0.1 Wh/(day.m³.°C). The average temperature at which water is stored is 40°C in a room at 15°C. The thermal losses of the tank are:

$$E_{Losses} = 0.1 \times 232 \times 0.4 \times (40 - 15) = 232 \text{ kWh/year}$$

1.3.4. Safety

Safety mechanisms must be put in place for low-temperature hot water installations, in which the temperature is no higher than 105°C to protect people and keep the installation in good condition. NF DTU 65.11 (September 2007), entitled “Safety components for building central heating installations”, lists these various

devices. Notably, we can point to safety aquastats, low-water pressure switches and generator flow rate controllers; safety valves to limit excess pressure in the hydraulic circuit, expansion tanks to allow for variations in volume due to the water's expansion as its temperature rises; and disconnectors to protect the drinking water network.

Numerous other components, which are not among the mandated safety devices, are needed to allow the heating installation to function properly and be easy to maintain: isolation valves, drainage valves, filters, mud drums, air separators, purgers, thermometers, manometers, pressure reducers, meters, water treatment systems, etc.

1.3.5. Production

There are numerous heat production systems, using:

- fossil fuels, such as coal, fuel oil, natural gas or propane boilers, or CHP generators;
- electricity, such as electric boilers or HPs. Note that HPs also use a proportion of renewable energy;
- renewable energy, such as thermal solar installations, biomass boilers (using wood, straw, solid waste, biogas, etc.), and geothermal power.

These systems' energy performance depends, first, on the intrinsic characteristics of their components, and, second, on the hydraulic design, control systems and the systems' behavior.

Today, numerous installations adequately deliver heating to buildings, but they are not operating in optimal conditions.

Consider a few examples:

- Gas-fired condensing boilers which produce little or no condensation: their advertised efficiency is very high, but because of an unsuitable hydraulic design and control system, the water return temperature is often too high for condensation to occur, so the efficiency falls.
- Wood-fired boilers that do not release their full amount of power: for example, medium-powered hybrid boilers using wood and gas. When the water return temperature is too low, the wood-fired boiler shuts down, and the gas boiler takes over. Thus, the coverage by the wood-fired boiler is sub-optimal.

– Micro-CHP generators that do not operate for an optimal time period: these may be coupled with gas-fired boilers, for example. If the hydraulic design and control system is not suitable, the gas boilers, which are much more powerful than the micro-CHP generator, rapidly bring up the installation's temperature, causing the micro-CHP generator to shut down.

– Short-cycling of HPs: the water volume in the installation is not sufficient to store enough heat, and the HP is switched on too frequently to raise the temperature. A timer in the machine called an “anti-short-cycle timer” (set at between 5 and 10 min) prevents the HP from restarting before that time is up to prevent premature wear and tear on the compressor (the system may not start up more than six times an hour). Thus, if the installation contains insufficient water, it cannot heat the spaces to the required ambient temperature.

Let us look, in more detail, at gas-fired boilers, wood-fired boilers, micro-CHP generators and thermodynamic machines.

1.3.5.1. *Gas-fired boilers*

During the First Industrial Revolution in 1827, the first tubular boiler was developed by the French engineer Marc Seguin, for use in rail transport. The first municipal gas-fired boilers emerged in the late 19th century and began replacing coal-fired boilers. In France, in the 1950s, the discovery of the Lacq natural gas deposit (where operations were finally halted in 2013) allowed for the development of gas-fired boilers to heat the buildings that were rebuilt after WWII.

Depending on their (nominal) power, boilers can be divided into three categories:

- low power $P \leq 70$ kW (boilers for private homes or small multiple-occupancy buildings) and $70 < P \leq 400$ kW (boilers for collective buildings);
- medium power $400 < P \leq 1,000$ kW (for large collective buildings and industrial premises);
- high power $P > 1,000$ kW (for heavy industry, heating network, etc.).

In this section, we discuss low- and medium-power gas-fired boilers producing water at a relatively low temperature ($T \Delta 110^\circ\text{C}$) to supply central heating installations for a collective building.

A gas-fired boiler is a generator designed to produce heat by burning a gas to heat the transfer fluid: water. The combustion of natural gas using oxygen from the

air takes place in a combustion chamber. The walls of the boiler separate the water from the combustion chamber and the heat is transmitted:

- simultaneously by radiation between the flame (or the mass on ignition) and the internal surfaces of the chamber and by forced convection between the hot exhaust gases and the internal surfaces of the chamber;
- by conduction through the walls;
- by convection between the water and the external surfaces.

A second heat transfer zone, after the chamber, allows for additional cooling of the fumes. In this exhaust-to-water heat exchanger, the exchanges take place by:

- forced convection between the exhaust and the internal wall of the exchanger;
- conduction through the wall;
- forced convection between the wall and the water.

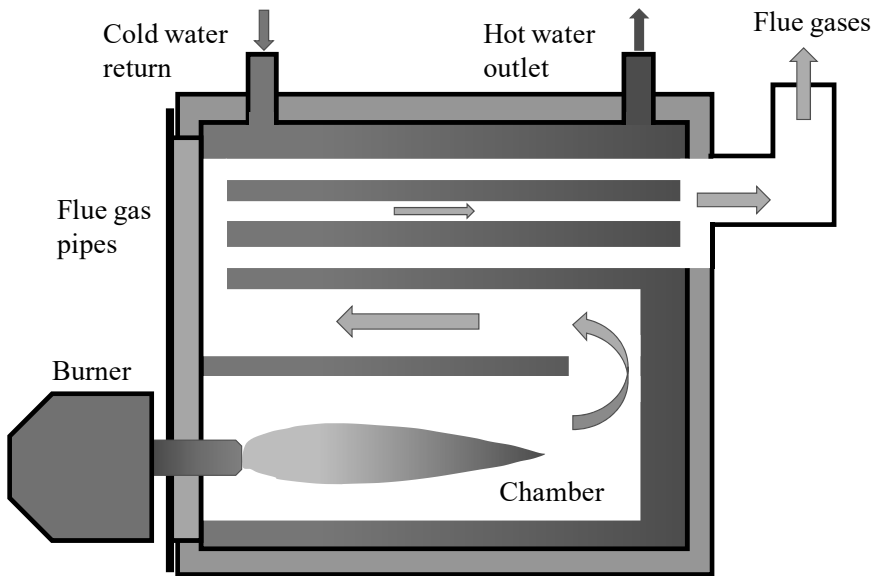


Figure 1.33. Example of a gas boiler with a three-pass exhaust tube

There are two major principles. The first is “exhaust tube” boilers, in which the exhaust gases circulate in the tubes and the water circulates around the outside (Figure 1.33) – this is the most widely used principle. The second is “water tube”

boilers, in which the opposite occurs: the water runs through the tubes and the exhaust circulates around the outside.

As combustion takes place in boilers, we need an air inlet to feed the flame with comburent. This is the role of the burner. We can distinguish between two technologies (Figure 1.34):

- Atmospheric burners: air is naturally entrained by induction at the gas injection buses (by the Venturi effect). Note that this technology is no longer commercially available, but it can be found in existing installations.

- Forced air burners: air is injected by a fan, and the chamber is pressurized. As the regulations have evolved, gas-fired boilers installed today are ventilated and condensing boilers.

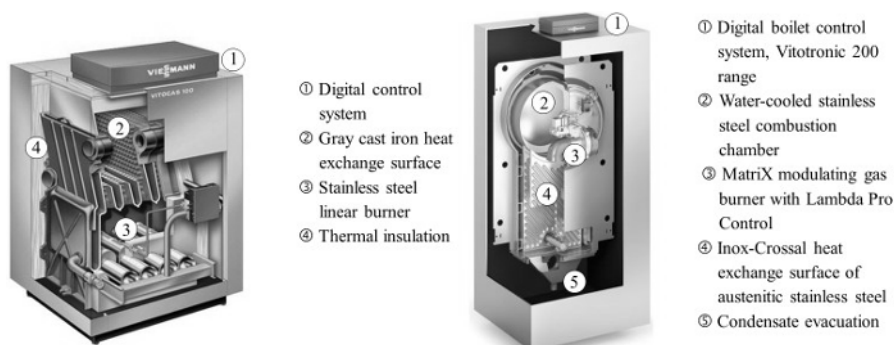


Figure 1.34. *Boiler with an atmospheric burner and ventilated air burner (source: Viessmann)*

The exhaust is generally evacuated through a flue by natural draw, mechanical extraction or the generation of pressure in the chamber, but certain low-powered boilers work without a flue, using the building's controlled mechanical ventilation system ("CMV gas"), or an airtight duct, potentially with a mechanical extractor ("ventouse" boilers).

Gas-fired boilers can be divided into three categories (Figure 1.35):

- regular boilers;
- low-temperature boilers;
- condensing boilers.

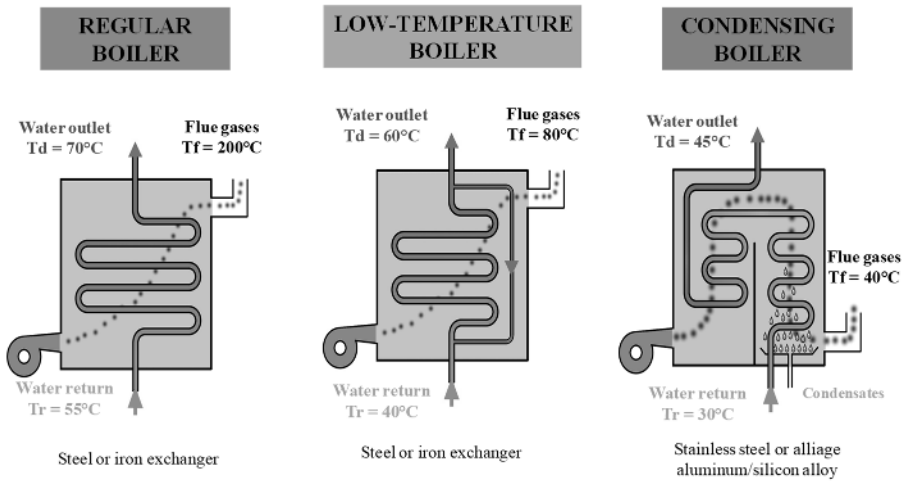


Figure 1.35. Regular, low-temperature and condensing boilers (Faure and Moimeaux 2018)

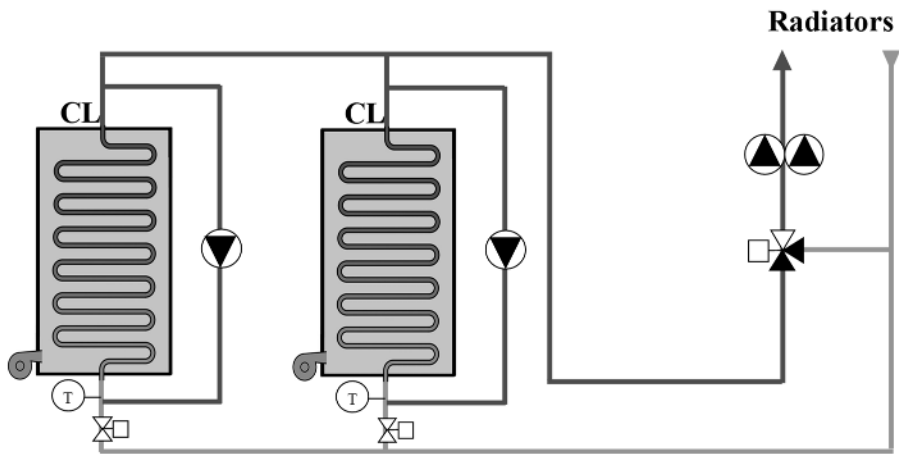


Figure 1.36. Simplified schematic view of a boiler room with regular boilers and a recirculation pump for protection from unsuitable return temperatures. CL = Classic Lambda

Regular boilers, developed in the 1920s, contain a cast-iron or stainless-steel heat exchanger. The water returning to the gas boiler must be around 55°C at least to prevent the water vapor in the exhaust gases from condensing. The exhaust gas

temperature is around 200°C . These are old boilers with an instantaneous LHV efficiency of around 92% (LHV = lower heating value).

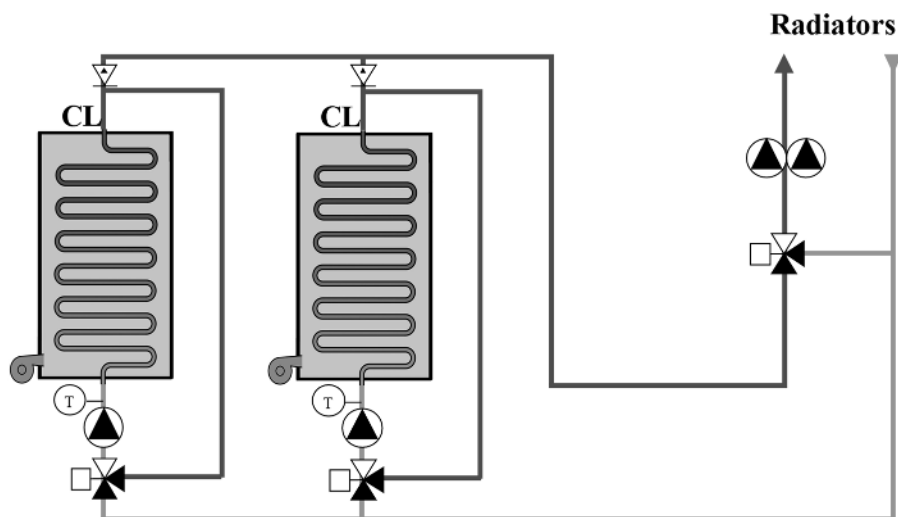


Figure 1.37. *Simplified schematic view of a boiler room with regular boilers with a feed pump and a three-way valve for protection from unsuitable return temperatures*

To protect the boiler from excessively low return temperatures, the solution generally adopted is to raise the return water temperature by mixing it with hot water taken directly from the boiler outlet. This is done by a recirculation circuit, with a specific circulation pump. Depending on where that pump is placed, we name it either a recirculation pump (Figure 1.36) or a feed pump (Figure 1.37). A temperature probe at the boiler inlet monitors the temperature, and a minimum of 55°C is set for gas boilers.

Depending on the setup, when the return temperature is greater than or equal to the minimum acceptable temperature:

- the recirculation pump is stopped to save energy;
- the three-way valve closes gradually, and the flow rate in the recirculation bypass decreases. The feed pump operates constantly while there is demand on the boiler.

Low-temperature boilers are high-efficiency boilers that further exploit the principle on which regular boilers are based. The instantaneous efficiency, 93–95%

LHV, is slightly higher than that of a regular boiler. First, a larger exhaust-to-water heat exchanger than that of a regular boiler increases the amount of heat that can be recovered from the exhaust (output at around 100°C); second, the system is designed to operate at a low water temperature.

When the temperature of the water in the boiler decreases, its efficiency increases, due to better cooling of the exhaust, decreased losses through the walls, and a lower thermal maintenance coefficient.

However, with this system, there is the danger of condensation, which is not desirable with this technology. Low-temperature boilers practically always use an internal pre-mixing system, capable of dealing with a very low return temperature: 35°C, for example (Figure 1.38).

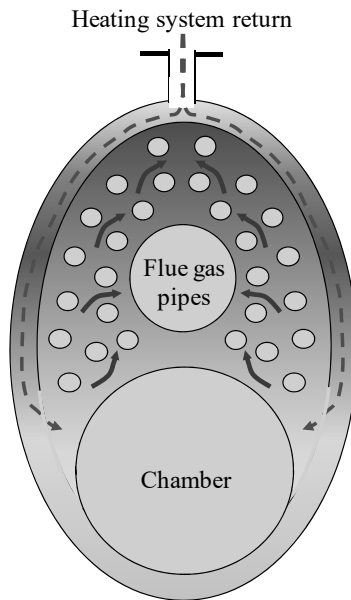


Figure 1.38. *Cross-sectional view of a low-temperature boiler*

The return water enters the element at the top (shown by the blue circle). It is directed to the outside of the element (the blue, yellow and green corona). Thus, it only passes through the shell and the exhaust ducts once it has been heated (red zone).

However, it is difficult to prevent transitory condensation when the burner restarts after a period of stoppage. There are two factors that designers may consider to prevent premature corrosion:

- choosing corrosion-resistant materials, such as cast iron (“gray” eutectic cast iron) or certain steel alloys;
- low inertia of the shell walls, ensuring the flame can quickly raise the boiler’s temperature.

During operation, it is also possible to avoid condensation by a judicious design of the heat exchanger walls. Recently, walls with non-uniform thermal resistance have been used in boilers that operate at low temperatures without condensation. To produce heat at low temperatures without causing the water to condense, the thermal resistance of the wall increases as the exhaust progresses through the exchanger to constantly keep the temperature of the wall on the exhaust side above the dew point. Manufacturers also use a range of techniques, such as fins with variable efficiency, or double walls separated by an air gap (see Figure 1.39).

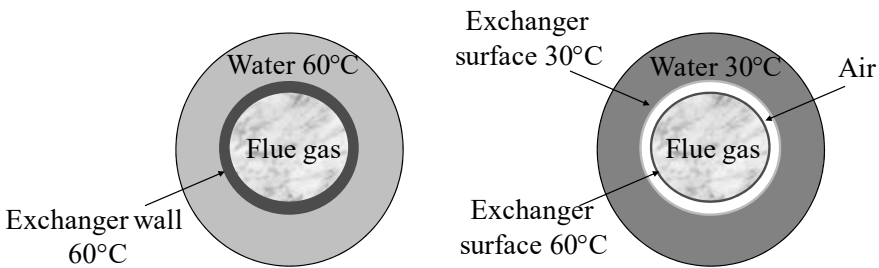


Figure 1.39. *Double-walled pipe with an air gap*

To benefit from the improved efficiency thus obtained, the boiler needs to behave differently from regular boilers:

- It is not helpful to recirculate the water or to use any device which increases the inlet temperature.
- The boiler produces water at a variable temperature, as close as possible to the temperature needed by the most demanding secondary network. This system is known as a tracker. If the boiler manufacturer stipulates a minimum output temperature, this must be taken into account when setting up the control system.

The hydraulic arrangement of the boiler depends on its characteristics. Manufacturers produce boilers with high water capacity (HWC) and low water capacity (LWC). The ratio, in liters per kilowatt, is the value that best represents a boiler's water content. It is the ratio of the amount of water the boiler contains to its nominal power. If that ratio is around 0.1 L/kW, the boiler has low water capacity and low inertia. If the ratio is greater than 1 L/kW, it is described as having high water capacity and therefore high inertia.

For an HWC boiler, there are no constraints on flow rate. The hydraulic setup is considerably simpler. There is no feed pump for each boiler, and thus no hydraulic decoupling tank. The pumps of secondary networks supply water to the boilers (see Figure 1.40). When the buildings' heating demands decrease, the 3-way valves of the secondary networks close, and the flow rate to the boilers falls (possibly even to zero). The burner power is modulated and, because of the high volume of water in the boiler, this scenario does not pose a problem.

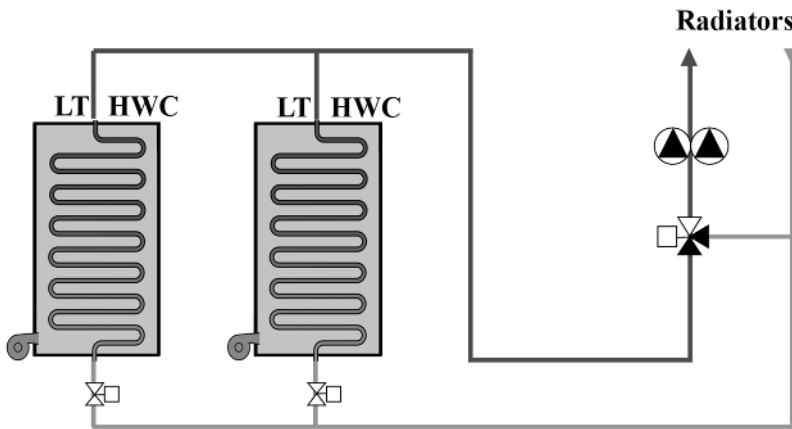


Figure 1.40. *Simplified schematic view of a boiler room with HWC low-temperature boilers*

When a boiler has a low water capacity (ratio = 0.1 L/kW), manufacturers generally impose a fixed flow rate. In this case, a feed pump is needed for each boiler, and a hydraulic decoupling tank provides hydraulic separation of the production apparatus and the secondary networks (Figure 1.41).

For boilers with a medium water capacity (ratio = 0.5 L/kW), a minimum flow rate is required. A recirculation pump that is smaller than the feed pump ought to be used; the HS is not needed.

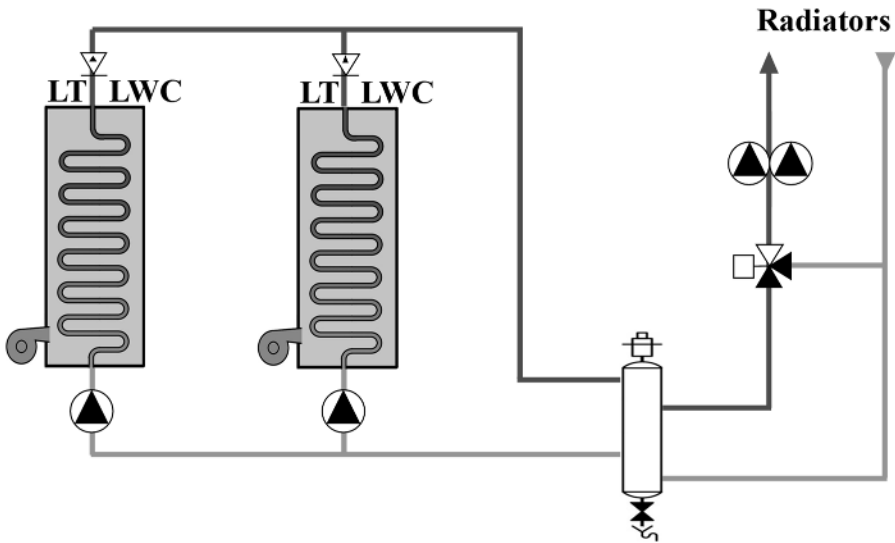


Figure 1.41. *Simplified schematic view of a boiler room with low-temperature, LWC boilers*

Condensing boilers were developed in the 1980s, in the wake of the first oil crisis in 1973. In France, the market truly took off following the passing of the thermal regulation in 2005.

Their special design means they can deliver much greater efficiency by harvesting the latent energy from the water vapor created during combustion. They can achieve efficiencies of 103–104% LHV. With the same level of power input, a condensing boiler consumes less fuel than a low-temperature boiler.

Fundamentally, condensing boilers recover as much of that latent heat as possible by cooling the exhaust to the fullest extent possible. This type of boiler poses the following technical problems:

- the risk of corrosion of the boiler's metal walls;
- the evacuation of acidic condensates (pH 4–5);
- evacuation of the exhaust as, once cooled, it is no longer able to generate natural draw (due to its low temperature), which becomes practically non-existent.

In addition, the installation must be designed so as to obtain as low a temperature as possible when the water enters the boiler to produce significant condensation from the exhaust gases.

Natural gas is composed of multiple gaseous hydrocarbons – notably methane, CH_4 (between 83 and 97%). The combustion of methane with atmospheric oxygen forms carbon dioxide (CO_2) and water vapor (H_2O).

In standard temperature and pressure conditions (STP) (0°C and $p_{\text{atm}} = 101,325 \text{ Pa}$), stoichiometric or ideal combustion (i.e. complete combustion without excess air) of natural gas is illustrated by Figure 1.42.

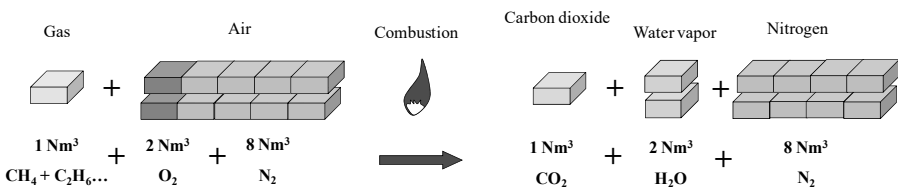


Figure 1.42. *Stoichiometric combustion of natural gas*

The thermal energy released by this combustion is first passed onto the boiler and, second, retained in the high-temperature exhaust gases. If we can harvest the energy from those gases by cooling them to ambient temperature, then theoretically, we have harnessed all the energy contained in the fuel, wasting none. This is known as the heat of combustion or heating value. However, as we saw earlier, the exhaust contains H_2O in the gaseous state. If we cool the exhaust, the water may condense into liquid, thereby ceding energy: the heat of vaporization or latent energy. If we can recover that energy, then we refer to the higher heating value (HHV). If, on the other hand, while recovering energy from the exhaust, we are not able to harness the latent energy from the water vapor, we speak of the lower heating value (LHV). The heat of combustion of a fuel is the amount of heat released by the complete combustion of one normal cubic meter (1 Nm^3) of gaseous fuel at 0°C , with the combustion products themselves being expressed in relation to the normal temperature and pressure conditions.

There are two different heating values:

- The LHV is the amount of heat recovered if the water in the exhaust is in the vaporous state.
- The HHV is the amount of heat released if the water in the exhaust condenses.

For H-gas: $\text{HHV} = 11.4 \text{ kWh/Nm}^3$ and $\text{LHV} = 10.3 \text{ kWh/Nm}^3$, so $\text{HHV} = \text{LHV} + 10.7\%$.

A condensing boiler delivers an energy gain on two levels (Figure 1.43):

- Latent heat gain: the condensation of the water vapor in the exhaust releases energy. For a gas boiler, the maximum such gain is 11% of the LHV.
- Sensible heat gain: the lowered temperature of the exhaust is recovered through the surface of the exchanger (from 150°C to 45°C).

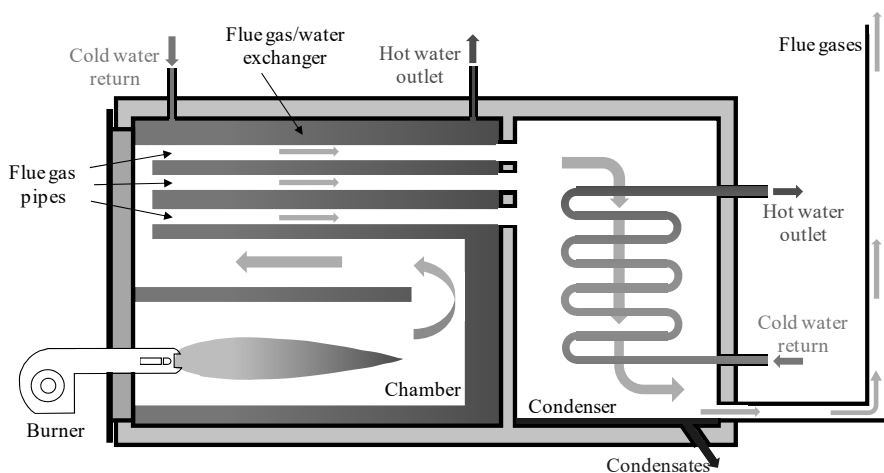


Figure 1.43. *Principle of a four-pipe condensing boiler*

To obtain optimal efficiency, we need:

- good combustion;
- as low a water return temperature as possible;
- a minimum loading rate;
- suitable generator power;
- suitable generator technology, hydraulic design and control system.

Good combustion must be as close as possible to stoichiometric or ideal combustion. The dew point for natural gas in such conditions is 59°C (Figure 1.44). If the water when it returns to the boiler is below 59°C , the water vapor in the exhaust will begin to condense. However, to obtain complete combustion, a slight

excess of air, between 10 and 30%, is crucial. The dew point decreases (to 55°C, for example) and, in order for the water vapor in the exhaust to condense, the water returning to the boiler must be cooler; during the season when heating is in demand, this will rarely be the case. Thus, the yearly efficiency will decrease. To stay abreast of the excess air involved in the combustion, a probe can be used to measure the level of oxygen (O_2) in the flue gases.

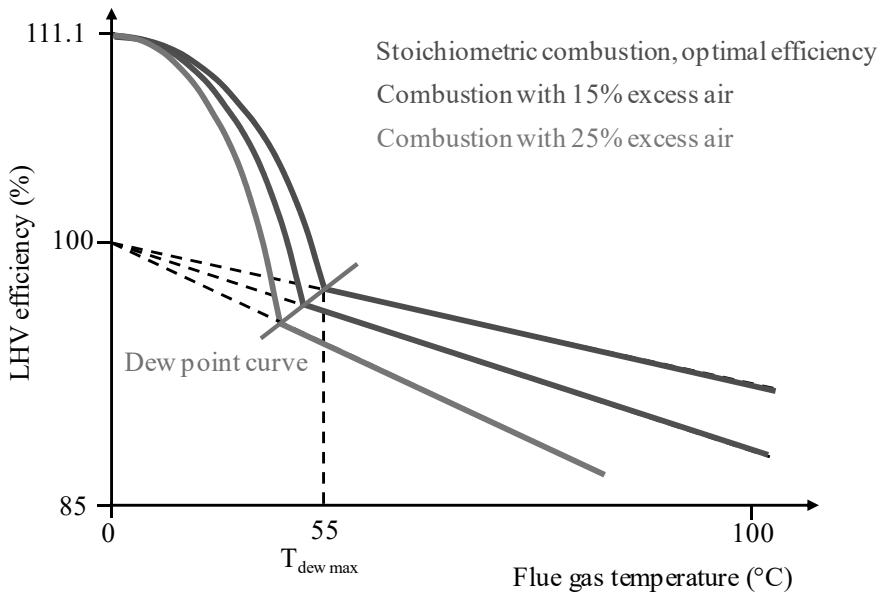


Figure 1.44. LHV efficiency as a function of the exhaust temperature and the excess air

The exhaust temperature needs to be as low as possible (Figure 1.45). Consequently, the temperature of the water when it returns from the heating circuit must be as low as possible. This can be achieved by a proper design of the hydraulic circuit, but primarily by having a lower operating temperature and using low-temperature heat emitters.

The load ratio, or burner modulation rate, must be minimal (see Figure 1.45). In a cascade of two condensing boilers with modulating burners, it is preferable to run the two boilers simultaneously at partial load (50%, for example), rather than running a single boiler at 100%. The exchange surface used is twice as large, so the exhaust is cooled better, favoring water vapor condensation and improving efficiency. For example, for a water return temperature of 45°C, the efficiency is

102%, whereas at 100% (maximum power), it is 97.5%. Thus, running the two boilers simultaneously at partial loads delivers a 4.5% gain in efficiency.

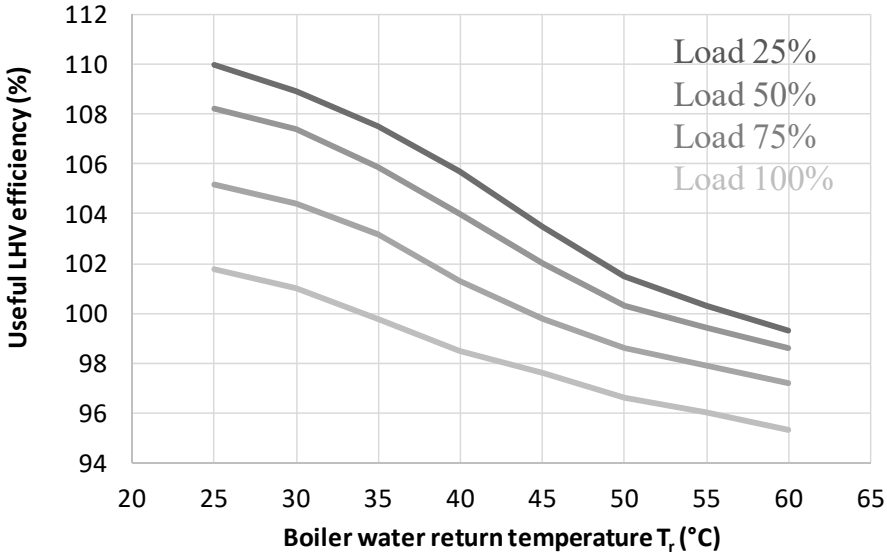


Figure 1.45. LHV efficiency as a function of the water temperature and the part-load ratio (Sébastien and Atlantic Guillot 2011)

To ensure continuity of service, it is advisable to design boiler rooms with multiple boilers rather than only one. The installed power must be sufficient to cover the heating, DHW and the various thermal losses of the networks. Surplus power may be required to cater for heating in very cold weather, and in intermittently occupied buildings, for the morning restart. However, this overpower must be optimized to limit the start/stop cycles of the modulating burners (modern burners modulate between 20 and 100%) and reduce losses due to pre-ventilation each time they fire up, and the losses each time they shut down. This helps improve seasonal efficiency.

Suitable generator technology and hydraulic design: to resist corrosion due to acid condensates, the exchangers in condensing boilers are made of stainless steel or aluminum alloys. Stainless steel exchangers generally have a high water capacity, while aluminum alloy exchangers have low water capacity. Manufacturers offer three types of condensing boilers: two-connector boilers, three-connector boilers and four-connector boilers (see Figure 1.46).

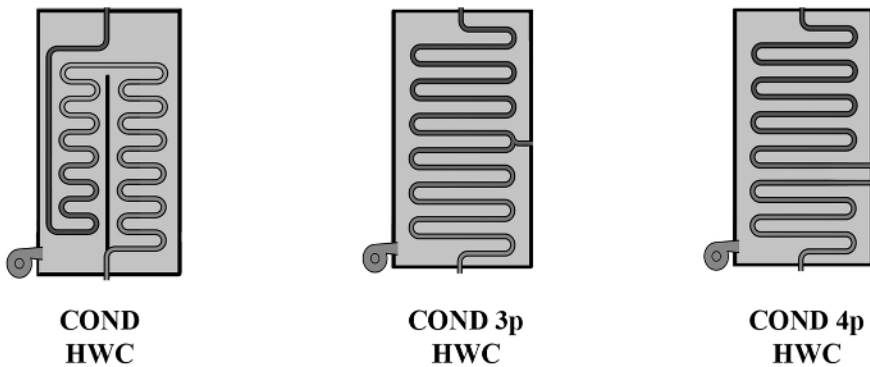


Figure 1.46. Two-, three- and four-connection condensing boilers

Two-connector condensing boilers comprise a single heat exchanger/condenser with one water outlet and one water inlet. These types of boilers are recommended if the networks run at a low temperature (i.e. when the water return temperature is lower than 40°C). Such is the case with carefully controlled networks with tailored operating conditions, underfloor heating (typically operating at $32/27^{\circ}\text{C}$), or radiators (operating typically at $65/50^{\circ}\text{C}$ or $55/45^{\circ}\text{C}$). The hydraulic setup of HWC boilers (Figure 1.47) is very simple. With the secondary networks controlled by a three-way valve with a “heating curve” and generators controlled by a “tracker system”, the water output and return temperatures are low, and efficiency is optimal.

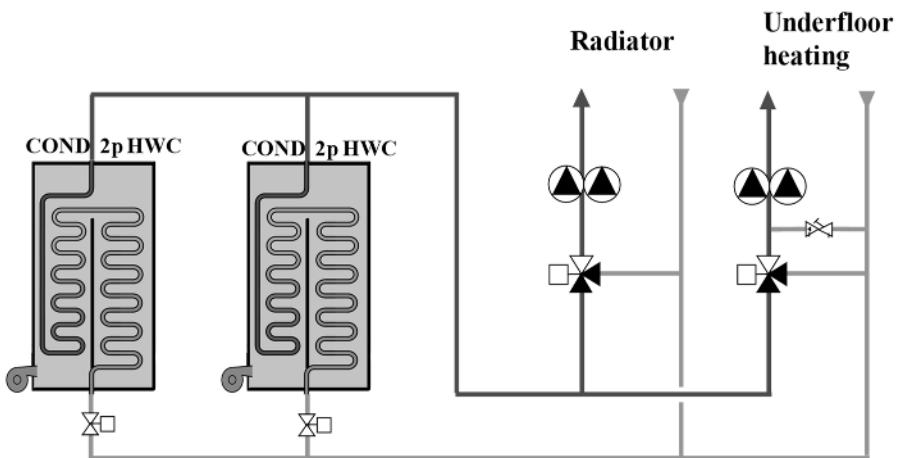


Figure 1.47. Simplified schematic view of a boiler room with two-connector, HWC boilers

For LWC condensing boilers (see Figure 1.48), the manufacturer sets a fixed or minimum inflow rate. Therefore, an HS is needed. When the flow rate to the primary circuit is greater than the secondary flow rate, then a flow in the HS heats up the returning water, which is not conducive to condensation. It is necessary to have a specific flow rate control system.

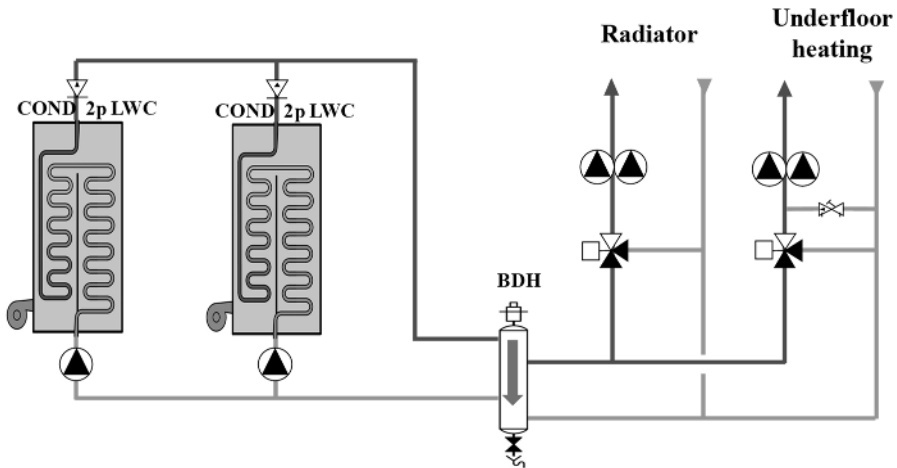


Figure 1.48. *Simplified schematic view of a boiler room with two-connector LWC boilers*

In the case of a heating system that includes both high-temperature networks with a high demand (e.g. for DHW) and low-temperature networks, three- or four-connection boilers may be needed.

Three-connection condensing boilers (Figure 1.49) have one out and two separate in (return) pipes: one for high temperature and the other for low temperature. This may be required when the installation has a high-temperature controlled network (e.g. radiators, operating at 80/60°C) and a low-temperature controlled network (e.g. underfloor heating, operating at 40/35°C). The separate returns promote condensation and improve efficiency.

Four-connection condensing boilers (Figure 1.50) have a different type of design. The chamber may be made up of steel, and the stainless steel condenser is separated hydraulically from the chamber. The high-temperature circuit (e.g. DHW operating at 80/60°C) is connected to the steel chamber, and the low-temperature network (e.g. a radiator network operating at 55/45°C) is connected to the condenser, enabling condensation to take place throughout the heating season.

However, the hydraulic setup and control system are more complex. During summer, only the DHW network is operational. Water is no longer supplied to the condenser directly. A motorized three-way valve can then be used to redirect the flow of water returning from the DHW network into the condenser.

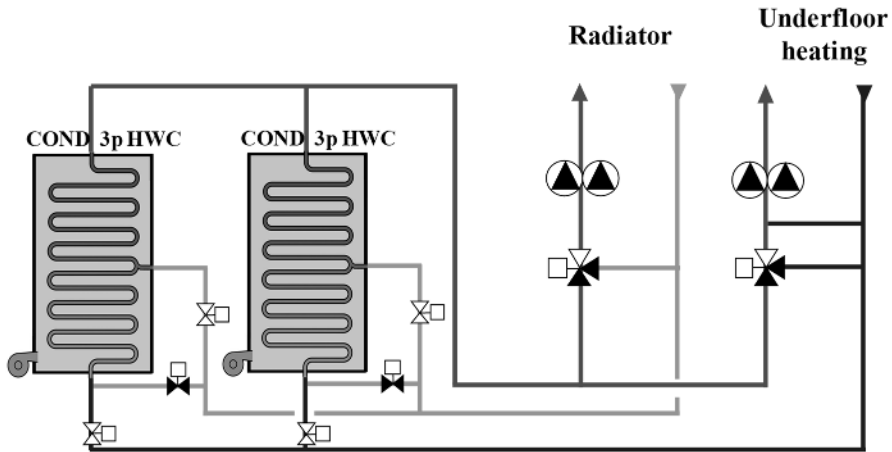


Figure 1.49. *Simplified schematic view of a boiler room with HWC three-connection boilers*

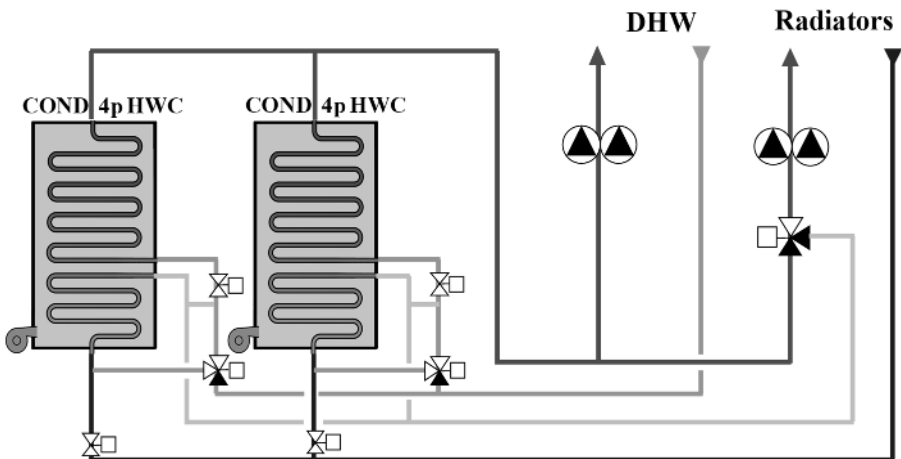


Figure 1.50. *Simplified schematic view of a boiler room with HWC four-connection boilers*

1.3.5.1.1. Production efficiency

The production efficiency describes the energy efficiency of an energy-production system. Such a system will be made up of one or more generators with varying characteristics. We distinguish between two types of efficiency.

– *The generator's useful thermal output η_U* is equal to the ratio of useful power transferred to the water P_U to the power absorbed by the boiler P_A in steady-state conditions:

$$\eta_U = \frac{P_U}{P_A} \quad [1.10]$$

The useful power in kW transferred to the water can be calculated by measuring the flow rate and water temperatures at the generator:

$$P_U = 1.16 q_{v \text{ water}} (T_d - T_r) \quad [1.11]$$

The volumetric specific heat of water is 1.16 kWh/(m³·°C); the volumetric flow rate of the water is written as $q_{v \text{ water}}$ (m³/h), and T_d and T_r are the generator outlet and return water temperatures in °C.

The power absorbed by the generator in (kW) is calculated on the basis of the measured fuel flow rate by:

$$P_A = q_{fuel} LHV \text{ (or HHV)} \quad [1.12]$$

The LHV and HHV are expressed in kWh/Nm³ for liquid and gaseous fuels and in kWh/kg for solid fuels. The fuel input rate q_{fuel} is expressed in Nm³/h for liquid and gaseous fuels and in kg/h for solid fuels.

If the fuel is characterized by its LHV, we obtain the “LHV efficiency” $\eta_{U/LHV}$. For example, this metric is used in regular boilers where the latent heat of condensation is not recovered from the water vapor in the exhaust. For condensing generators, it is the “HHV efficiency” $\eta_{U/HHV}$, which must be used.

$$\begin{aligned} \eta_{U/LHV} &= \frac{1.16 q_{v \text{ water}} (T_d - T_r)}{q_{fuel} LHV} \text{ and } \eta_{U/HHV} \\ &= \frac{1.16 q_{v \text{ water}} (T_d - T_r)}{q_{fuel} HHV} \end{aligned} \quad [1.13]$$

Thus:

$$\eta_{U/LHV} = \eta_{U/HHV} \frac{HHV}{LHV}$$

For gas:

$$\frac{HHV}{LHV} = \frac{11.4}{10.3} = 1.11$$

– *The generator's overall thermal efficiency η_G* (sometimes incorrectly referred to as the combustion efficiency) is the overall ratio of power transferred to the water P_U and to the environment P_P , to the power absorbed by the generator P_A , in steady-state conditions. This metric comes into play when the boiler is in a space that needs to be heated (e.g. a domestic boiler installed in the kitchen), as exchange through the boiler's walls no longer constitutes a heat loss – the energy contributes toward the heating of the building.

$$\eta_G = \frac{P_U + P_P}{P_A} \quad [1.14]$$

The useful power transferred to the water can also be determined indirectly by evaluating the following various losses.

Heat loss from the exhaust P_F : $P_F = P_{FS} + P_{FL} + P_{FI}$

– Heat loss from the exhaust in the form of sensible heat P_{FS} : the combustion products are evacuated at a certain temperature, which depends on the generator technology.

– Heat loss from the exhaust in the form of latent heat P_{FL} : the exhaust from the combustion contains water vapor, which is energy in latent form. In a condensing generator, a portion of that latent energy can be harvested. In regular generators, all the energy is lost.

– Losses due to unburned products in the exhaust P_{FI} : the exhaust may contain carbon monoxide (CO), unburned carbon, traces of hydrogen formed by the dissociation reactions and traces of hydrocarbons not burned or created by recombination. All these products are evacuated with the exhaust, although had they been fully oxidized, they would have released a certain amount of heat that is part of the original LHV. If combustion is homogeneous, these losses are negligible.

In the case of homogeneous combustion, the exhaust losses P_F in kW can quickly be evaluated using the semi-empirical Siegert formula. The simple formula can be

used to directly calculate the overall efficiency of a boiler η_G using an electronic analyzer, measuring the level of carbon dioxide in the flue gases γ_{CO_2} in (%), the temperature of the flue gases T_F and the temperature of the air aspirated into the burner T_A .

$$P_F = 0.01 P_A K \frac{T_F - T_A}{\gamma_{CO_2}} \quad [1.15]$$

The Siegert factor K depends on the fuel type:

- $K = 0.47$ for natural gas;
- $K = 0.31$ for propane;
- $K = 0.53$ for butane;
- $K = 0.57$ for domestic fuel oil;
- $K = 0.59$ for heavy fuel oil;
- $K = 0.78$ for wood.

Heat loss due to residues P_R : $P_R = P_{RS} + P_{RI}$

– In sensible heat in solid residue P_{RS} : this is inert hot ash created by the combustion in the case of solid fuels.

– In unburned residue P_{RI} : the ash may also contain unburned products, in the form of carbon, tar and various heavy hydrocarbons, which have an inherent heating value. In the case of liquid or gaseous fuels, residue losses are non-existent.

Environmental losses P_p : These losses are attributable to all the parts of the generator that exchange heat with the space in which the generator is installed by both radiation and convection. Environmental losses are difficult to assess and depend on the quality of the generator's thermal insulation, the temperatures of the walls, the value of the surface heat exchange coefficients and the temperature of the space. Generally, they amount to around 0.5% of the absorbed power for modern generators and up to 3% for older ones.

The steady-state heat balance of the generator in kW is written as:

$$P_A = P_U + P_F + P_R + P_P$$

$$P_U + P_P = P_A - P_F - P_R$$

$$\eta_G = \frac{P_U + P_P}{P_A} = \frac{P_A - P_F - P_R}{P_A}$$

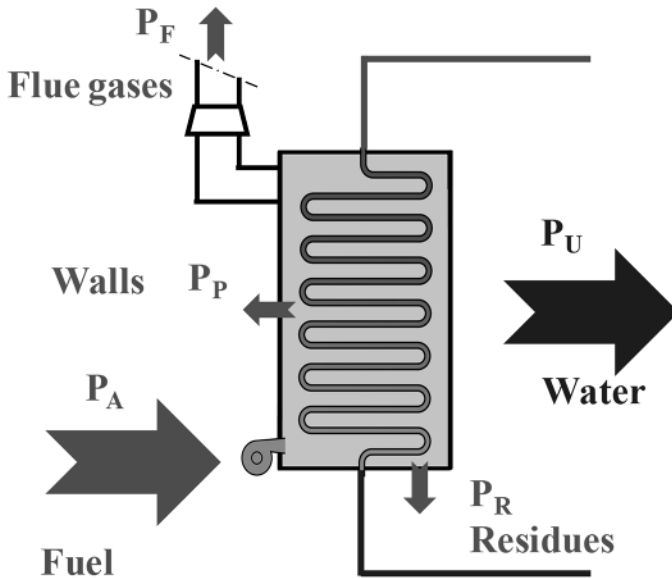


Figure 1.51. Heat balance of a boiler

In the case of liquid or gaseous fuels, the residue losses are zero, and the overall efficiency is calculated by the following simplified formula:

$$\eta_G = \frac{P_A - P_F}{P_A} = 1 - \frac{P_F}{P_A} = 1 - 0.01 K \frac{T_F - T_A}{\gamma_{CO_2}} \quad [1.16]$$

1.3.5.1.2. Calculating boiler efficiency: an example

In a series of analytic measurements to evaluate the efficiency of a regular gas boiler (natural gas), in steady-state, we measure the following parameters:

- flue gas temperature T_F : 180°C;
- burner aspirated air temperature T_A : 15°C;
- CO₂ level in flue gases γ_{CO_2} : 8%;
- boiler water flow rate $q_{v \text{ water}}$: 21.6 m³/h;
- boiler output water temperature T_d : 80°C;

- boiler inlet water temperature T_r : 60°C;
- gas flow rate q_{fuel} : 54.46 Nm³/h.

$$P_U = 1.16 \times 21.6 \times (80 - 60) = 501 \text{ kW}$$

$$P_A = 54.46 \times 10.3 = 561 \text{ kW}$$

$$\eta_{U/PCI} = \frac{P_U}{P_A} = \frac{501}{561} = 89.3\%$$

$$P_F = 0.01 P_A K \frac{T_F - T_A}{\gamma_{CO2}} = 0.01 \times 561 \times 0.47 \times \frac{180 - 15}{8} = 54.4 \text{ kW}$$

$$P_P = P_A - P_U - P_F = 561 - 501 - 54.4 = 5.6 \text{ kW}$$

$$\eta_{G/LHV} = \frac{P_A - P_F}{P_A} = \frac{P_U + P_P}{P_A} = \frac{501 + 5.6}{561} = 90.3\%$$

The instantaneous useful LHV efficiency is 89.3%. The flue gas losses are 9.7%, and the losses through the walls represent 1% of absorbed power. The annual production efficiency is also known as the seasonal efficiency η_s . Based on the useful energy from water measured by a thermal energy meter E_U in kWh and the energy absorbed in fuel E_A in kWh across a heating season, the seasonal efficiency is:

$$\eta_s = \frac{E_U}{E_A} \quad [1.17]$$

Over the course of a heating season, the generator usually runs at part-load to respond to the heating demand of the spaces for given weather conditions and internal loads. It is therefore necessary to know how the boiler behaves when it needs to deliver thermal power below its nominal power. Three scenarios need to be considered.

- The generator has an on/off burner: it operates at part-load by alternating periods of full-power operation and complete stoppage. When it is in use, the nominal efficiency is reached quickly (see Figure 1.52). On the other hand, when stopped, the boiler acts as a heat emitter, exchanging energy with the surrounding environment and the air, which continues to pass through the chamber as a result of

the flue's natural draw. The power P_A lost by the water while the burner is at rest is not equal to the power P_P lost through the walls while the boiler is operational. Indeed, when the burner is off, the losses from the water to the environment remain the same (losses through walls in contact with the water), but there are no longer any losses between the flue gases and the environment (losses through dry walls).

In addition, while the burner is off, the water also loses heat through the walls of the chamber and through the exchanger in which the air circulates by natural convection and escapes through the flue. A shutter to block the air circuit when the burner shuts off can decrease these losses but not eliminate them entirely.

- The generator has a burner with two profiles:
 - operation at nominal profile (“On” or “All”);
 - operation at a reduced rate – for example, at half its nominal power (“Little”);
 - complete stoppage (“Off” or “Nothing”).

When the need falls below the reduced rate, the burner works in cycles, alternating between “Little” and “Off”. Thus, the efficiency decreases, depending on the stoppage time. At the reduced rate, the burner is constantly on, in “Little” mode. There is no more stoppage, and the efficiency returns to its maximum level.

When the need exceeds the reduced profile, the burner alternates between “On” and “Little”. Although there is no stoppage, the efficiency falls slightly when the need increases because the efficiency at full power is slightly lower than the efficiency at reduced power (see Figure 1.52). Indeed, the flue gas is cooled more at the reduced rate, as the flue gas-to-water heat exchanger is oversized in this scenario. The setup on a regular boiler of a two-mode burner is therefore an advantageous compromise between a simple burner with a single mode (on/off) and a modulating burner, which is more costly and more complex to install.

- The generator is fitted with a modulating burner: it operates at part load, but in a steady regime, it varies very slowly between 30 and 100%. By decreasing the flow of fuel, we see a proportional decrease in the flow of flue gases. The boiler's exhaust-to-water heat exchanger is designed for operation under nominal conditions. Thus, with a reduced flow of fuel, the exchanger is oversized and the exhaust outlet temperature is lower. Thus, the efficiency is optimal at a part load (Figure 1.52).

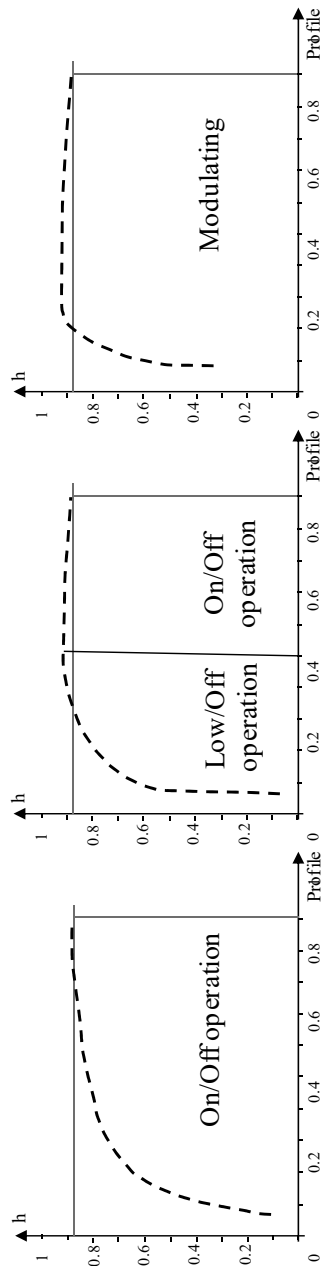


Figure 1.52. Evolution of the efficiency as a function of the burner's profile

1.3.5.1.3. Calculating seasonal efficiency: an example

A cascade of two condensing gas boilers with modulating burners supplies heat to a tertiary-sector building. The installation has a gas meter and a thermal energy meter, measuring the useful energy transferred to the water at the cascade output. Over the course of a heating season, the measured gas consumption was 32,800 m³ and the useful energy transferred to the water was 333 MWh. The gas is delivered at 20 mbar and an average of 15°C.

The volume of gas consumed in real conditions ($T_1 = 273 + 15 = 288$ K and $p_1 = 101,325 + 2,000 = 103,325$ Pa) must be adjusted to correspond to the same normal conditions ($T_0 = 273$ K and $p_0 = 101,325$ Pa) as the HHV:

Volume of gas consumed at normal temperature and pressure:

$$V_0 = V_1 \frac{p_1 T_0}{p_0 T_1} = 31,705 \text{ Nm}^3$$

Gas energy consumed at HHV:

$$E_{A/HHV} = 31,705 \times 11.4 = 361,437 \text{ kWh}$$

Useful energy transferred to water:

$$E_U = 333,000 \text{ kWh}$$

Seasonal HHV efficiency:

$$\eta_{S/HHV} = 100 \frac{E_U}{E_{A/HHV}} = 92.1\%$$

Seasonal LHV efficiency:

$$\eta_{U/LHV} = \eta_{U/HHV} \frac{HHV}{LHV} = 92.1 \times 1.11 = 102.2\%$$

In the case of energy production with condensing boilers, the HHV efficiency is the relevant factor: 92.1%. An efficiency greater than 100%, such as that quoted for the LHV – in this case, 102.2% – does not take account of the latent energy of condensation of the water vapor in the flue gases.

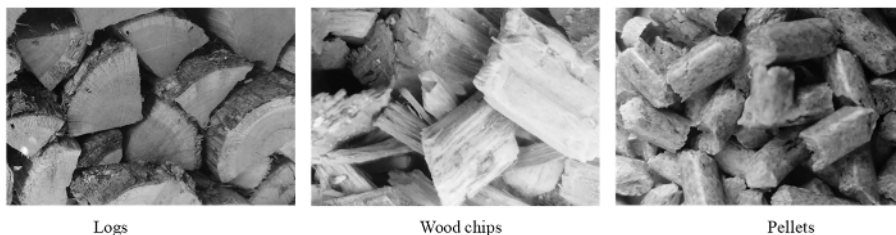
1.3.5.2. Wood-fired boilers

1.3.5.2.1. Context and issues

France has around 15 million ha of woodland (accounting for approximately 28% of the nation's territory), which are largely privately owned (around 75%). It is the third most forested country in Europe, behind Sweden and Finland. Biomass is France's main source of renewable energy, used primarily to generate thermal power. The breakdown of the vegetation growth in France is as follows: around 44% for industrial use (timber, furniture, paper, etc.), 26% for firewood, and 30% remaining uncut, expanding the area of forest. France's forests grow every year (+ 0.4% surface area per year), and an additional 4.5 Mtoe per year can be used without impacting the sustainability of forestry. Wood has long been recognized as a viable alternative fuel source.

One of the benefits of using wood is that it has a low environmental impact. Indeed, the CO₂ emitted in the combustion process was originally sequestered by the growing vegetation during photosynthesis. Therefore, burning wood does not cause an increase in the overall quantity of CO₂, as burning fossil fuel does. Using wood, therefore, is a step in the right direction in decreasing greenhouse emissions. Using 4m³ of wood, we would save 1 toe and prevent the emission of 2.5 metric tons of CO₂. It is a local energy source. Using it helps to maintain the forests and create jobs (a wood-fired power station creates four times more jobs than a gas-fired power station). However, burning wood emits dust, NO_x and fine particles. Thus, suitable filtration technologies must be used.

This section discusses how to use wood for thermal power, by burning it in a purpose-designed boiler ("wood-energy"). Other plant materials (straw, cane, corn cobs) can also be used to produce thermal energy. The wood used may be prepared in a traditional fashion, such as logs (rounds, wedges or charbonnettes), or in more modern forms which are better suited for automatic boilers, such as chips or granules, also known as pellets (Figure 1.53).



Logs

Wood chips

Pellets

Figure 1.53. *The different types of wooden fuels*

1.3.5.2.2. *Composition of wood*

Wood is made up of organic matter, minerals and humidity. The average gravimetric composition of organic matter of anhydrous (i.e. completely dry) wood is as follows:

- carbon (C) 50%;
- hydrogen (H₂) 6%;
- oxygen (O₂) 43%;
- nitrogen (N₂) 1%.

It produces relatively little ash – between 0.3 and 1.5% of the dry mass, depending on the species of tree. However, bark produces a higher level of ash, between 3 and 10%.

The moisture content of wood varies between 10 and 70% (on wet basis). Moisture can be quoted “on wet basis”, H_B , meaning relative to the total mass of wet wood (m_{wet}), or “on dry basis”, H_S , meaning relative to the mass of dry material (m_{dry}). Unlike other players in the field, those in the wood energy sector generally speak in terms of moisture on wet basis.

Moisture on wet basis (%):

$$H_B = 100 \frac{m_{wet} - m_{dry}}{m_{wet}}$$

Moisture on dry basis (%):

$$H_S = 100 \frac{m_{wet} - m_{dry}}{m_{dry}}$$

Relation between H_B and H_S :

$$H_B = \frac{H_S}{100 + H_S} \text{ or } H_S = \frac{H_B}{100 - H_B}$$

30% moisture on wet basis corresponds to 43% moisture on dry basis.

The moisture content of wood is measured by a standardized process of kiln-drying between 101 and 104°C for 24 h (French norm NF M 03-002 (AFNOR 1995)). By comparing the weight of the dry wood to that of the fresh, wet wood, we can calculate the moisture that was in it. A non-standardized but quick and

cheap method is to use a microwave to dry the wood in a few minutes (this produces an error of roughly 5% in comparison to the kiln-drying method).

The moisture of wood has an impact on its heating value, its density, the combustion techniques to be used and the biochemical changes it undergoes during storage. The LHV of wet wood LHV_{wet} in (kWh/kg) varies as a function of the species of tree, but above all, as a function of the moisture on wet basis H_B in (%):

$$PCI_{wet} = \left(1 - \frac{H_B}{100}\right) PCI_{dry} - 0.00166 \frac{H_B}{100} \tag{1.18}$$

For dry wood, we can take the overall LHV as being near to 5 kWh/kg.

The LHV of wood chips with 30% moisture on wet basis is 3.5 kWh/kg, and for 50% moisture on wet basis, it is 2.5 kWh/kg. The energy loss is 28%. The LHV of pellets with 10% moisture on wet basis is 4.5 kWh/kg.

Fuel	Useful energy (kWh)	Lower Heating Value, LHV	Production efficiency	Density (kg/m ³)	Mass (tons)	Volume (m ³)
Logs, H _b = 30%	10,000	3.5 kWh/kg	0.85	250	2.86	13.4
Wood pellets, H _b = 10%	10,000	4.5 kWh/kg	0.85	800	2.22	3.3
H natural gas	10,000	10.4 kWh/Nm ³	0.92	0.8	0.96	1,045.2

Table 1.4. Comparison of wood and gas fuels

To produce 10,000 kWh of useful energy, we need to burn 2.86 tons of wood chips at 30% gross moisture (i.e. 13.4 m³) or 2.22 tons of pellets at 10% gross moisture (i.e. 3.3 m³) or 1,045 Nm³ of natural gas. Wood chips, whose density is around 250 kg/m³, require much more storage space than the pellets, which are around 800 kg/m³ (see Table 1.4).

The moisture contained in the wood is an important parameter in terms of the energy efficiency, but it is also important for the generator. If we burn wood that is too dry in a chamber designed for moist wood, the generator will overheat, leading to the formation of clinker above 900°C and the risk of damage.

Drying is the process whereby water is ejected from the wood into the ambient atmosphere. For freshly-cut wood with around 50% moisture, the process begins

with the desorption of the free water, until it reaches around 25%. This first phase takes place fairly rapidly, and does not cause shrinkage. The following phase, to reduce the moisture below 25%, is much slower, because it relates to bound water.

So-called air-dry wood is wood which has reached hygroscopic equilibrium with the ambient environment (under shelter). With France's climate, this corresponds to around 20% gross moisture ("commercially dry" wood). For reasons of cost, the wood used for heating is typically dried naturally in the open air. The time taken for this natural air drying depends on the size of the pieces of wood. For split wood or logs, the drying process will take between 1 and 2 years. It is preferable to dry wood that is to be used as chips before it is shredded (to below 30% moisture) to prevent rotting during storage. After it is shredded, the drying process continues and the moisture drops to around 20% gross (under shelter).

1.3.5.2.3. Wood combustion

The exothermic reaction of combustion of wood $\text{C}_4\text{H}_6\text{O}_3 + 4\text{O}_2 \rightarrow 4\text{CO}_2 + 3\text{H}_2\text{O}$, on average, gives the following neutral combustion characteristics (for air-dried wood, 30% moisture):

- the volume of dry air necessary for neutral combustion, known as the combustive power, is $3.5 \text{ Nm}^3/\text{kg}$ raw wood;
- the volume of dry smoke (CO_2 , H_2O , N_2), known as the dry smoking capacity, is $3.5 \text{ Nm}^3/\text{kg}$ raw wood;
- the volume of moist smoke, called the moist smoking capacity, is $4 \text{ Nm}^3/\text{kg}$ raw wood;
- the volumetric rate of CO_2 emissions in the dry neutral smoke is 20.5%.

Wood is a solid fuel, so it is difficult to ensure perfect contact with the gaseous comburent. This generally means that a great deal of excess air – often over 60% – is used. For wood with 30% gross moisture in real-world combustion with a 70% air excess:

- the real dry air volume is $6 \text{ Nm}^3/\text{kg}$ raw wood;
- the dry exhaust volume is $6 \text{ Nm}^3/\text{kg}$ raw wood;
- the moist exhaust volume is $6.5 \text{ Nm}^3/\text{kg}$ raw wood;
- the volumetric level of CO_2 in the dry neutral exhaust is 12%;
- the volumetric composition of the moist exhaust is as follows:

$$\% \text{CO}_2 = 11\%, \text{H}_2\text{O} = 8.4\%, \text{O}_2 = 8\%, \text{N}_2 = 72.6\%.$$

The excess air must be very carefully controlled. It needs to be high in order to ensure the wood burns well and limit the pollutant emissions of unburned gaseous substances. However, too much excess air increases the exhaust volume and sensible heat losses and decreases the efficiency of combustion.

Therefore, we must find a compromise. The excess air is controlled by a probe that measures the level of oxygen in the exhaust gases. The goal is to keep it at a fixed level (e.g. $O_2 = 8\%$ for wood chips with 30% gross moisture, with 70% excess air), adjusting the air/wood mixture by varying the speed of the secondary air fan.

When heated, wood releases a large amount of gases (nearly 80% of its mass), much of which is combustible. Thus, the gaseous part and the residual solid part must be burned simultaneously. Therefore, wood burners are a specialized technology.

The wood used may contain a large quantity of moisture. This moisture is released by the drying process. For very wet products (freshly cut wood, wet bark), this preliminary phase may affect which burner technology is used. In addition, as the level of moisture may vary from one wood delivery to the next, the generator must be able to adapt easily to variable fuel properties.

Though ash production remains low (generally less than 1% of the gross mass), it is necessary to plan for its removal from the chamber and prevent the risks of the minerals in the ash from fusing (between 500 and 800°C) to form clinker, which could clog up the moving parts of the mechanisms.

There are several phases to the combustion of wood (Figure 1.54).

- *Drying*: this phase takes place between 100 and 200°C. It is endothermic and corresponds to the release of the moisture contained in the wood.

- *Pyrolysis*: between 200 and 500°C, the lignin and cellulose undergo pyrolysis, meaning that the gaseous products (flammable gaseous hydrocarbons and inert gases CO_2 , CO , H_2O) are released from the wood when heated.

- *Oxidation*: between 500 and 1,300°C, the combustible elements may then catch fire, if there is oxygen present, in the form of a diffusion flame (i.e. where the fuel and the oxygen combine at the point of ignition, rather than being premixed). The combustion temperature is between 1,100 and 1,300°C. The solid residue burns slowly and without a flame in the form of embers. The combustion temperature is near to 800°C. Energy is released as heat during the process of oxidation.

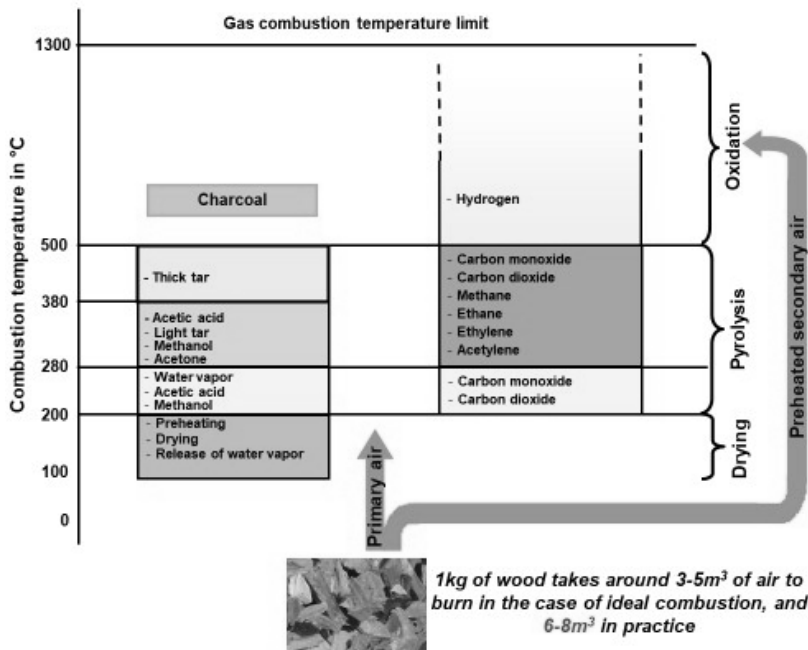


Figure 1.54. Combustion of wood

A wood-fired boiler, and especially the part comprising the burner, must be very carefully designed to have a dual air intake, to burn first the gaseous products resulting from the pyrolysis, and second the solid residue, to keep the chamber hot, so that at the end of the cycle, the solid residues have been completely burned.

1.3.5.2.4. Wood-fired boilers

Over 8 million households in France use logs, wood chips or pellets for their primary or secondary source of heating.

Log burners

Wood burning in its traditional form is found almost exclusively in the heating of single-family houses, with a living area of around 100 m², using 20–30 steres of wood per year. The “stere” is a traditional unit of volume, corresponding to 1 apparent cubic meter of stacked firewood. Depending on how the logs are arranged, the quantity of wood that a stere represents may actually vary significantly.

Manufacturers offer a number of stand-alone log-burning heating appliances. They are stand-alone in that they are not connected to a radiator network to deliver heat to different rooms in the home: open fireplaces (efficiency H 10–30%), closed fireplaces and inserts (H 30–50%), conventional stoves (H 50%), modern stoves (H 80%) and cooking stoves (H 65%). Energy harvesting systems exploiting the combustion gases from fireplaces and woodburning stoves can diffuse the hot air into rooms, which are not covered by the radiating effect, and thereby improve the system's overall efficiency.

Generators such as the hydraulic stove, the rocket mass heater (70–85%) and the log-burning boiler (H 90%) produce hot water for a heating network and for DHW.

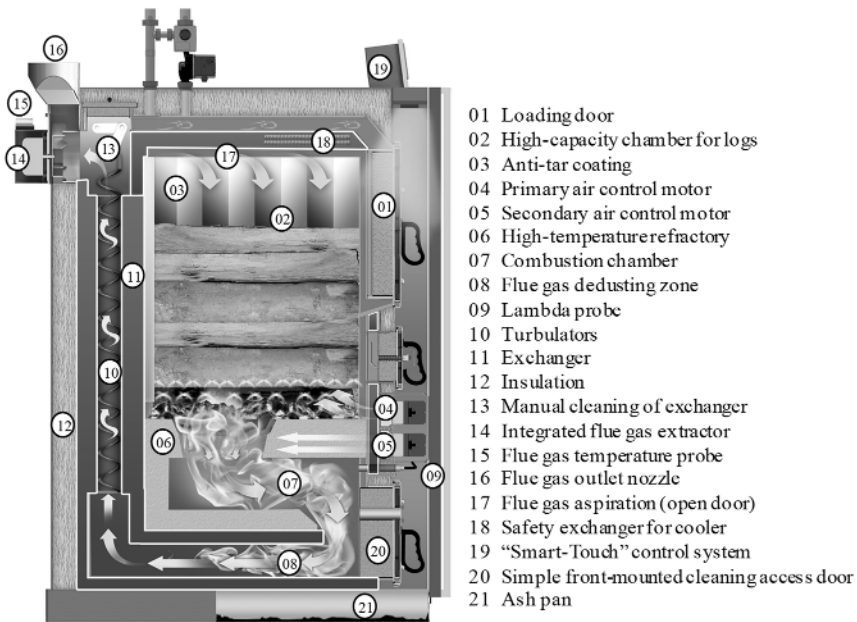


Figure 1.55. Example of a log-burning downdraft boiler (source: Hargassner)

In a log-burning boiler (see Figure 1.55), the wood burns on a grate that allows the comburent – air – to enter and to be evenly distributed throughout the wood payload. The grate must be narrow enough to act as a hopper, at the bottom of which

there is a buildup of embers, which remain incandescent for a long time. The grate then allows the ash to fall through. The comburent air is brought in at two different levels to allow for the two phases of combustion: primary air for the combustion of the solid part and secondary air for the combustion of the gaseous part. The comburent air may be brought in by natural draw or by a fan. Depending on the direction of the air flow, manufacturers offer down-draught, up-draught or horizontal boilers. The high-temperature exhaust passes over an exchanger to heat the water and provide both heating and DHW. The power may range between 30 and 100% of the nominal power. The normalized efficiency advertised by manufacturers varies between 90 and 94%.

Pellet-burning boilers

For certain uses, such as a large private home, industrial premises or a small collective heating installation, manufacturers offer automatic pellet-burning boilers, whose efficiency is between 94 and 96%. Some manufacturers are developing condensing technologies using graphite condensers, delivering LHV efficiencies of over 104%.

The wood supply runs constantly and automatically (using a worm screw and/or compressed air). The pellets, which are made by compression at 100 bars and/or extrusion without any additions, are delivered by truck or in sacks and require less storage volume than wood chips. However, pellets are more expensive than wood chips.

The pellets are injected into the bottom of the combustion chamber, either by a worm screw or by gravity (Figure 1.56). A rotary valve prevents uncontrolled fire by isolating the combustion chamber from the storage silo. Combustion is triggered automatically by an electric igniter. The primary air (beneath the grate) and the secondary air (above the flames) are injected by a fan. At the top of the stove, the exchanger heats the water by recovering the sensible energy from the combustion gases, and the latent heat too in the case of a condensing boiler. Mobile turbulators in the exhaust pipes increase heat exchange and also provide automatic mechanical cleaning of the heat exchanger. The combustion gases are evacuated by a flue pipe using an extractor fan. The boiler walls are insulated to minimize thermal losses. The ash is collected in an ash pan at the bottom, which needs to be emptied regularly (around once a week). The boiler power can be modulated between 20 and 100% by varying the flow rate of pellets. An oxygen probe in the exhaust enables the combustive/fuel mix to be adjusted to ensure good combustion.

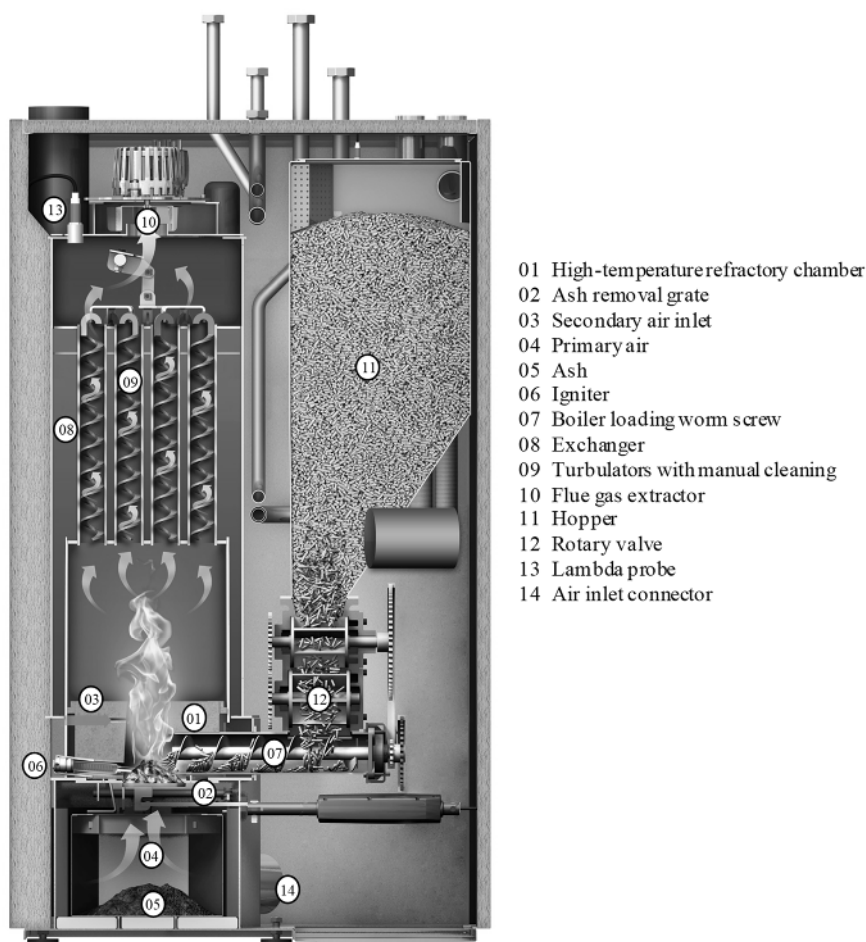


Figure 1.56. Example of a pellet-burning boiler (source: Hargassner)

Wood chip-burning boilers

Wood chip-fired technology is found almost exclusively in the heating of collective or industrial buildings. Most installations are boilers that are automatically supplied with fuel, with low power (under 500 kW) or medium power (between 500 and 1,500 kW), often in rural areas, connected to a small heating network that serves several buildings. A smaller number of installations are automatically supplied boilers for district heating, with high power (over 1,500 kW), connected to extensive

heating networks. In this case, the wood is used in combination with additional heat production from fossil fuels. This dual-fuel arrangement is not always implemented with low- and medium-powered boilers. Chips require more storage volume than pellets, but chips are cheaper.

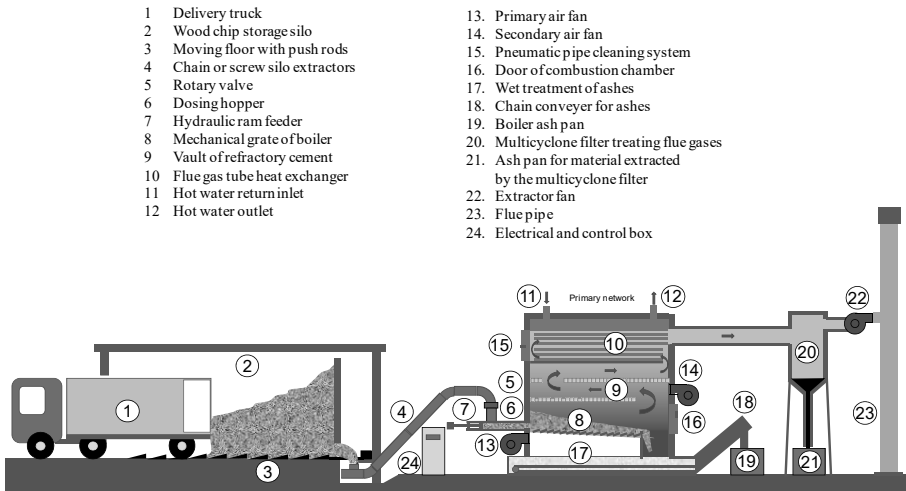


Figure 1.57. *Example of an installation with a medium- and high-powered wood chip boiler*

These boilers are made of steel, with a vault of refractory materials, protecting a proportion of the surfaces exposed to flame. The fuel is moved onto an inclined grate by the ram feeder (Figure 1.57). This action may be supplemented by an oscillating grate. The different phases of combustion take place as the fuel works its way down, with the ashes falling into the last stage of the process.

The flue gas-to-water heat exchanger generally uses multiple-path exhaust pipes for low- and medium-powered boilers. Moving mechanical devices are sometimes introduced into the pipes both to turbulate the flue gases and clean up the pipes, removing ash deposits (shaking in the case of vertical pipes, rotation for horizontal pipes). Pneumatic devices may also be used. In very high-powered systems, we tend to find water tube boilers.

An extractor evacuates the exhaust gases through a flue pipe. Before ejection, these gases are treated with a cyclone filter in low- and medium-power boilers, and by an electrostatic or sleeve filtration system in high-power boilers.

Ash can be removed automatically in dry conditions (the ash is extracted from the boiler by a worm screw into a moving container) or wet (the ash falls into a water tank, from which it is evacuated by a scraper conveyor or worm screw, in the form of sludge, into a storage bin). The ash recovered in this way can be reused as agricultural fertilizer once it has been screened for any metal elements.

We must also have a storage facility (a hangar, silo, container, waterproof pit), which delivery trucks can access, housing enough fuel for 4–6 days' uninterrupted operation. The fuel is automatically carried from storage into the boiler by scraper devices, bladed systems, a conveyor belt, worm screw or compressed air.

The cost of a wood-fired heating system (excluding financial aid specifically for this purpose) is three to five times more than that of a gas-fired system offering the same level of power. This overspend may be compensated by the fact that the fuel itself costs two to three times less, though maintenance costs are also 1.5–3 times higher for a wood-fired system. The price of firewood is heavily influenced by the cost of transporting it, which favors the use of local resources, and proximity between the heating system and waste-generating activities.

Wood-chip boilers are most efficient when running at their maximum rate. That efficiency drops when they are at partial power because of the boilers' thermal inertia. In modern boilers, the power is contingent upon the level of residual oxygen in the moist flue gases (exiting the boiler), measured with a so-called lambda/zirconia oxygen sensor. In this way, it is possible to optimize the combustion of wood from various origins with widely differing moisture levels. For dual-fuel wood/gas heating systems, the average efficiency also depends on the installation's sizing, hydraulic design and control system.

1.3.5.2.5. Sizing, design and control system

When designing a wood heating system, there are various potential solutions:

- *“Wood-only” heating system*: the site's heating demands are served entirely by one or more wood-fired boilers – for example, when no other form of energy is available on the site. Such systems are low-powered.

- *A dual-fuel heating system*: the site's heating demands are covered primarily by a wood-fired boiler but supplemented by a gas-fired boiler or an HP, for example. This solution offers the following advantages:

- limiting the excess costs inherent in a “wood-only” system, while ensuring the resource is put to good use. Indeed, over 70% of demand can be satisfied by a wood boiler, representing only 40–60% of the total installed power. Limiting costs is

particularly important when renovating existing installations in which part of the gas-fired system needs to be reconditioned and conserved;

- continuity of energy production if there is a risk of disruption to the wood collection and distribution supply chain, and in the event of extreme weather conditions;

- a diversity of energy sources, which will allow us to adapt to long-term shifts in fuel costs.

The sizing of installed power is important in ensuring a wood-fired heating system works optimally. If the design power of the wood-fired generator is poorly chosen at the design phase, it can lead to:

- reduced efficiency of the production system;
- overconsumption of wood and any auxiliary resource;
- higher running costs;
- rapid degradation of the control mechanisms;
- heavy pollutant emissions.

With a “wood-only” heating system, the installed power must cover all the site’s needs, calculated in reference to that site’s design outdoor temperature conditions.

With a dual-fuel heating system, the sizing for the wood-fired system and the auxiliary supply must be guided by an analysis of the energy demand and the heat load duration curve. This curve represents the number of days on which power is required. It is based on the frequency of each outdoor temperature, heat losses and solar and internal heat gains.

Consider an example. A building’s thermal losses are 2,100 kW for a design outdoor temperature of -4°C and an indoor temperature of 20°C . For the building considered in this study, Figure 1.58 shows the number of days as a function of the outdoor temperature, and Figure 1.59 shows the corresponding heat load duration curve. The solar and internal heat contributions cover around 10% of the losses. If the power of the wood-burning boiler is chosen as 1,100 kW, which is 50% of the maximum power, the wood-burning system will cover over 80% of the demand – that is, the proportion of energy derived from wood in relation to the site’s total demands (red area). The backup boiler, with 1,100 kW of power, covers less than 20% of the energy demands when it is cold (blue area). The solar and internal heat contributions cover 10% of total energy use (green area).

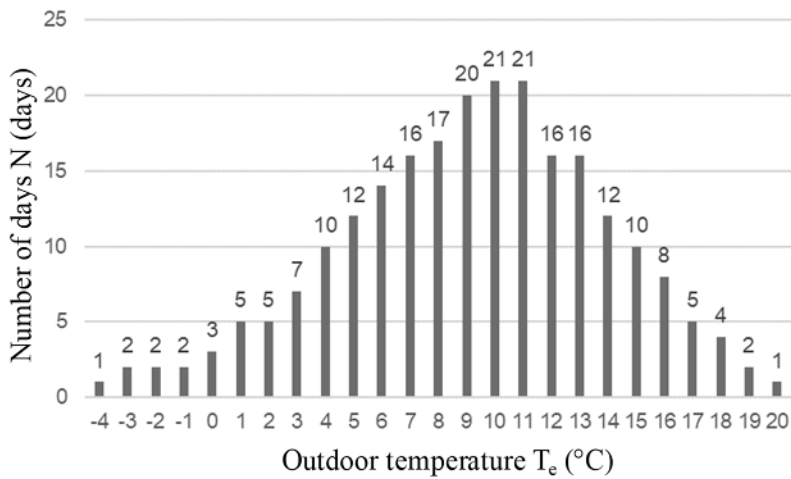


Figure 1.58. Number of days per outdoor temperature for a site

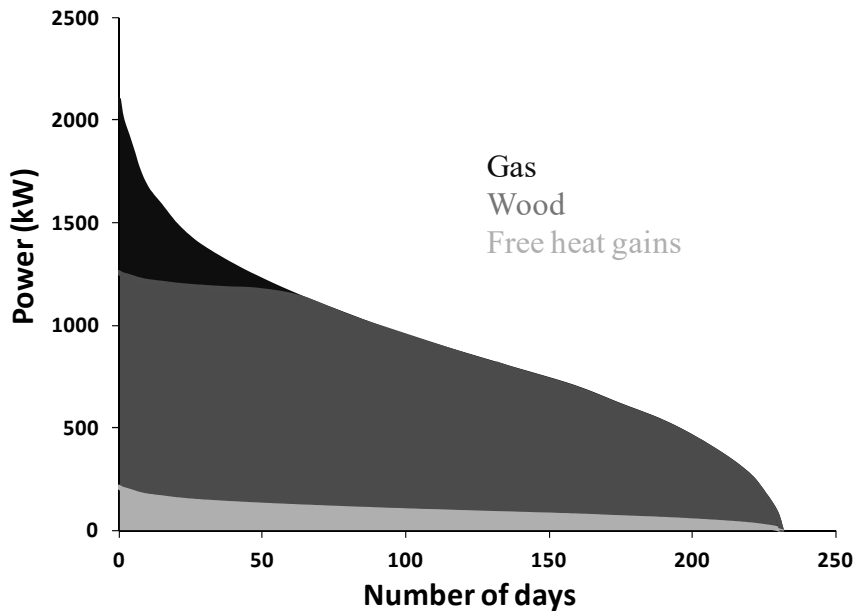


Figure 1.59. Heat load duration curve of power for heating

In the case of dual-energy systems such as wood/gas or wood/HP combinations, the heat load duration curve shows that it is crucial to avoid overdimensioning the installed wood heating capacity so as to limit investment costs. A wood boiler providing approximately 50% of the total installed capacity can achieve over 80% wood usage.

To maximize the proportion of energy derived from wood and achieve optimal efficiency, special attention must be paid to the hydraulic system's layout and control system. There is a wide range of technical solutions available for this purpose.

In non-condensing wood-fired boilers, it is crucial to maintain a return water temperature of at least 65°C to avoid corrosion caused by acidic condensates. A fixed or minimum water flow through the boiler's heat exchanger must be maintained. This requires a load pump, plus a three-way valve with a recycling bypass to ensure the return temperature does not fall too low, thereby protecting the boiler (see Figure 1.60).

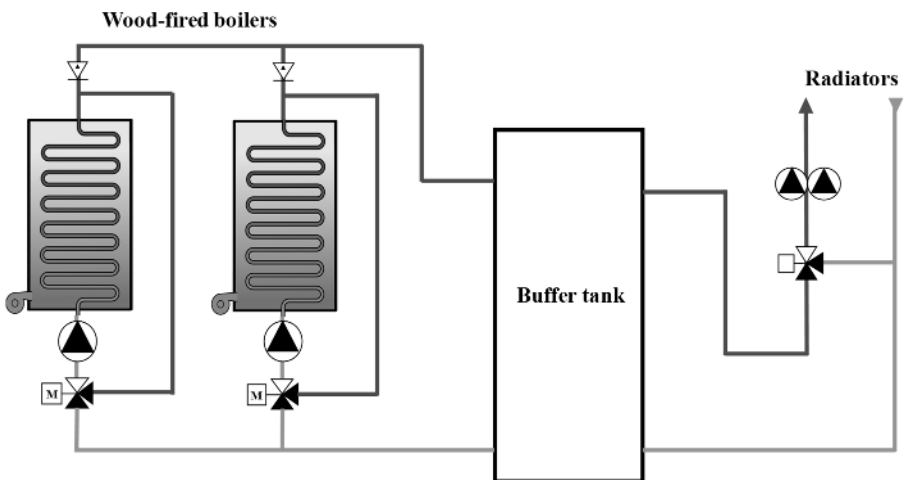


Figure 1.60. *Simplified schematic view of a wood-fired boiler system with a four-pipe buffer tank*

To avoid operating the wood boiler at low power, a buffer tank should be included for heat storage. The reservoir should be sized to handle at least 1 h of full-power operation.

The tank's status is measured by several temperature sensors at different heights on the tank. If the heating demand from the system decreases, the boiler can continue to run at its minimum power by charging the tank. It is only put into ember retention mode (or shut down in the case of boilers with automatic ignition) once the tank is fully charged. The boiler will only power up or resume operation when the tank is nearly depleted.

In such setups, a four-pipe buffer tank is installed as an HS. However, this can cause problems. If the tank is too wide, it may lead to differentiated hydraulic flow paths. A height-to-diameter ratio of around 3:1 is recommended.

This wood/gas configuration uses a two-pipe “open buffer” tank (see Figure 1.61). It allows for hydraulic decoupling of the production and distribution circuits. This setup is widely used in horticultural and market garden greenhouse systems. It can store the surplus thermal energy from wood burning, but the filling and emptying of the tank are not regulated. Consequently, when the gas boiler activates, it may inadvertently reheat the buffer tank, which is a waste of wood.

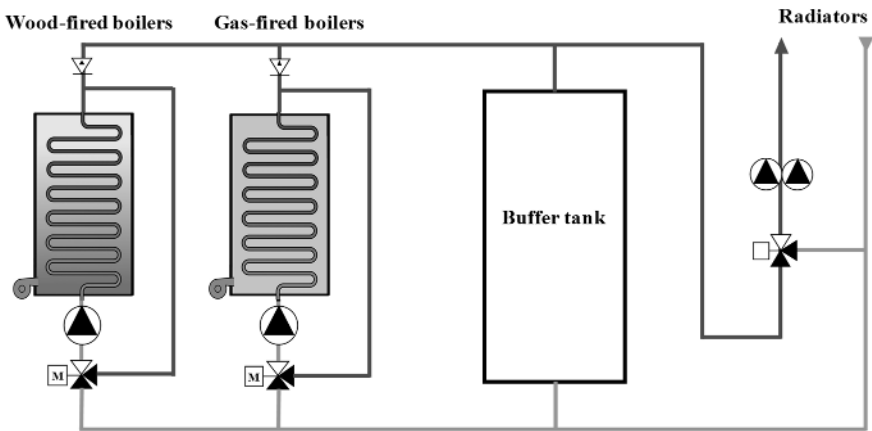


Figure 1.61. *Simplified schematic view of a wood/gas boiler room with a two-pipe tank with open buffer configuration*

In this arrangement (see Figure 1.62), priority is given to the wood boiler so it can deliver its full power, but this power must not be limited by low return temperatures. This occurs, for example, in intermittently occupied buildings during morning start-up. A solution must therefore be found to supply high-temperature return water.

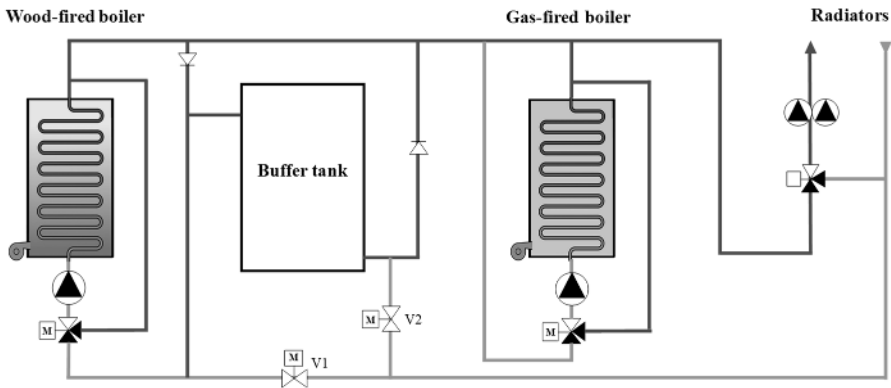


Figure 1.62. *Simplified schematic view of a dual-energy wood/gas boiler system*

In the morning, the wood boiler must be restarted before the network pumps begin operating. The wood boiler heats up again, and a return temperature of 70°C must be reached before the three-way valve opens and releases hot water to the networks. The wood boiler heats the buffer tanks to $70/75^{\circ}\text{C}$, and the three-way valve gradually opens until fully open. When the network restarts, the two-way valve V1 remains closed. The cold water returning from the networks passes through the buffer tanks. The boiler, supplied with water at 70°C , delivers full power. To prevent the gas boiler from starting, the network start-up schedule is staggered. If the flow rate of the wood boiler's primary pump exceeds that of the secondary networks, the buffer tanks continue to be heated up. All networks are operational, and demand is at its highest since it is the morning restart period. If the wood boiler's primary pump flow rate is lower than the total flow rate of the secondary circuits, the wood boiler continues to deliver its full output, and the buffer tanks provide additional support. The gas boiler is still not running. When the average temperature of the buffer tanks is equal to the return temperature of the networks, the two-way valve V1 opens and the two-way valve V2 closes. If the return temperature to the boiler drops below 70°C , the three-way valve closes again, and the wood boiler no longer delivers its full output. When the combined output of the wood system and buffer tanks is no longer sufficient, the gas boiler takes over. With a well-calibrated system and correctly sized buffer tanks, the gas boiler should only activate when external temperatures are very low or when the wood boiler is undergoing maintenance. This wood/gas coupling should ensure a high percentage of wood usage throughout the year. A dedicated control scheme and functional analysis must be defined to precisely govern its operation.

1.3.5.3. Combined heat and power units

The first CHP units were developed in the sugar industry in the 1920s, followed by widespread adoption in various other industries during the first oil crisis of the 1970s. The 2000s saw such systems being used to meet the energy needs of large buildings, and they began being used in residential buildings in 2009.

CHP refers to the simultaneous production of two forms of energy from a single primary energy source. In this section, we focus on the simultaneous generation of heat and electricity from natural gas.

Manufacturers provide several types of systems:

- external combustion Stirling engine;
- gas-fired internal combustion engine (ICE);
- gas turbine;
- fuel cell, etc.

CHP systems are classified on the basis of their power output. Micro-CHP units have a maximum capacity of 50 kW_{el} (as defined by European Directive 2004/8/EC). In France, the threshold is based on EDF subscription bands and is set at 36 kVA (the connection limit for the “*tarif bleu*” residential rate). The most commonly used systems are the Stirling engine and the gas internal combustion engine. These systems are sized to heat a building while simultaneously producing electricity for self-consumption. Any surplus electricity can be fed back into the grid (Table 1.5).

Pelec. in kVA	< 36 kVA	36-250 kVA	250-1,000 kVA	1,000-12,000 kVA	>12,000 kVA
Classification	Micro	Mini	Small	Medium	Large
Electrical power threshold and technologies	1 – 50 kW Stirling engine/ICE/Micro and Mini turbines				
		50 kW – 1 MW ICE/Mini turbines			
			1 – 6 MW ICE and turbines		
	Turbines			5 – 125 MW	

Table 1.5. Classification of different CHP systems

The advantage of a CHP system lies in the efficiency gain compared to the equivalent heat and electricity when produced separately (see Figure 1.63).

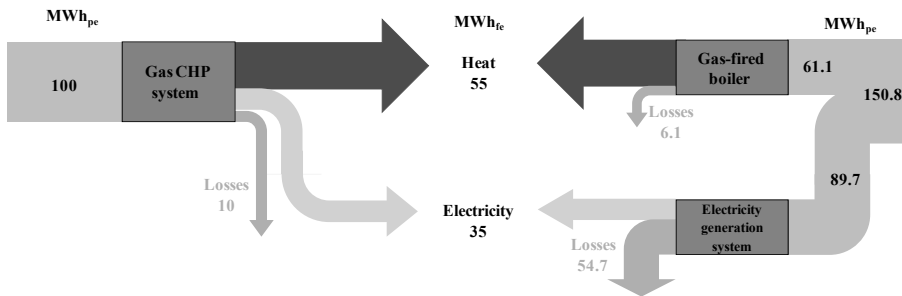


Figure 1.63. Energy comparison between a gas CHP system and a separate production system (CEGIBAT)

A gas micro-CHP generator with 55% thermal efficiency and 35% electrical efficiency consumes 100 MWh_{pe} (primary energy) to produce 55 MWh_{fe} (final energy) of heat and 35 MWh_{fe} of electricity. Its overall efficiency is therefore 90%.

To meet the same energy demand using a separate system, consisting of a gas condensing boiler with 90% average efficiency and the French national electricity generation system with 39% average efficiency, the primary energy consumption would be 150.8 MWh_{pe}.

The overall efficiency of micro-CHP is 90%, whereas the overall efficiency of the separate system is only 60%. Thus, to meet the same final energy needs, CHP uses significantly less primary energy.

CHP also helps to smooth national electricity demand peaks by producing electricity locally at times when grid-supplied electricity is more carbon-intensive (e.g. winter peaks).

Although costly in terms of initial investment, this system offers advantages when a building has high, relatively stable and simultaneous demands for both heat and power.

The greater the building's demand for heating and DHW, the higher the electricity production. Depending on the building's thermal requirements, savings of 20–40% can be made on the electricity bills.

1.3.5.3.1. Micro-CHP boiler with Stirling engine

In individual homes, a boiler incorporating a gas micro-CHP unit with a Stirling engine can produce 1 kW of electricity and 5–8 kW of thermal energy in the form of DHW. The additional thermal energy is supplied by the gas boiler. This system, also known as a “thermogenic boiler” or “CHP boiler”, is not designed to make a home fully self-sufficient in electricity production but to operate in conjunction with the grid. It can supply up to 70% of the annual electricity needs.

1.3.5.3.2. Gas-powered micro-CHP unit with a thermal engine

The micro-CHP with a gas thermal engine produces hot water and electricity simultaneously (Figure 1.64). It delivers electrical power in the range of 6–36 kW_{el}. The ratio of electrical input to thermal energy output is approximately 2–3. For example, a 9 kW_{el} micro-CHP produces around 20 kW_{th}. These systems are designed for collective residential and small commercial buildings. The operating principle is that of a thermal engine that generates electricity. The engine’s thermal losses are recovered from the oil, engine block and exhaust gases. Electrical efficiencies are around 30%, compared to just 10% for a Stirling engine. However, thermal efficiency is lower: about 55% for a thermal engine compared to 90% for a Stirling engine.

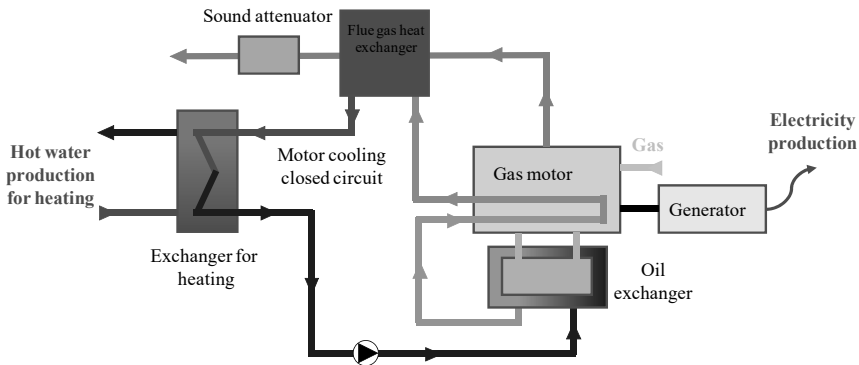


Figure 1.64. Principle of a gas-powered micro-CHP unit with a thermal engine

A gas-powered micro-CHP unit is always paired with one or more boilers, which follow the building’s thermal load. The system’s sizing must be determined according to both the thermal and electrical demand of the building, and the intended use for the electricity: full self-consumption, self-consumption without surplus feed-in, partial feed-in or total feed-in. To ensure optimal performance of a micro-CHP system, the following conditions must be met:

- a stable and sufficiently high-power output that satisfies the building's heating and power demands simultaneously;
- maximum possible operating time for good financial viability (over 3,000 h per year);
- operation at 100% load for the maximum time possible. Its power can be modulated from 100 to 50%, but its efficiency decreases from 95 to 90%;
- a limited number of start/stop cycles (no more than 1,000 per 3,000 h of operation). Frequent cycling reduces lifespan and increases maintenance costs. A buffer tank of around 40 L/kW_{th} is recommended as an initial guideline;
- the lowest possible water return temperature (between 30°C and 50°C) to ensure the gas condensing micro-CHP generator works properly. If the water input temperature is higher, thermal output will decrease by approximately 0.33% per additional degree Celsius. For standard gas micro-CHP generators, the return temperature should be above 55°C;
- a constant water flow rate through the micro-CHP generator, calculated on a nominal temperature difference of 20°C.

To meet these various constraints, its integration into the boiler room's hydraulic system and its settings must be configured appropriately. A micro-CHP generator usually delivers less power than the boilers. If the system is not controlled correctly, the higher capacity of the boilers may override the micro-CHP generator, reducing its runtime and, therefore, its electricity output and thus return on investment.

Consider the example of a collective housing block. The heating and DHW needs are met by a cascade of two low water-capacity condensing gas boilers (150 kW each) and a gas condensing micro-CHP unit with a thermal engine (9 kW_{el} and 20 kW_{th}) to supply part of the electricity. Optimal operation is achieved with operating conditions of 50/30°C and a constant flow rate of 0.85 m³/h. The constraints of the micro-CHP unit are operating conditions of 85/65°C, a return temperature between 30 and 65°C, and a limited number of starts (1,000 starts for 3,000 h of operation per year).

In order to meet the manufacturer's constraints on flow rate and temperature for both the condensing boilers and the condensing micro-CHP generator, pumps must be used on the primary circuit. The micro-CHP generator is connected to a buffer tank for optimal performance (see Figure 1.65). A hydraulic separation tank separates the primary and secondary circuits (which include two low-temperature radiator networks and a high-temperature storage DHW system).

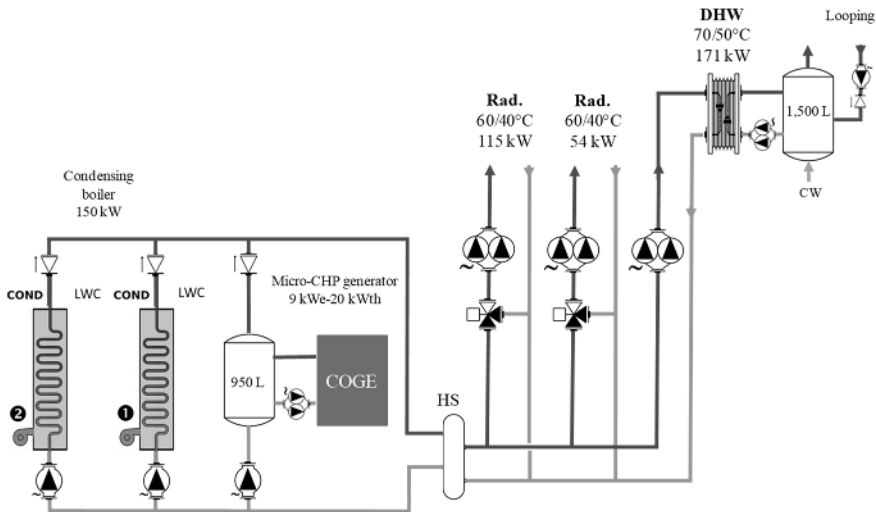


Figure 1.65. Micro-CHP generator integrated into a gas boiler system, where HS is the hydraulic separator

The design of such a system can be optimized. When the primary flow rate through the HS exceeds the secondary flow rate, a downward vertical flow develops within the separator, warming the return water and thereby limiting condensation in the generators. One solution is to regulate the cascade of primary pumps while maintaining the minimum required flow rates. When the DHW tank is no longer at its set temperature, its pump is activated and the primary circuit must produce high-temperature water (70°C). The water returns from the networks at high temperatures, and the generators no longer condense efficiently. A technical solution would be to supply the DHW heat exchanger and the radiator circuits in series.

1.3.5.4. Heat pumps

1.3.5.4.1. History and market for heat pumps

The HP is a thermodynamic system in which a fluid undergoes cyclical transformations, exchanging energy with its surroundings. Foundational work in this field was carried out in the 1830s by scientists including Sadi Carnot, James Prescott Joule, Lord Kelvin and Rudolf Clausius.

In the United States, Willis Haviland Carrier developed the first centrifugal refrigeration system in 1920. Air conditioning systems were gradually introduced

into commercial and industrial buildings, followed by residential homes, and then motor vehicles in the 1950s.

In Europe, the air conditioning and HP market began to grow in the 1970s. The arrival of Japanese manufacturers in the 1980s, offering high-performance equipment, accelerated the development of the European market. In France, HPs accounted for 9% of primary renewable energy production in 2017.

Since the implementation of the RT 2012 and RE 2020 building regulations, the HP market has continued to evolve. Approximately 700,000 units were installed in France in 2018. Air-to-air HPs are the most widely adopted technology, as they can be easily installed and are relatively inexpensive.

1.3.5.4.2. General principle of HPs

Our surrounding environment contains abundant renewable energy – in the air, water and ground. However, the temperature of these readily available sources is not sufficient to directly meet buildings' heating and DHW needs.

An HP is a thermodynamic machine designed to extract heat from one medium and deliver it to another medium at a higher temperature. This type of system now commonly provides the majority of heating for residential buildings and produces DHW by drawing thermal energy from outside air, ground, or, in some cases, from groundwater – that is, from sources colder than the building interior.

Consider an underfloor heating system using energy from the ground. At a depth of 2 m, the earth is approximately 12°C. The underfloor heating system requires water at around 35°C. The ground's temperature is therefore insufficient to supply the system directly.

In fact, thermal energy always flows from hot to cold – not vice versa. Clausius stated in the second law of thermodynamics, “Heat can never pass from a colder to a warmer body without some other change, connected therewith, occurring at the same time”. To transfer energy in the opposite direction, that is, from a cold source, such as the ground at 12°C, to a warm one, such as the underfloor heating system at 35°C, a dedicated system called an HP must be used to supply energy to the system.

The cold source at temperature T_c cools further by transferring a quantity of heat to a refrigerant, initially at low pressure (cold source). Then, the hot source, at temperature T_h , can be warmed by absorbing heat from the refrigerant (hot source) once the refrigerant has been pressurized by external energy input (see Figure 1.66).

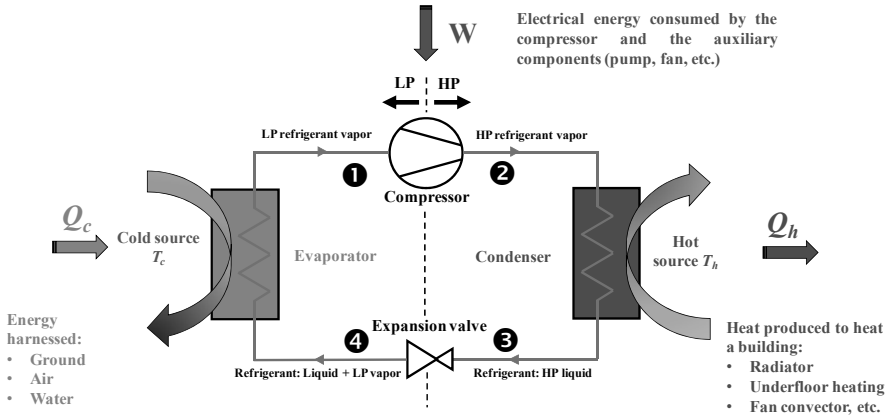


Figure 1.66. *Illustration of a heat pump with mechanical compression*

HPs are often reversible. The principle on which they operate will be discussed in section 1.3.5.4.8. They can alternately produce heat (e.g. in winter to heat a building) and cooling (e.g. in summer to cool it).

In general, a thermodynamic machine operating between two reservoirs (hot and cold) can serve:

- heating demands → this is known as an HP;
- cooling demands → this is known as a refrigeration unit or air conditioner;

– simultaneous heating and cooling demands → In French, this technology is known as a “thermofrigopompe” that refers to these refrigeration systems or HPs for simultaneous heating and cooling

This third type is used in buildings that have concurrent heating and cooling demands in different zones (such as hospitals, swimming pools, aquariums and commercial buildings). It is the most efficient solution, as it meets both energy needs using a single energy source.

Note that there is no fundamental difference between the working principle of a HP and a refrigeration unit. What differentiates them is the system’s intended purpose.

1.3.5.4.3. Different HP principles

In an HP, a refrigerant undergoes cyclic transformations involving energy exchanges with the external environment through four main components: the compressor, condenser, expansion valve and evaporator. In the condenser, the high-pressure (HP) refrigerant condenses and transfers heat to the hot source. In the evaporator, the opposite occurs: the low-pressure (LP) refrigerant vaporizes and cools the cold source. An expansion valve allows the refrigerant to pass from HP to LP. To raise the refrigerant from LP to HP, three main technologies are used:

- Electrically powered mechanical compressor.

- Ejector system: the mechanical compressor is replaced by an ejector (or ejecto-compressor). The installation includes a refrigeration circuit for cold production and another circuit to produce motive steam. This method allows much of the mechanical compression energy to be substituted with thermal energy (e.g. from solar or geothermal sources).

- Absorption system: the mechanical compressor is replaced by a boiler. A refrigerant-rich solution (e.g. water) is brought to boil by an external heat source (solar, wood, etc.), which vaporizes the fluid. The refrigerant separates from the absorbent (e.g. lithium bromide). This vapor travels to the condenser and transfers its heat to the hot source. The supercooled liquid expands on experiencing a sudden pressure drop across the expansion valve into the evaporator, where it then evaporates. The vapors travel from the evaporator into the absorber, where they are met by the absorbent, which reconstitutes the refrigerant-rich mixture through hygroscopic action. This solution is pumped back in so that the cycle can begin anew.

1.3.5.4.4. Principle of a mechanical compression HP

The most commonly used systems in the building sector are mechanical compression HPs.

The cycle followed by the refrigerant is shown in Figure 1.66:

- Between points ❶ and ❷, the vapor form of the refrigerant at low pressure is compressed. This results in a hot, HP gas at the compressor outlet.

- Between points ❷ and ❸, the gas flows through a heat exchanger (the condenser), and the refrigerant releases a quantity of heat, Q_h , to the hot source (air or water). At the condenser outlet, the refrigerant is now in liquid form but still at high pressure. It is important to note that the air or water passing over the condenser

must be colder than the gaseous refrigerant; otherwise, the refrigerant cannot release heat.

– Between points ③ and ④, the liquid refrigerant undergoes a sudden expansion in the expansion valve, transforming into a LP liquid–vapor mixture.

– Between points ④ and ①, the liquid–vapor mixture of refrigerant flows through a second heat exchanger (the evaporator), where it fully evaporates by absorbing a quantity of heat Q_c from the cold source. At the evaporator outlet, the refrigerant is now vaporous and still at low pressure. It should be noted that the air or water circulating through the evaporator must be warmer than the expanded refrigerant; otherwise, the refrigerant cannot absorb heat.

1.3.5.4.5. The Mollier diagram

The Mollier or enthalpy diagram is commonly used for studying refrigeration cycles (Figure 1.67). The axes are pressure p and specific enthalpy h . This diagram has several sets of curves pre-drawn: isobars ($p = \text{Const.}$) in (bar); isenthalps ($h = \text{Const.}$) in (kJ/kg); isotherms ($T = \text{Const.}$) in ($^{\circ}\text{C}$); isochores ($v = \text{Const.}$) in (m^3/kg); vapor quality lines ($x = \text{Const.}$); and isentropes ($s = \text{Const.}$) in (kJ/kg), and also the saturation curve, delimiting the area where the two phases of the refrigerant (liquid and gas) coexist. By treating the cycle as a series of quasi-static transformations, the Mollier diagram allows the cycle followed by the refrigerant to be represented graphically according to measured parameters, from which the other variables of state can be deduced at any point in the cycle.

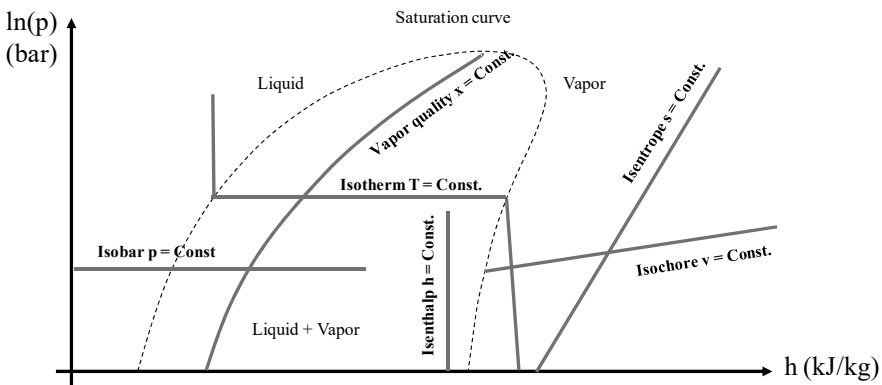


Figure 1.67. Mollier diagram

1.3.5.4.6. Coefficient of performance

The performance of a HP is not assessed by its efficiency but by its coefficient of performance (COP), which is defined as the ratio of “useful energy”, that is, the energy produced at the condenser, to “payable energy” to operate the thermodynamic device – that is, the electricity consumed in the case of a mechanical compression HP.

For example, a COP of 3 means that for every 1 kWh of electricity used, the HP produces 3 kWh of heat. Therefore, 2 kWh are extracted from the air, water, or ground by the evaporator (Figure 1.68). This energy is renewable and free. Note that in France, HPs are classified as renewable energy systems, though they consume electricity from nuclear sources.

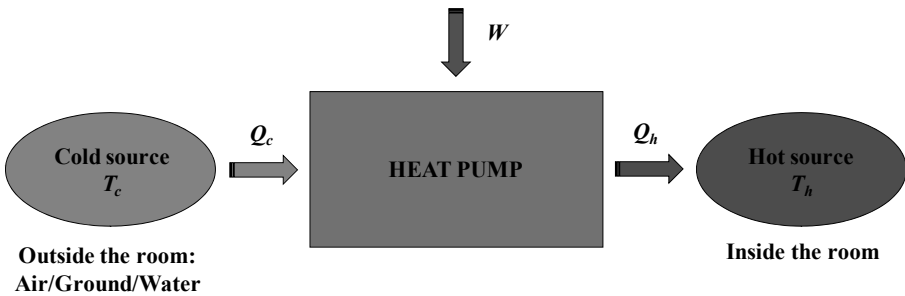


Figure 1.68. General operating principle of a heat pump

The coefficient of performance of the Carnot cycle COP_{cycle} , also called the COP_{ideal} of the cycle or the COP_{max} of the thermodynamic machine, corresponds to a COP with no loss, operating between a cold source at temperature T_c and a hot source at temperature T_h in kelvin.

The idealized Carnot cycle with phase change is a reversible cycle with maximum efficiency (Figure 1.69). It includes:

- two isobaric transformations during which phase changes occur, at constant temperature, in the condenser $\textcircled{2} \rightarrow \textcircled{3}$ and the evaporator $\textcircled{4} \rightarrow \textcircled{1}$;
- two reversible isentropic, or adiabatic, transformations, corresponding to the compression $\textcircled{1} \rightarrow \textcircled{2}$ and expansion $\textcircled{3} \rightarrow \textcircled{4}$ phases.

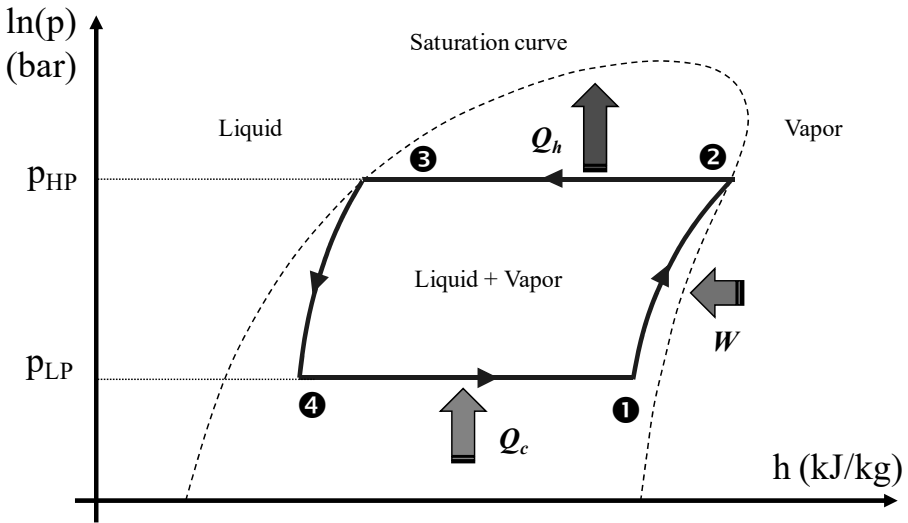


Figure 1.69. Theoretical Carnot cycle for a single-stage mechanical compression heat pump

The first law of thermodynamics states that the change in internal energy ΔU of a closed system is equal to the sum of energy exchanges with its environment. These include the compression work W and the heat exchanges with the hot source Q_h and the cold source Q_c :

$$\Delta U = W + Q_h + Q_c \quad [1.19]$$

Conventions:

- $Q_h < 0$, as the refrigerant loses energy to the hot source;
- $Q_c > 0$, as the refrigerant gains energy from the cold source;
- $W > 0$, as compression work is applied to the refrigerant.

The second law of thermodynamics states that the change in entropy ΔS of a closed system is the sum of internal entropy exchanges ΔS_{int} and the entropy exchange with the external environment ΔS_{ext} :

$$\Delta S = \Delta S_{int} + \Delta S_{ext} \quad [1.20]$$

During a reversible transformation, the variation in internal entropy is zero:

$$\Delta S_{int} = 0$$

The variation in entropy linked to thermal exchanges with the environment at the cold source ΔS_c and the hot source ΔS_h is:

$$\Delta S_{ext} = \Delta S_c + \Delta S_h = \frac{Q_c}{T_c} + \frac{Q_h}{T_h}$$

Thus:

$$\Delta S = \Delta S_{int} + \Delta S_{ext} = \frac{Q_c}{T_c} + \frac{Q_h}{T_h} \quad [1.21]$$

During a cycle, the variations in the system's internal energy and entropy are zero. In light of the first and second laws of thermodynamics, we can write:

$$Q_h + Q_c = -W \text{ and } \frac{Q_c}{Q_h} = -\frac{T_c}{T_h}$$

For an HP:

$$COP_{ideal (hot)} = -\frac{Q_h}{W} = \frac{T_h}{T_h - T_c} \quad [1.22]$$

For a refrigeration system:

$$COP_{ideal (cold)} = \frac{Q_c}{W} = \frac{T_c}{T_h - T_c} \quad [1.23]$$

For a thermo-frigo-pump:

$$COP_{ideal (cold + hot)} = \frac{T_h + T_c}{T_h - T_c} \quad [1.24]$$

The COP for cooling is also known as the *energy efficiency ratio* (EER).

Consider an external air-to-water heat pump that heats a building by means of underfloor heating, where the system delivers heat to the underfloor circuit at an average temperature of $T_h = 35^\circ\text{C}$ (the hot source), while extracting energy from the outside air at $T_c = 0^\circ\text{C}$ (the cold source):

$$COP_{ideal (hot)} = \frac{35 + 273}{(35 + 273) - (0 + 273)} = 8.8$$

If, instead, the heating is delivered by radiators with an average temperature of $T_h = 55^\circ\text{C}$, using outside air at $T_c = 0^\circ\text{C}$ (cold source):

$$COP_{ideal(hot)} = \frac{55 + 273}{(55 + 273) - (0 + 273)} = 6$$

For a ground-water geothermal HP with a cold source temperature of $T_c = 10^\circ\text{C}$ used for low-temperature heating at $T_h = 35^\circ\text{C}$:

$$COP_{ideal(hot)} = \frac{35 + 273}{(35 + 273) - (10 + 273)} = 12.3$$

In practice, because of the thermodynamic irreversibilities and energy losses in the compressor, the real COP is around half of the ideal COP.

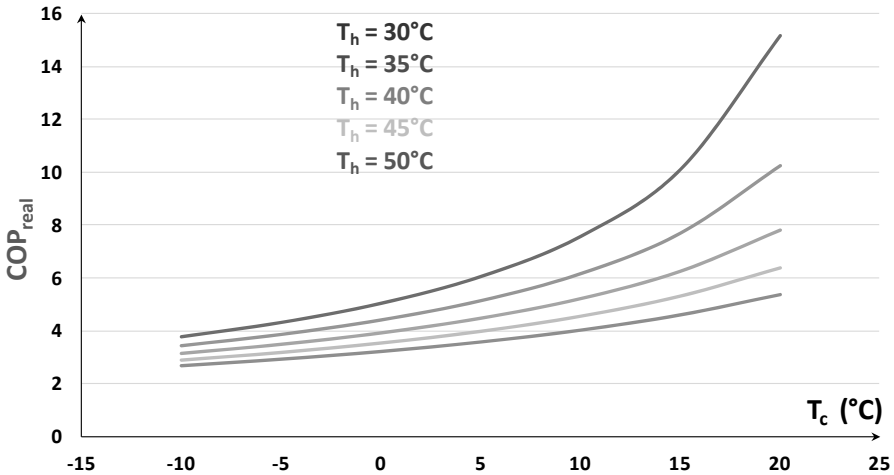


Figure 1.70. Evolution of the ($COP_{real} = 0.5 COP_{ideal}$) as a function of T_h and T_c

We can see that, the smaller the temperature difference between the hot and cold sources, the higher the HP's COP. A 1 K decrease in the temperature of the cold source will decrease the COP by around 2%. A 1 K increase, on the other hand, will increase the COP by 1.8% (Figure 1.70).

All high- and very-high-efficiency heating solutions using a HP, developed in recent years, rely on this principle of minimizing the temperature difference between the cold source, from which we draw heat, and the hot source, which is the fluid to

be heated. By minimizing this temperature difference, we maximize the HP's efficiency (average SCOP), and thus minimize energy consumption.

To achieve this objective, first, the heat emission systems must operate at low or very low temperature. During the design phase, we need to increase the emitter surface areas in order to satisfy the heating demand with water temperatures between 25 and 45°C higher than the space's set temperature. Underfloor heating using hot water is particularly well suited to this purpose. Note that modern buildings, built since RT 2012 came into force, have low heating demands.

In addition, HPs are more efficient if the cold source is at a relatively high temperature. For an air-to-air or air-to-water HP, it is preferable to use air extracted from the CMV system, from an unheated space, a veranda or patio, rather than air from the outside. In winter, when the outside air is cold, the COP decreases sharply, although this is when the heating demand is highest. For water-to-water HPs, the COP is generally higher, because the resource used tends to be warmer: a horizontal or vertical geothermal collector, groundwater, wastewater or foul water.

The COP of the real cycle is determined on the basis of the energy balance of the real cycle using refrigerant. It describes the performances of the HP's thermodynamic cycle, taking account of energy lost at the compressor.

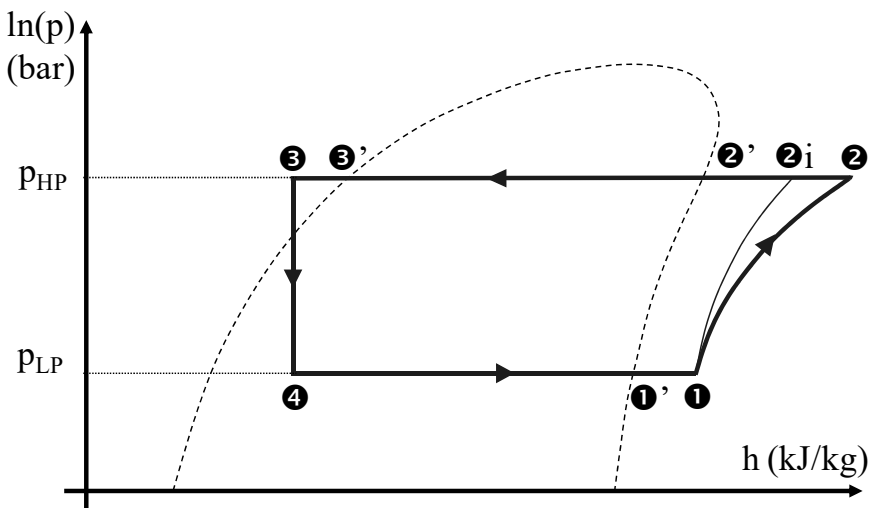


Figure 1.71. *Real cycle of a single-stage mechanical-compression heat pump*

The real cycle deviates substantially from the Carnot ideal cycle, because in practice, the transformations are not reversible. In Figure 1.71, the pressure drops in the exchanges and pipes are not considered.

The main differences from the Carnot ideal cycle are as follows:

– The compression phase ①→② is neither perfectly adiabatic nor reversible, because there is always friction. Real compression is not isentropic but polytropic. An isentropic efficiency η_i takes account of these phenomena. It is defined thus:

$$\eta_i = \frac{h_{2i} - h_1}{h_2 - h_1}$$

where h_1 is the enthalpy at aspiration into the compressor in (kJ/kg); h_2 is the backflow enthalpy of the compressor for a real polytropic compression; and h_{2i} is the enthalpy at point 2i at the end of compression (if the compression was isentropic, then $s_{2i} = s_1$).

The compression power added, also known as the indicated power P_i , in (W), is expressed thus:

$$P_i = q_{m_{FF}}(h_2 - h_1) = \frac{q_{m_{FF}}(h_{2i} - h_1)}{\eta_i}$$

where $q_{m_{FF}}$ is the flow rate of refrigerant in a closed circuit in the HP (kg/s). It is a function of the refrigerant's density when aspirated by the compressor ρ_a (kg/m³), the swept volume V (m³), the volumetric efficiency η_v and the compressor rotation speed N (rpm).

$$q_{m_{FF}} = \rho_a V \eta_v \frac{N}{60}$$

The volumetric efficiency η_v is the ratio between the aspirated volume flow rate and the swept volume flow rate of refrigerant in the compressor. Among other things, it includes dead volume losses and imperfect airtightness of the valves and piping segments. During the preliminary phase of a project, the following empirical formula can be used to calculate the volumetric efficiency as a function of the compression rate τ :

$$\eta_v = 1 - 0.04 \tau \text{ where } \tau = \frac{p_{HP}}{p_{LP}} = \frac{p_{ref}}{p_{asp}}$$

where p_{ref} is the absolute pressure of backflow of the compressor, which is the high pressure of the thermodynamic machine in (bar), and p_{asp} is the absolute pressure of aspiration of the compressor, or the low pressure of the HP.

Isentropic efficiency η_i characterizes irreversible phenomena (friction and heat losses to the outside) during compression. For standard operation of a compressor whose compression rates are not too high, the isentropic efficiency is close to the volumetric efficiency η_v .

– The HP condensation phase $\textcircled{2} \rightarrow \textcircled{3}$ is divided into multiple steps:

- desuperheating of the vapor $\textcircled{2} \rightarrow \textcircled{2}'$;

- condensation of the vapor $\textcircled{2}' \rightarrow \textcircled{3}'$;

- supercooling of the liquid formed by condensation $\textcircled{3}' \rightarrow \textcircled{3}$ to improve the COP and prevent vapor formation in the pipe, which gives us the expansion value. The supercooling is between 4 and 7°C.

The power transferred by the refrigerant to the hot source is:

$$\Phi_{cFF} = q_{mFF}(h_3 - h_2)$$

– The expansion phase $\textcircled{3} \rightarrow \textcircled{4}$ is neither adiabatic nor reversible. It is a Joule–Thomson isenthalpic expansion (i.e. $h = \text{Const.}$). After expansion, we have a LP liquid/vapor mixture (fog) at low temperature.

$$h_3 = h_4$$

– The LP evaporation phase $\textcircled{4} \rightarrow \textcircled{1}$ is divided into several steps:

- evaporation of the refrigerant $\textcircled{4} \rightarrow \textcircled{1}'$

- superheating at output from the evaporator $\textcircled{1}' \rightarrow \textcircled{1}$ to ensure that the fluid is all in the vapor state in the compressor to maintain the condition of compressor. The superheating is between 5 and 8°C.

The power which the refrigerant takes from the cold source is:

$$\Phi_{fFF} = q_{mFF}(h_1 - h_4)$$

The cycle's COP (COP_{cycle}), from the perspective of the refrigerant, expresses the ratio of the “exploitable” enthalpy difference, which is used to generate cooling, to the “payable” enthalpy difference, which is used to run the machine.

For a heat pump:

$$COP_{\text{heating cycle}} = -\frac{\Phi_{cFF}}{P_t} = -\frac{h_3 - h_2}{h_2 - h_1}$$

For a refrigerator:

$$COP_{cooling\ cycle} = \frac{\Phi_{cFF}}{P_i} = -\frac{h_1 - h_4}{h_2 - h_1}$$

For a thermo-frigo-pump:

$$COP_{(heating+cooling)\ cycle} = \frac{-\Phi_{hFF} + \Phi_{cFF}}{P_i} = \frac{-(h_3 - h_2) + (h_1 - h_4)}{h_2 - h_1}$$

The real thermodynamic cycle of the refrigerant in the heat pump is plotted, on the Mollier diagram, on the basis of:

- the refrigerant's evaporation pressure, which corresponds to LP;
- the refrigerant's condensation pressure, which corresponds to HP;
- the refrigerant's temperature as it enters the compressor;
- the refrigerant's temperature as it leaves the compressor;
- the refrigerant's temperature as it leaves the condenser.

The actual COP (COP_{actual}) expresses the ratio between the useful calorific power, which is actually extracted or recovered, and the electrical power used to run the machine. It includes thermal losses from the compressor, the condenser and the evaporator (see Figure 1.72).

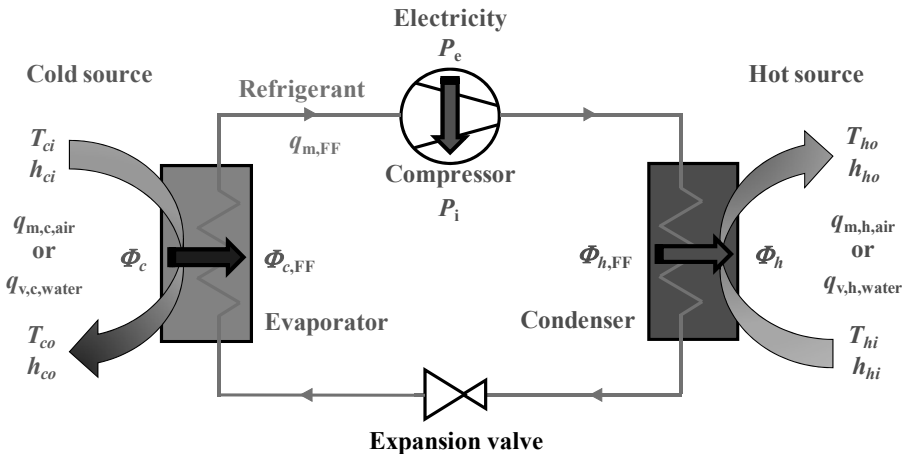


Figure 1.72. Diagram of a heat pump, with the relevant annotations

The actual coefficient of performance COP_{actual} is given as:

For a heat pump:

$$COP_{\text{actual heating}} = -\frac{\Phi_h}{P_e}$$

For a refrigerator:

$$COP_{\text{actual cooling}} = \frac{\Phi_c}{P_e}$$

For a thermo-frigo-pump:

$$COP_{\text{actual (heating + cooling)}} = \frac{-\Phi_h + \Phi_c}{P_e}$$

Φ_h represents the power supplied to the external heat source by the condenser:

$$\Phi_f = -\Phi_{fFF} \text{ if the heat exchange is lossless}$$

– For water:

$$\Phi_f = 1.16 q_{vc,water} (T_{fo} - T_{fi})$$

where Φ_h is the power at the condenser in (kW); 1.16 kWh/(m³.K) is the volumetric heat capacity of water; $q_{vc,water}$ is the volumetric flow rate of water to the condenser in m³/h; T_{fo} is the water temperature at output from the HP, in K; and T_{fi} is the water temperature at input from the HP.

– For air:

$$\Phi_f = q_{mc,air} (h_{fo} - h_{fi})$$

where $q_{mf,air}$ is the mass flow rate of air to the condenser, in kg/s; h_{fo} is the enthalpy of the air entering the condenser, in kJ/kg_{as}; and h_{fi} is the enthalpy of the air exiting the condenser.

At the condenser, the air is warmed. Only sensible energy is exchanged. Therefore, the calculation can be performed using the following practical formula:

$$\Phi_f = 0.34 q_{vc,air} (T_{fo} - T_{fi})$$

where $0.34 \text{ Wh}/(\text{m}^3 \cdot \text{K})$ is the volumetric heat capacity of air; $q_{vf,air}$ is the flow rate of air to the condenser, in m^3/h ; T_{fo} is the water temperature as it exits the HP, in K, and T_{fi} is the water temperature as it enters the HP.

Φ_c represents the power ceded to the outdoor cold source by the evaporator:

– For water:

$$\Phi_c = 1.16 q_{vc,water} (T_{co} - T_{ci})$$

where Φ_c is the power at the evaporator on the side of the cold source, in kW; $1.16 \text{ kWh}/(\text{m}^3 \cdot \text{K})$ is the volumetric heat capacity of water; $q_{vf,water}$ is the volumetric flow rate of water to the evaporator, in m^3/h ; T_{fs} is the water temperature as it exits the HP, in K; and T_{fe} is the water temperature as it enters the HP.

– For air:

$$\Phi_c = q_{mc,air} (h_{co} - h_{ci})$$

where $q_{mc,air}$ is the mass flow rate of air to the evaporator in kg/s, h_{ci} is the enthalpy of the air as it enters the evaporator in kJ/kg_{as} and h_{co} is the enthalpy of the air as it exits the evaporator.

At the evaporator, the air is cooled. However, when the dew point of the air is lower than the temperature of the evaporator surface, the water vapor in the air condenses, forming condensates that need to be evacuated. Thus, the air is cooled and dehumidified. Therefore, there is an exchange of both sensible energy (variation of the air temperature) and latent energy (variation of the moisture in the air). Consequently, the enthalpies must be included in the calculation of the total power exchanged.

P_e represents the electrical power absorbed by the compressor. Based on the indicated power P_i , which takes account of losses due to friction, we must include the mechanical losses characterized by the mechanical efficiency η_m (around 0.9), and the electrical losses of the motor driving the compressor, characterized by the electrical efficiency η_e (between 0.5 and 0.95):

$$P_e = \frac{P_i}{\eta_e \eta_m} = \frac{q_{mFF}(h_2 - h_1)}{\eta_e \eta_m} = \frac{q_{mFF}(h_{2i} - h_1)}{\eta_e \eta_m \eta_i}$$

The electrical power absorbed by the compressor motor is calculated on the basis of the voltage U in (V), the current intensity I in (A) and the phase-shift coefficient $\cos(\varphi)$ of the motor (around 0.8):

In a single-phase motor:

$$P_e = U I \cos (\varphi)$$

In a three-phase motor:

$$P_e = \sqrt{3} U I \cos (\varphi)$$

The different *COPs* presented here (ideal, cycle and actual) do not take account of the various pieces of connected equipment that are outside of the HP but affect its performance. In order to precisely define HPs' performances, we need to consider the electricity consumption linked to the auxiliary components (pumps, fans, etc.), defrosting cycles and heat losses from the networks. Thus, there are several different types of COP:

- The manufacturer's COP is measured in a laboratory under controlled conditions. For air-to-air and air-to-water HPs, the outdoor air temperature is 7°C, and for water-to-water HPs, the water temperature is 10°C.

- The overall COP includes the consumption by the auxiliary components (fans, pumps, etc.), defrosting cycles and heat losses from the networks. It is lower than the manufacturer's COP.

- The annual COP is a measure of the pump's performance, in a given context, over a year. This is the lowest COP.

- The SCOP (seasonal coefficient of performance) works in the same way as the previous indicator but considers variations in outdoor temperature over a heating season.

$$\text{SCOP} = \frac{Q_c}{Q_e} \quad [1.25]$$

For reversible HPs, the European Union has established energy labels that indicate the values of the SCOP (for heating mode) and SEER (for cooling mode) for a system operating in predefined conditions.

$$\text{SEER} = \frac{Q_f}{Q_e} \quad [1.26]$$

For reversible HPs, EUROVENT certification defines the ESEER (European Seasonal Energy Efficiency Ratio). Using this metric, we can compare machines on the basis of their ability to maintain high efficiency whatever the refrigerant charge, and thus minimize annual energy consumption. The ESEER is calculated as follows:

$$\text{ESEER} = 0.03 \text{ EER}_{100\%} + 0.33 \text{ EER}_{75\%} + 0.41 \text{ EER}_{50\%} + 0.23 \text{ EER}_{25\%}$$

where the terms $\text{EER}_{i\%}$ correspond to the energy efficiency for cooling at a load i , and the coefficients represent the percentage of the time for which the machine is supposed to operate at the load i over the season.

1.3.5.4.7. Outdoor sources for an HP

HPs use various outdoor sources to supply thermal energy to the heating networks of buildings:

- Air: the air is a useful resource, being available in unlimited quantities, for free. However, the air temperature varies over the year, which inevitably affects the HP's performances.

- Water: we can use groundwater, water from a well, a pond, a river or the sea, provided the regulations are respected in terms of intake and discharge. The water is stabler in temperature than air, which delivers a higher SCOP and prevents the defrosting cycles seen in HPs that use outdoor air as their source.

- Ground: an underground collector uses the energy contained in the surface layer of soil. There are various principles of collectors that could be used for this type of technology. The ground temperature is relatively stable all year round, meaning that a high COP can be achieved. Not having to draw from a water resource may be seen as a further benefit.

1.3.5.4.8. Reversibility of a HP

A HP is generally reversible. In the field of heating, the term “reversible” does not have the same meaning as in thermodynamics. A reversible HP can operate in heating or cooling mode. A four-way valve at compressor output can be used to reverse the refrigerant cycle (Figure 1.73). The condenser becomes an evaporator. Using an HP for cooling increases the building's electricity consumption, so this is not often done.

The compressor may have a constant rotation speed, with an on/off control system. In this case, the hydraulic network must have sufficient thermal inertia to limit the number of times the compressor turns on and off: the actual startup is very

energy-hungry, while continuous operation consumes less energy. If the inertia is not sufficient, then a buffer tank must also be installed.

Alternatively, the compressor may have variable rotation speed, responding to variations in the frequency of the electric current supplying the motor (this is an “inverter” system). The power of the machine adapts to the needs of the building. Such is the case with “scroll” compressors, controlled by inverter technology. These compressors comprise two spirals: one is fixed, the other moves in an orbital fashion, creating compression. The regulation system is more precise, energy consumption falls, and the number of restarts is limited.

The expansion valve may be thermostatic or electronic.

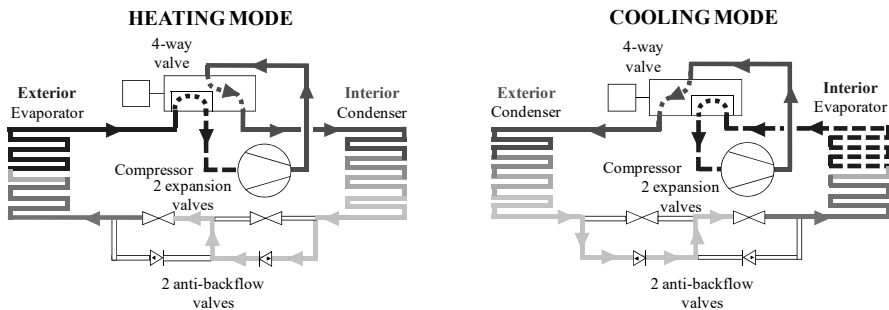


Figure 1.73. Working principle of a reversible heat pump

1.3.5.4.9. Air-source HPs

Air-to-air HPs take energy from the outside air to produce hot air, which is diffused through a network of ducts or indoor units (console, wall-mounted or ceiling-mounted vents). The system may be unitary or split. In a split system, an external unit usually houses the compressor, the outside air evaporator with its fan and the expansion valve; the internal unit comprises the ambient air condenser and its fan. A refrigerant circuit links the two units. This system is generally reversible, so can provide cooling as well.

The temperature of the outside air (cold source) varies throughout the year. When it is cold, the heating demand is high. However, as discussed earlier, the COP of the HP decreases when the outdoor air temperature falls. Thus, the heating power decreases, precisely when the heating demand of the building is increasing. An electrical auxiliary supply is required so as not to oversize the HP.

An HP's overall performance must also take account of the electricity consumption by the two fans and the defrosting cycles in the evaporator in winter. The machine may reverse the cycle and use hot gas to defrost itself, defrost using an electrical coil, or simply shut down and allow the evaporator to thaw naturally.

When installing the external unit, we must take account of the noise nuisance, which it may cause, and ensure good air circulation.

For *air-to-water HPs*, the energy from the outside air (or from an indoor space) can be used to produce low-temperature hot water (around 50°C) to supply the emitters (underfloor heating, low-temperature designed radiator, fan-convactor, etc.). In the same way as air-to-air HPs, they may be in one module or split, reversible, and controlled either in on/off mode or modulatable. They include a hydraulic module (circulator, expansion tank, safety valve, etc.), with an auxiliary supply.

1.3.5.4.10. Water HP

Water-to-water HPs use a water-rich resource (groundwater, a lake, a well, the sea, wastewater, etc.) to extract part of the thermal energy to supply hot water to a heating system (usually underfloor heating, radiators designed for low temperature, fan-convectors, etc.). As the temperature of that resource is relatively stable, the system performs well. It is important to check the quality of the resource in terms of flow rate and temperature over time. The initial outlay is high, because of the water intake and outlet technology that must be installed. When a borehole has to be drilled, it must be declared to the State beforehand, and the work must be done by a specialist borer.

1.3.5.4.11. Ground-source HP

HPs with underground collectors use the energy contained in the top layer of the ground. The ground temperature is very stable. It is topped up regularly by exposure to the sun ($H\ 20\ \text{W/m}^2$), rainwater, convective exchanges with the outside air and geothermal heat flow ($H\ 0.1\ \text{W/m}^2$). These systems are often referred to as geothermal HPs. The terms “geosolar heat pump” or “ground-source heat pump” are better suited to the reality. An average annual COP of 3.3 can easily be achieved. The collector is costly to install. There are various types of underground collectors:

- Horizontal collector filled with glycol-water mixture (Figure 1.74): The loops of pipe (limited to around 100 m) are made of PEX (external diameter 25–40 mm), buried at a depth of between 60 and 120 cm, with 33–80 cm between the pipes so as

not to excessively deplete the ground energy. There are collectors of varying geometric shapes (on one or two levels, across the whole of the surface or in a trench). In total, the surface area of the collector must be between 1.5 and 2 times the surface area of the building (poorly insulated buildings). In addition, the collector must be at least a certain distance from foundations, hydraulic networks (3 m), non-hydraulic networks (1.5 m) and trees (2 m). To prevent any risk of freezing, the fluid circulating in the collector is a mixture of glycol and water (15–40% glycol for protection between -5 and -40°C , depending on the region). The extracted thermal power is around $30\text{--}40\text{ W/m}^2$ (depending on soil composition); over the course of a season, this can equate to between 50 and 70 kWh/m² of extracted energy. The water-to-water HP supplies hot water to the low-temperature emitters in the building.

– Refrigerant horizontal collector: This system is made up of loops (limited to 60 m in length) of sheathed copper pipes (9–15 mm in diameter), buried at a depth of between 60 and 70 cm, with at least 40 cm between them. A clear surface area of around 1.5 times the area to be heated is needed for the installation of the collector. The power taken is around 40 W/m^2 . In the collector (evaporator), the refrigerant takes thermal energy from the ground and transfers it through the HP to the underfloor heating system with direct expansion (condenser). In this case, it is a ground-to-ground HP. This system, which is generally reversible, does not require an auxiliary supply. There is a significant volume of fluid present in the installation, and there are risks of freezing.

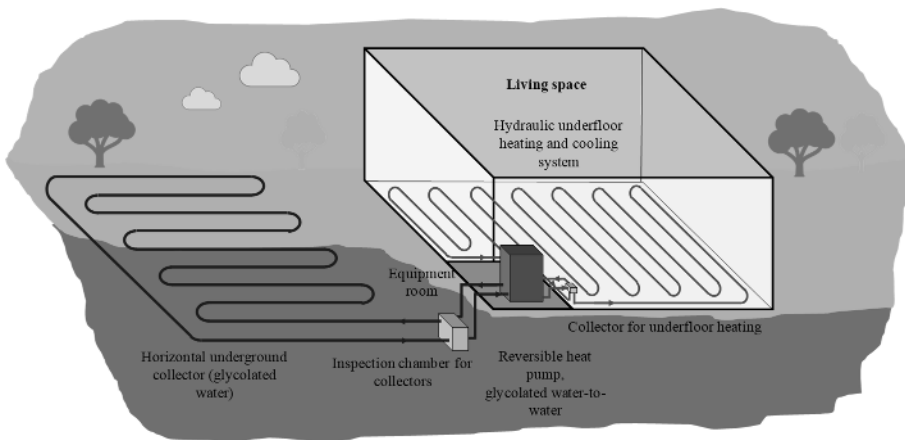


Figure 1.74. *Glycol/water-to-water heat pump with horizontal collector*

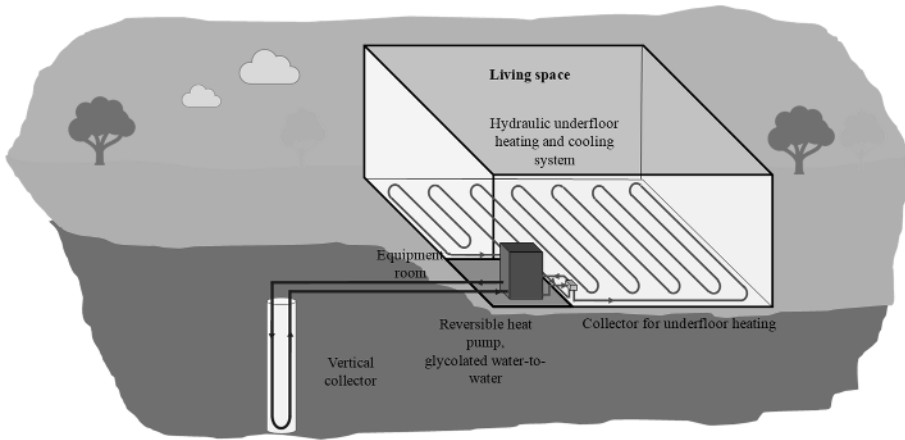


Figure 1.75. *Glycol/water-to-water heat pump with a vertical collector*

If the heat emitters (underfloor heating, for example) work by circulating hot water, this is a ground-to-water HP. This system allows the possibility of altering the energy production system over time.

– Water-circulation vertical collector (Figure 1.75): It is also known as a vertical geothermal probe. This system comprises a U-shaped PEX tube (external diameter 25–40 mm) installed vertically in a borehole (15–20 cm in diameter and 10–200 m deep). The borehole is then filled with bentonite (which has heat conductivity of 2.2 W/(m.K) and a low expansion coefficient) to stabilize and ensure good heat exchange between the tubes and the ground (the average thermal conductivity of dry soil is around 0.7 W/(m.K)). At higher power levels, multiple probes may be installed in parallel, spaced 5–10 m apart, to prevent thermal interactions between them. The levels of power extracted (per linear meter of probe) are 30–70 W/m, which can equate to 65–100 kWh/m energy captured over the course of a heating season. Overall, this solution is more costly than horizontal collectors. The main benefit of this solution is that it requires only a small area of ground. For example, two 50 m probes spaced 6 m apart could heat a 120 m² living space.

– Thermoactive foundations and geothermal piles: To keep installation costs down, geothermal collectors may be integrated into the foundations of a building. For certain buildings, the pipes may be placed in the wall footings; in this case, we speak of thermoactive foundations. For constructions with deep foundations, the PEX tubes may be implanted in the foundation piles. These are known as geothermal piles.

– Geothermal baskets: To cover the heating, DHW and cooling demands of high-performance buildings (compliant with RT 2012 or RE 2020) and limit installation costs, compact geothermal collectors known as geothermal baskets are available. The basket is between 30 cm and 1.5 m in diameter, with a height between 2 and 3 m. Such a system is capable of extracting between 700 and 1,400 W depending on the type of basket and soil type. It is quick and easy to implement, without the need for any heavy work, with simple equipment and a reduced surface area. For example, an RT 2012-compliant house of 100 m² will need seven 1 kW baskets, spaced 4 m apart and connected to a 7 kW glycol/water-to-water HP.

1.3.6. District heating networks (wood/gas)

A district heating network is a distribution system for heat produced centrally, which can serve multiple users. It comprises (Figure 1.76):

– One or more centralized heat production units: wood and/or gas heating system, household waste incineration plant, medium-temperature geothermal energy, etc.

– A primary heat distribution network, with a supply circuit transporting the primary fluid from the production system to the delivery point, and a return circuit bringing the fluid back to the production system. The fluid may be hot water (60–110°C for residential and tertiary-sector buildings), superheated water (110–180°C for certain processes) or steam (200–300°C for large networks such as that of the *Compagnie parisienne de chauffage urbain* (CPCU – Paris District Heating Company) and/or buildings such as hospitals which use steam).

– Exchange substations where the heat is transferred to each building's secondary network to deliver heating and DHW.

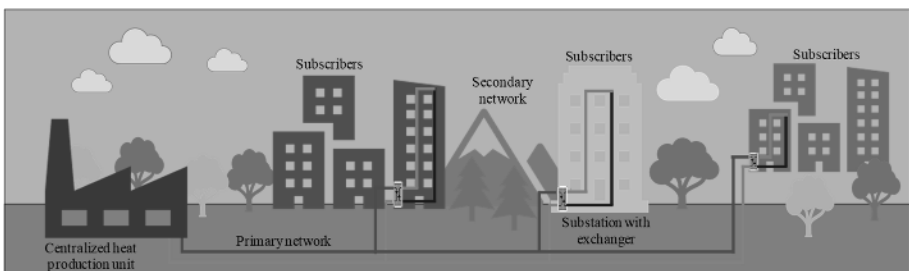


Figure 1.76. Simplified schematic view of a district heating network

To decarbonize energy production and foster the development of renewable and recovered energy, numerous district heating networks are currently being developed. In France, only 5% of heating demand is covered by district heating networks. However, this type of heating emits less than half as much greenhouse gas as fuel oil or gas (data from the Syndicat national du chauffage urbain (SNCU)). In Europe, over 6,000 district heating networks are active.

France's Law on the Energy Transition for Green Growth sets the target of quintupling the amount of heating and cooling delivered by such networks by 2030, compared to 2012: this will involve connecting the equivalent of 4–5 million additional homes to the networks.

To achieve this, the *Programmation pluriannuelle de l'énergie* (PPE – Multiyear Energy Plan), for 2023 and 2028, defined the targets for renewable energy to come from heating networks. The national objectives are 24.4 TWh of heating from renewable energy and energy harvesting by 2023, and between 31 and 36 TWh by 2028. In order to achieve that objective, an action plan needs to be put in place:

- building new networks when technically and financially possible;
- extending or densifying existing networks – for example, in the context of urban redevelopment projects;
- developing renewable energies and energy recovery to replace fossil energies in existing networks.

At the time of writing, there are over 800 district heating networks distributed across the whole of the French territory – most of them in urban centers. They serve the equivalent of around 2.4 million homes – around 3 million people. The heat delivered by these networks in 2019 was estimated at 25.6 TWh, which equates to around 41,000 connected buildings. The vast majority of this energy is consumed by residential buildings (54%) and tertiary-sector buildings (36%) (data from a survey by the *Fédération des services énergie environnement* (FEDENE – Federation of Services for Energy and the Environment) and the SNCU) (see Figure 1.77).

In 2020, renewable energies accounted for around 60% of the heat used in these networks. Biomass had also increased considerably, from 1% of the energy mix in 2005 to 22.8% in 2020. The proportion of heat drawn from waste-to-energy plants had increased by around 50%, accounting for 27.3% of the energy mix in 2020 (see Figure 1.78).

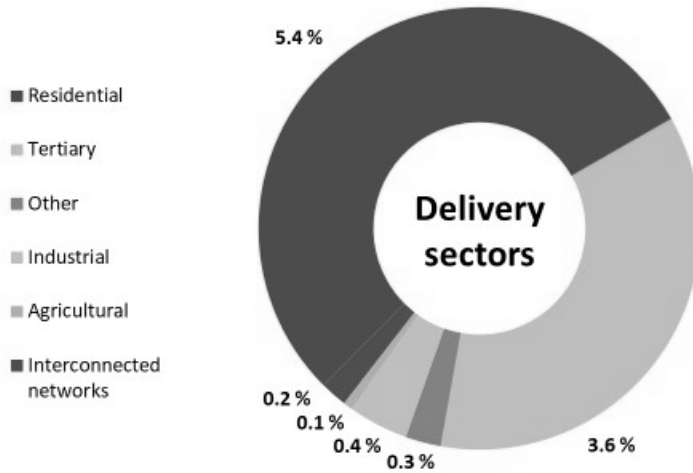


Figure 1.77. *Delivery of heating networks in France in 2019, by sector (source: FEDENE/SNCU survey)*

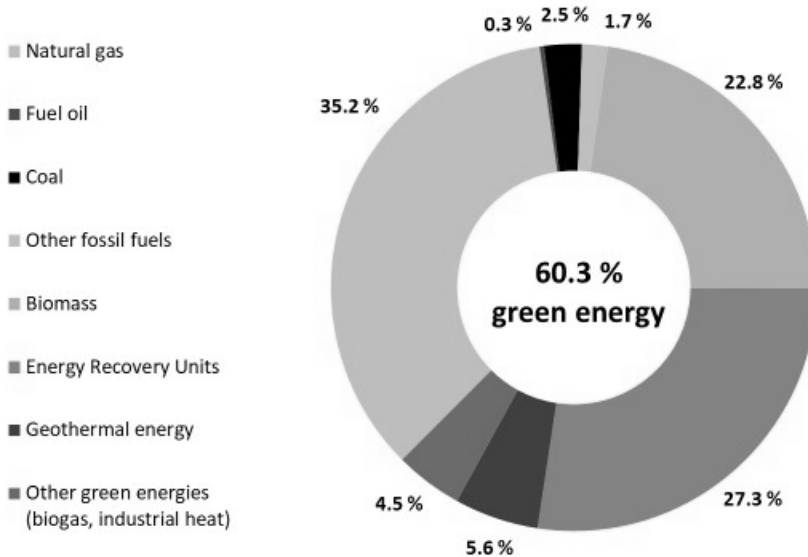


Figure 1.78. *Energy mix (incoming energy) of district heating networks in France in 2020 (source: FEDENE/SNCU survey)*

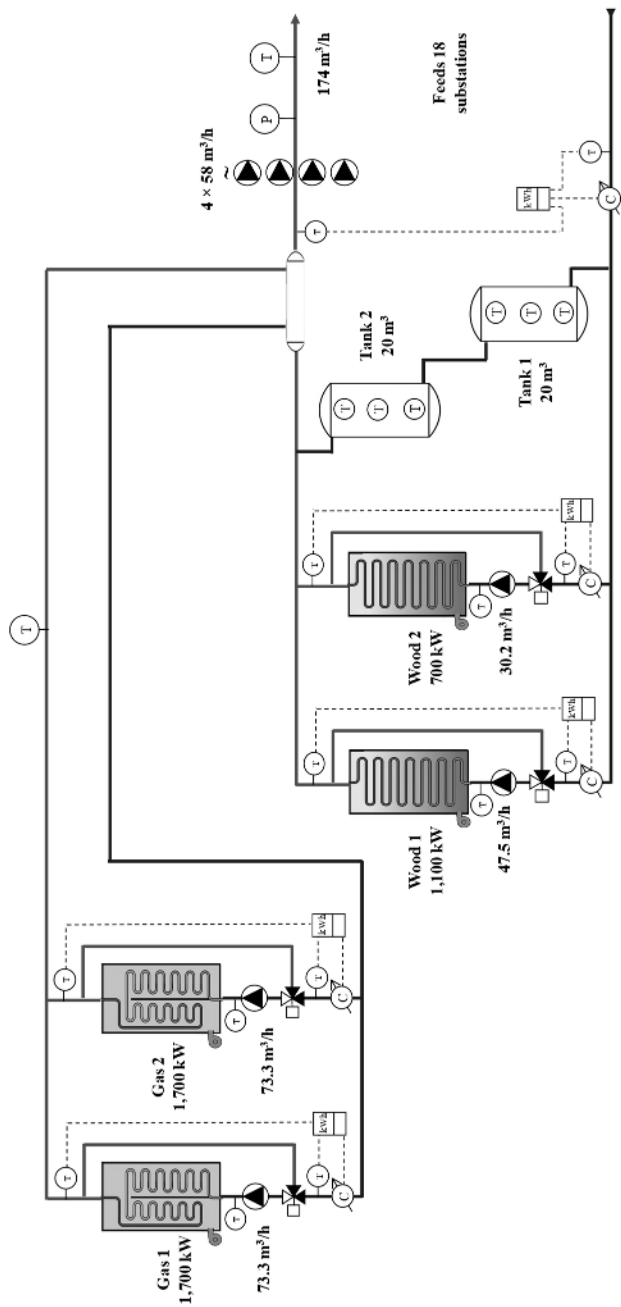
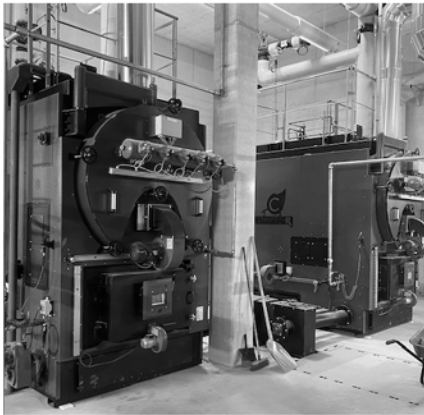


Figure 1.79. Simplified schematic view of the dual-fuel (wood/gas) production of the Biomass district heating network of Limoux

As a case study, let us look at the dual fuel network in Limoux, which was built by the *Syndicat audois d'énergies et du numérique* (SYADEN – Aude Regional Energy and Digital Association), and came into operation in 2022. This system provides heating and DHW to both residential and tertiary buildings. The production part comprises two wood-chip boilers with a nominal power of 700 kW and 1,100 kW, coupled with two 20 m³ accumulative hydro-tanks, supplemented by two gas-fired boilers with a power of 1700 kW. Note that 2.6 km of piping carries hot water (90–110°C) to 18 substations. Two 20 m³ accumulative hydro-tanks store heat for use during cold periods and for system restarts (see Figures 1.79 and 1.80).



a)



b)

Figure 1.80. Energy production in the wood/gas district heating system of Limoux.

(a) Two wood-chip boilers and (b) two gas boilers (source: Fabrice Castela and David Gely, with the permission of ENGIE Solutions)

The installation is controlled by a centralized technical management system. With optimized management, over 90% of demand was served by the wood boilers over the course of the first heating season, in the year 2022–2023.

1.4. Conclusion and prospects

The energy performance of hot-water heating systems is crucial for the buildings' sustainability and energy efficiency.

At present, gas boilers are widely used. They provide constant and reliable heat, but their carbon footprint remains a major challenge. Wood-fired boilers, on the

other hand, are considered more sustainable because they run on a renewable resource. However, they do require careful supply-chain management and control of fine particulate emissions. CHP generators, which produce both heat and electricity, offer an effective solution to get the most out of the primary energy that is, inevitably, used. However, often, they are only used in large buildings, because the initial cost is high. HPs are a fast-growing alternative, because they use renewable energy from air, water or sunlight to produce heat. They deliver high energy efficiency and a significant reduction in greenhouse emissions. However, their efficiency may be affected by the weather, and they may require a more substantial initial investment.

In the future, it is imperative to limit the heating demand and turn towards increasingly sustainable and smart building heating solutions. Artificial intelligence and smart networks will help to more efficiently manage the energy demand, while advances in thermal and electrical energy storage technologies will optimize the use of intermittent renewable energy sources, helping drive down the use of fossil fuels.