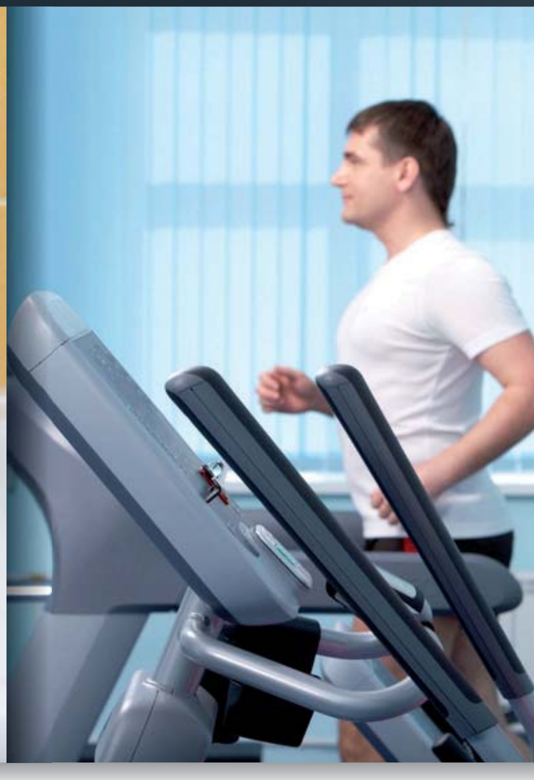
Guidelines for setting up Multifunction systems

RG66006968

COOLING + HEATING + SANITARY HOT WATER =

ONLY ONE SYSTEM FOR 365 DAYS/YEAR USAGE.



Hi Warm

Multifunctional split-type unit,
for residential applications
capacity from 12 up to 33 kW
condensing unit for indoor installation and
refrigerant/air heat exchanger for outdoor installation

MCP

Multifunctional unit, for residential, centralized and light-commercial applications capacity from 7 to 41 kW packaged unit for outdoor installation

Galletti Multipurpose solutions: one system, many answers

Benefits of Multifunctional systems

> Environmentally friendly thanks to excellent COP and EER values
> Reduction of fossil fuels consumption
> No risk of explosions, fire or intoxication from fossil fuels
> Fully programmable, with remote online supervision
> Low maintenance requirements thanks to the absence of parts subjected to wear
> Low noise operation
> No local emissions of CO_2 or polluting particles
> High energy availability extracted in many ways from the environment
> Eco friendly and fed by renewable energy sources

Dear Reader,

We would first like to thank you for taking your time to read this guide. The information given is based on the intuition (and trial and error) and efforts put in by all the companies that make up the Galletti Group.

We have tried to summarize certain points on setting up a system consisting of multifunction units. While reminding you that this information does not replace professional experience, which remains the best guarantee for the final result, we would like to give the reader some advice in our belief that the real benefits of a "true" multifunction unit can only be fully exploited if combined with a system able to make full use of its potential and (we are saying this for intellectual honesty) compensate for its limitations, when the real working conditions permit full use of the same unit.

We believe that the future of environmental convenience lies in better integration between the efforts of those who design and construct the units, on one hand, and those who design and construct the systems, on the other: for only then can we reduce the environmental impact of our work, without compromising final convenience.

We aim to give the correct definition of a "multifunction system" rather than of a multifunction unit connected to a plumbing system.

A final consideration: we believe that setting up a system based on electrically powered units is a farsighted choice: for if, as we are led to believe, the amount of electricity produced at zero impact is to increase in the future, in proportion to other forms of energy, the system we will set up today will become "greener" over the years to come. We would like to think that, in this way, our work today can look to the future.

While we make every effort to avoid inaccuracies, please let us know if you find any in this guide. We welcome all feedback.

Galletti S.p.A.

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A note for the reader

All the opinions, suggestions and diagrams in this guide
are not binding for Galletti SpA.
The Professional is solely responsible.

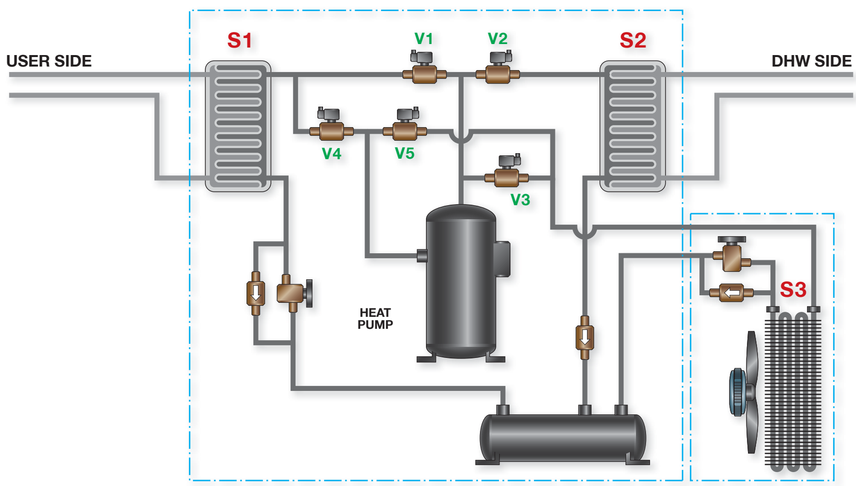
The functions of a multifunction unit

From now on, we will use the term “multifunction” in its "true" sense: in this case, a unit capable of total recovery during the summer period (not temporary cycle inversion to produce hot water for domestic use, resulting in the inevitable loss of comfort and efficiency). It is a so-called “4 pipe” unit that has to be hydraulically connected to two separate systems: The first is the building’s heating/air-conditioning system; the second is the hot water production system.
Generally, multi-functionality can characterise any air-to-water or water-to-water unit, with one or two circuits, in single package or split version, with ON/OFF compressor or inverter (the type with permanent-magnet motor, also known as BrushLess-DC).

In the interest of simplicity, let us first consider a **single-circuit air-to-water unit**. We can summarize the 5 different work modes of a “true” multifunction unit, during the different seasons, as follows:

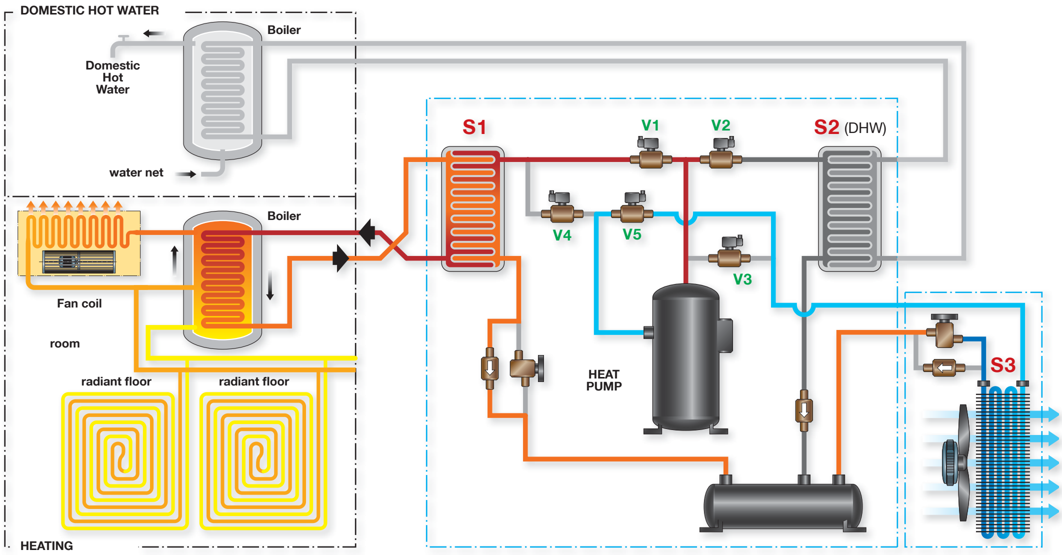
1. **SUMMER PERIOD:** Production of chilled water like a traditional chiller (water evaporation on "user side" and air condensing on "coil side").
2. **SUMMER PERIOD:** Production of chilled water with full recovery of condensing heat (total water condensing, for the production of domestic hot water (DHW) with dedicated plates).
3. **WINTER PERIOD:** Production of hot water for heating, like a traditional HP, heat pump (air evaporation on “coil side” and water condensing on “user side”).
4. **WINTER PERIOD:** Production of DHW, temporarily stopping the production of water for heating (prioritizing DHW); in other words, air evaporation on “coil side” and water condensing for the production of DHW with dedicated plates.
5. **MID SEASON:** Production only of DHW (in summer or winter) with evaporation by the coil.

Below is a diagram illustrating the principle of the cooling circuit of a multifunction unit:

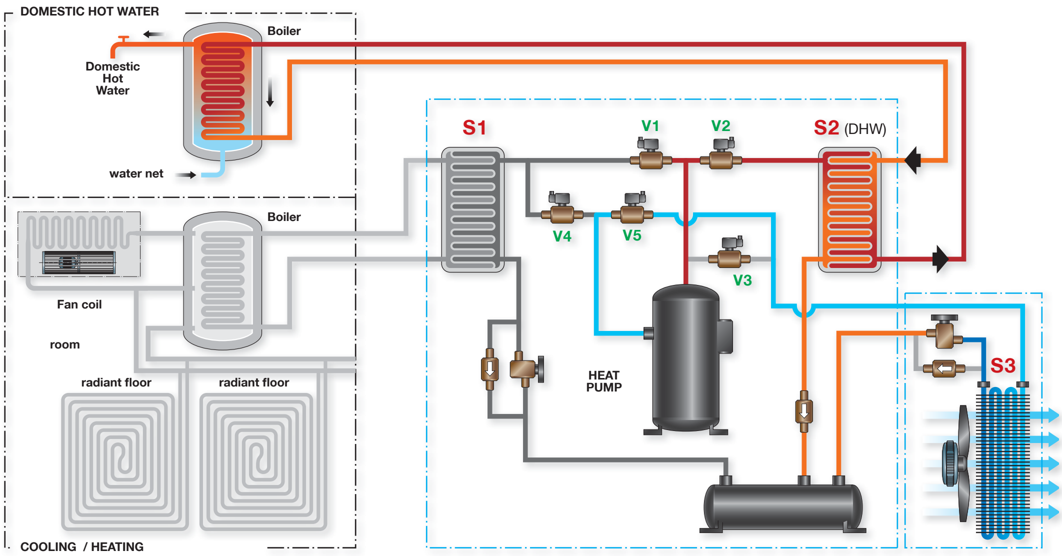


The flow of coolant in the various work modes is controlled by the solenoid valves shown in the picture, which open and close according to the type of function required.

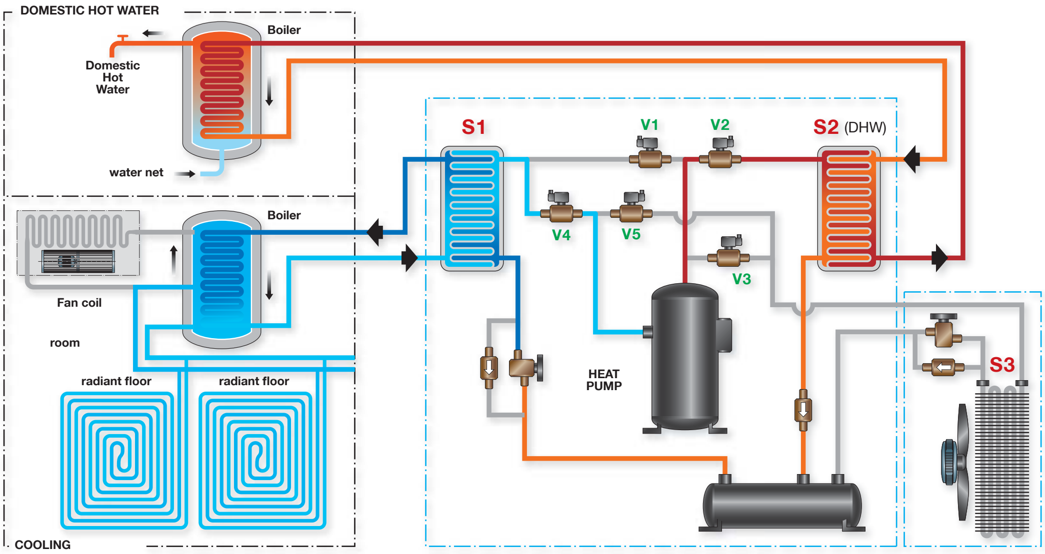
Winter Period



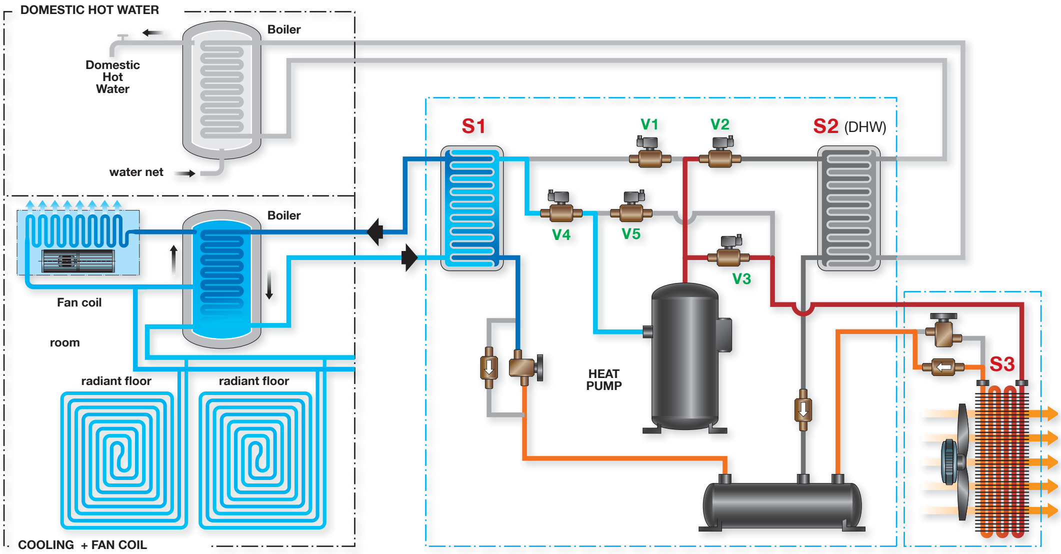
Domestic Hot Water / Mid Season



Summer Period (total heat recovery)



Summer Period

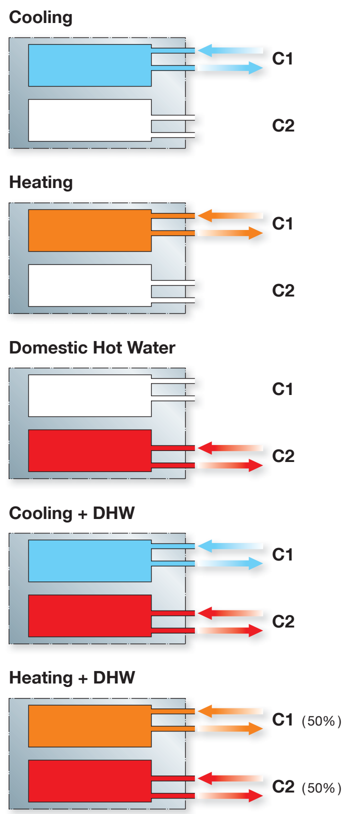


For **water-to-water units**, the following points can be made (the finned coil is replaced by a plate exchanger that uses water on “source side”:

• In the **SUMMER PERIOD:** Condensing water is carried out by the exchanger on “source side”.
• In the **WINTER PERIOD:** For the production of water for heating, evaporation is carried out by the plate exchanger on the “source side” and condensing is carried out on the “user side”; the same applies to the production of domestic hot water, with condensing not by the plates on the side of the system but by the plate on the side of DHW.
• **MID SEASON:** Production of DHW with evaporation still by the plate heat exchanger on “source side”.

For water-to-water versions, the source can be ground water or the fluid of a geothermal system, and so forth.

Lastly, in the case of a **dual-circuit unit**, it should be specified that it is possible for a circuit to produce DHW and another to continue condensing hot water for the heating system. This solution ensures maximum flexibility. Below are the possible uses of a dual-circuit unit:



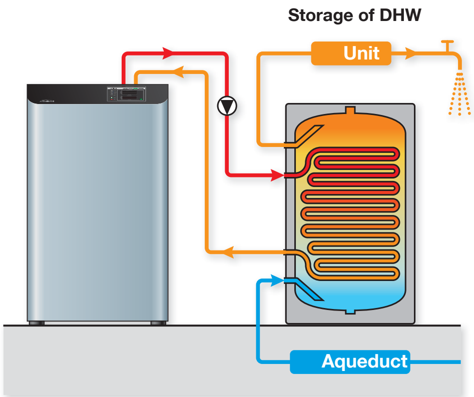
As can be seen, this type of unit can be used all year round. This is possible by a highly “flexible” cooling circuit controlled by a system that requires particularly advanced hardware and software.

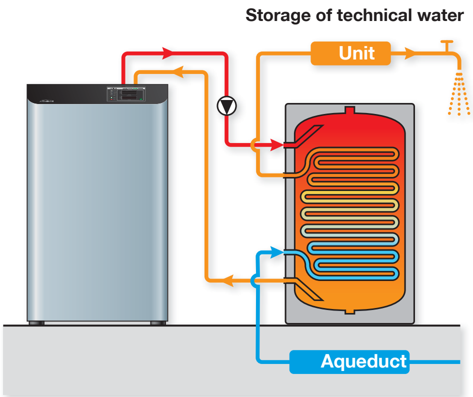
As mentioned above, the production of DHW is prioritized: this means that, regardless of the system work mode (heating or cooling), the unit will switch to the production of DHW when the signal is emitted by the sensor in the DHW tank or by a similar thermostat.

A few points:

First of all, the multifunction units (ON/OFF or modulating ones) are NOT suitable for “rapid heating", so thermal storage for the production of DHW is required.

The production of domestic hot water involves the storing of thermal energy in a “technical” water tank; the water for domestic use has to be heated by means of a stainless steel coil in the tank; this avoids the storage of domestic water and no anti-Legionella cycle is required (refer to the hydraulic diagrams for further details). This solution, which we recommend, is described in greater detail in the section on the storage of DHW.





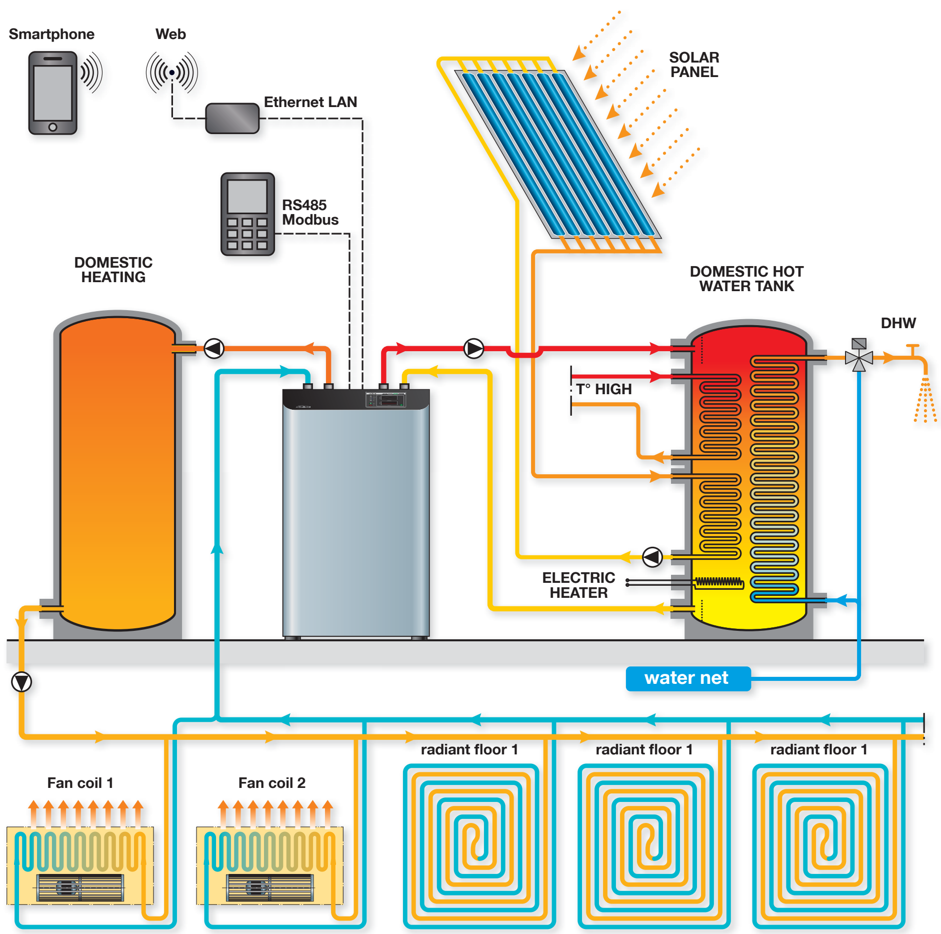
If however the designer decides to make provisions for the storage of DHW, the choice of tank must be suitable for the storage of potable water and be fitted with a coil (for use by the multifunction unit) the size of which permits thermal exchange proportional to the power of the unit, taking into account its operating temperature and the temperature at which the DHW is stored in the tank. Moreover, the DHW must not make contact with the plumbing circuit of the multifunction units (measures must be taken to ensure these are kept separate).

An anti-Legionella cycle is also required (it is programmed in the software of the multifunction units) which generally requires an additional heater (e.g. an electric heater). This does however involve higher energy costs which can be avoided with technical water storage.

With regard to the control of Legionella, there is an annex containing an extract of the “Linee Guida Trento e Bolzano per la prevenzione e controllo Legionellosi (Trento and Bolzano Guidelines for the prevention and control of Legionella), Official Gazette no. 103 of 05.05.2000”.

The main components of a multifunction unit

The main components of a multifunction unit are described briefly with reference to the diagram below:



The multifunction unit

As mentioned before, this is the heart of the multifunction system. It is a unit with 4 connections for use with two completely independent plumbing circuits: on the one hand, for the production of hot/chilled water for heating/cooling, and on the other, for the production of DHW.

The unit can be configured in whichever way is considered best for integration with the system (for example, the system’s and/or DHW circulation pumps can be installed either inside or outside the unit, but still be managed by the control logic with contacts on the unit’s electrical board).

The part connected to the heating system can be fitted (or not) with internal and/or external storage tanks (see further on) while inside the system there is always storage for the energy required to produce DHW.

The system can also be fitted with thermal solar panels (connected to the DHW tank by means of a coil and, therefore, hydraulically independent of the multifunction unit) and another source of high temperature that can be fed to the tank by means of a suitable coil at the top of the same.

To control prioritisation of the production of DHW, the unit comes with a temperature sensor to be inserted in a sump of the buffer tank. This sensor activates the production of DHW whenever the temperature in the tank falls below a configurable threshold value.

Alternatively, a clean contact for an external thermostat can be connected to the terminal block on the electrical board.

LCP

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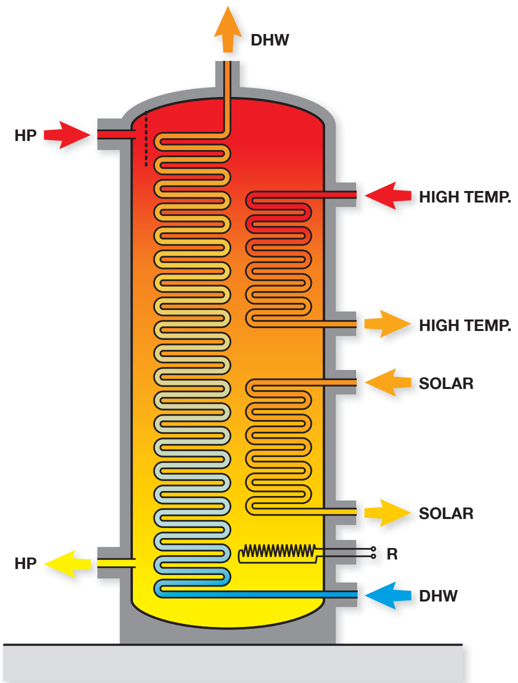
The DHW tank

As mentioned above, the function of the DHW tank is to store a quantity of energy (and at temperature) to meet the daily need for DHW, at the required temperature. As energy is stored rather than DHW (so we refer to "technical water"), the tank has to be fitted with a stainless steel coil for instant production. The exchange surface of this coil has to be such as to guarantee the production of hot water at least at the minimum temperature of comfort (e.g. 40 °C).

Otherwise, the hydraulic couplings for connection to the multifunction unit are without coil: the unit uses all the water in the tank.

In this way there is always sufficient water in the tank to guarantee correct operation of the unit (heat for DHW without unnecessary ΔT), without the compressor constantly switching on and off due to low water content and, therefore, reduced exchange capacity of the coil.



As can be seen in the picture, the coil for producing DHW is always the one with the greatest surface area. The coils for solar energy and high temperature (optional) have a smaller surface area (due to the higher temperature of the water heated with these sources)

It is advisable to install an "emergency" heater at the bottom of the tank (if possible) the absorption of which is no more than the maximum power absorbed by the multifunction unit. In the event of breakdown, it is still possible to "load" the tank at night and guarantee the production of DHW at an at least acceptable temperature.

A few points should be made concerning the hydraulic connections for the multifunction unit (marked “HP” on the drawing). Firstly, the water **inlet** is at the **bottom** and the **inlet for the hot water** produced by the multifunction unit is at the **top** of the tank. The hydraulic couplings (drawn with a dotted line) are fitted with **flow moderators** that preserve stratification in the tank by preventing hot water entering at the top from mixing with chilled water at the bottom.

Stratification is essential to maintain the required amount of energy and the production of DHW at a sufficiently high temperature.

The top part of the tank, therefore, always has to be kept at a temperature (e.g. 50°C) above that of the hot water (e.g. 40 °C). At the bottom, instead, as the temperature of the water taken from the mains is much cooler (e.g. 15°C), the thermal exchange is sufficient even if the temperature of the technical water is the same as that of the water from the multifunction unit.

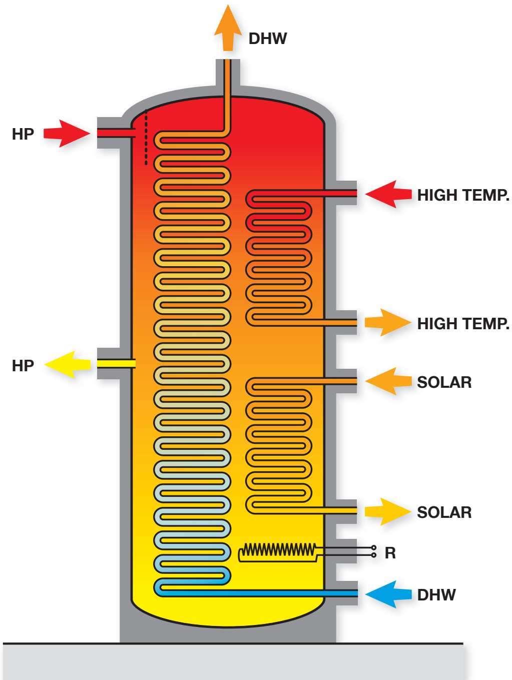
A few extra points should be made on maximising the efficiency of a tank with coil for connection to solar panels.

In the hottest season, it is likely that the energy required to produce DHW will be covered mainly by the multifunction unit in full recovery (naturally, the unit must also be used for air-conditioning). With that in mind, the presence of the thermal solar panel is not indispensable during the hottest season.
The same cannot be said of the season when air conditioning is not required, and the benefit of full recovery is not exploited. In this period, the thermal solar panel can provide part of the energy for DHW without consuming any (or virtually no) electrical energy.
This is possible, providing the temperature of the water in the tank (where the coil of the solar panel is located) is low enough to guarantee thermal exchange with the water produced by the panel, which is not as hot as that during the warmer months.
In other words, the bottom part of the tank (thanks to stratification) should preferably be kept at a temperature lower than that of the water from the multifunction unit, but still above that of the water from the mains. In practice, one should consider two superimposed areas with the coil at the bottom and the DHW inlet above the solar coil.

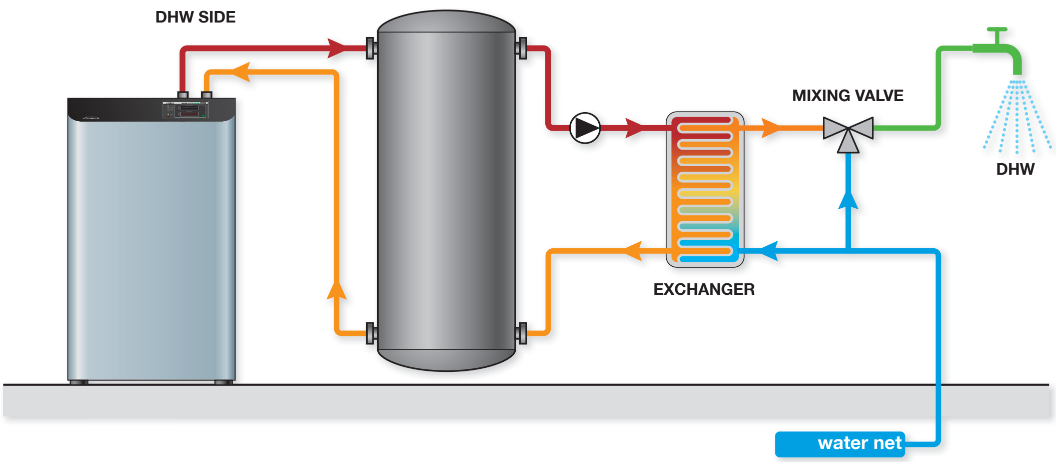
The figure below illustrates the tank described above:

The mains water that enters the bottom is in any case pre-heated by the energy received from the solar panel, before arriving at the top of the coil where the exchange is with the technical water heated by the multifunction unit.

The information below on estimating the minimum amount of water for the production of DHW (by the multifunction unit) apply, in this case, to the top part of the same tank (in other words, the quantity of water should be between the inlet and outlet, marked “HP” in the picture).



In many cases, it is common to use the so-called DHW “preparer”: It consists essentially of a tank with external plate exchanger that heats water from the mains in a quick and efficient manner. The circulation pump shown in the picture is activated by a water flow switch when the valve opens to let in the mains water, or opens intermittently to maintain the temperature of the plates of the external exchanger.



The points to take into account when sizing the system concern the content of the technical water tank and the properties of the plate heat exchanger.

Note: there are applications where the DHW tank is in fact used for storing hot water (from full recovery), for example to serve an air conditioning unit. In which case, the rules to be applied are the same ones as for the system's inertial storage.

Another key point is the **insulation of the DHW tank**: this must limit dispersion as much as possible while also maintain the temperature for a sufficiently long period of time. In fact, the interval between the tank "loading" phase and the phase for acquiring energy for DHW can be quite long. The typical type of insulation used is polyurethane foam about 100 mm thick, with λ=0.038 W/mK.

The system's inertial storage

As a general rule, this is simply an inertial storage, the sole function of which is to contain a sufficient amount of water to guarantee correct operation of the system (we will go into more detail further on).

There are several possibilities, some better than others. In particular:

I. Tank in series on the water return line (entering the unit).
This used to be the most common solution to limit variations in the temperature of the return line and to limit the number of compressor start-ups/hour. This type of solution is no longer necessary in terms of the compressor as the number of start-ups/hour is now controlled electronically. However, on the return line, the tank subjects the units to a "stepped" operating temperature (for example – in cooling mode - compressor ON = chilled water; compressor OFF = hot water)

II. Tank in series, on the delivery line (leaving the unit, towards the units)
The advantage of this solution is that it reduces variations in temperature for the system's units. However, the work load of the unit is the same as that of the system and if, for example, the units are fitted with 2-way valves, one has to take into account the possibility of variations in load. Diverter valves (or three-way valves) eliminate this problem.

III. Tank in parallel (hydraulic decoupling with booster pump in the direction of the system)
This is the most advanced solution. It permits decoupling the unit's work load in relation to the system's work load (that can go down to zero, for example in the case of critical start-ups, no thermal requirement, etc.). The engineering work and adjustment methods, however, are slightly more complex.

Excluding the solution with the tank on the return line, for the stated reasons, we can summarize the functions of the system's storage as follows:

1. Reduces variations in temperature towards the system's units. Although it is possible to reduce the minimum water content in a system using sophisticated control algorithms, low inertia units still do not work as well with "stepped" inlet water temperature.
Further information on reducing variations is given in the annex entitled "Inertial effects of system storage".

2. Maintaining the delivery temperature to the units in the event of the heating/cooling phase stopping due to the priority assigned to "calling" the DHW.

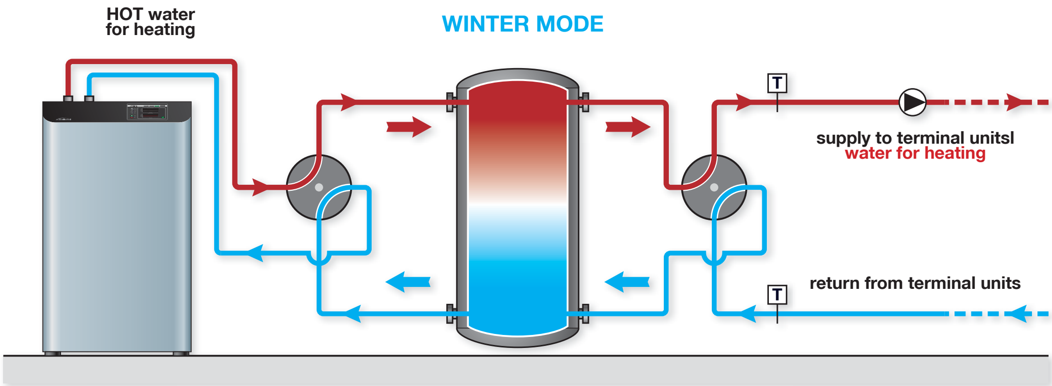
3. Decrease in delivery temperature to the units, in heating mode, during the defrost phase of the air-to-water unit, in winter-time.
This effect can be attenuated using "units" of high thermal inertia, but in all the other cases it represents a source of discomfort for the end user.

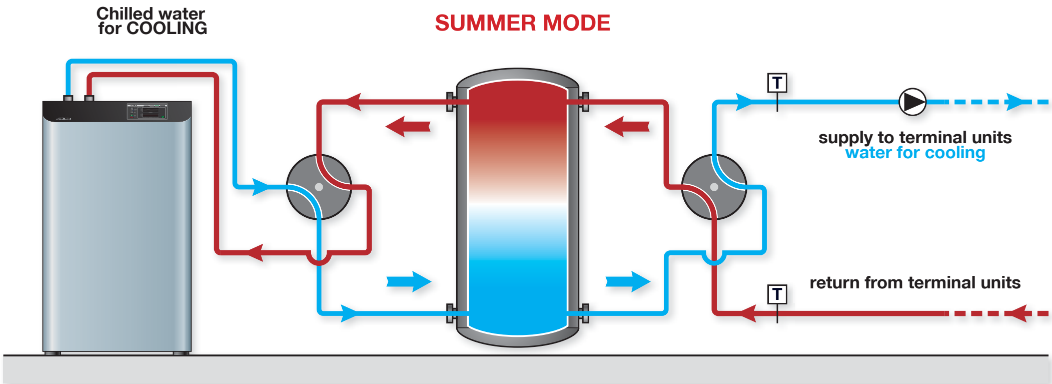
As mentioned above, it is possible that the tank shown in the diagram can also act as a "hydraulic separator" to ensure complete independence between the work load of the unit and that of the distribution system. In which case, careful attention has to be paid to loss of energy due to remixing of the delivery and return flows.

The figure below illustrates the installation of a circuit where two 4-way valves permit the flow of water to and from the tank following the natural course and, therefore, avoiding the loss mentioned above.

The coloured arrows refer to the operation during the heating phase (to the left of the tank is the unit that produces hot water and, to the right, the distribution system).

The arrows are inverted in summer mode: the chilled water produced enters at the base of the tank, as does withdrawal to the system.





NOTE: The water content in the system can vary considerably according to work conditions in the event of capacity control steps (for example, with two-way valves on the coils or similar). This has to be taken into account when sizing the tanks.

Room units

When talking of the units of systems, we mean those devices that distribute heat in rooms (we will refer, in the interest of simplicity, to the winter period).

A few points need to be made concerning their properties; if we consider simply two types of unit, the fancoil and underfloor radiant system, we can make a few comparisons and distinctions.

1. The first point, which applies to units of both high and low inertia, concerns the temperature of the water. A system consisting of steam compression units will not, in general, be able to produce water at the same temperature when there is variation in external conditions. In particular, operation in “heat pump” mode is affected by the temperature of external air, in terms of maximum thermal power and in terms of the maximum temperature of the water produced. One therefore needs to stabilize the most critical conditions of the unit, determine the maximum available thermal power, the temperature of the water flow to the units, and the integrated thermal power (net of the defrost phases – to be discussed further on).

2. Regardless of the type of unit, it must guarantee sufficient thermal power for that required, at the temperature of water produced by the unit. Practical problems can be encountered when, in critical conditions, the unit's thermal power is reduced (inevitable in thermo-dynamic terms) as is also the maximum temperature of the water produced, with the consequent decrease in the units’ exchange capacity, causing insufficient heating of the rooms. There are examples of perfectly working installations that combine radiators and heat pumps, but the units have to be selected with care due to the temperature range of the water.

3. The speed at which the environment is heated up has to be considered in relation to the intended use. A radiant system is not generally suitable for applications that require fast heating; in the case of constant use, however, it can work at temperatures that are more "convenient" in terms of the energy efficiency of the thermal unit. For air-conditioning, use "as required" (as in the case of direct expansion split systems) is not technically possible.

4. The defrost phases of air-to-water units should be considered from two points of view: The integrated power (net of that taken, for defrosting, from the water of the system) must be sufficient to meet requirements, and the temperature of the water in the system remains sufficient to permit correct exchange for the units. In the case of units of high inertia (e.g. for radiant floors), their thermal capacity can be sufficient to guarantee the reserve of energy required for defrost. In the case of units of low inertia (e.g. fancoils), one needs to guarantee sufficient water content (as described in the relevant section).

Controlling humidity and ambient temperature

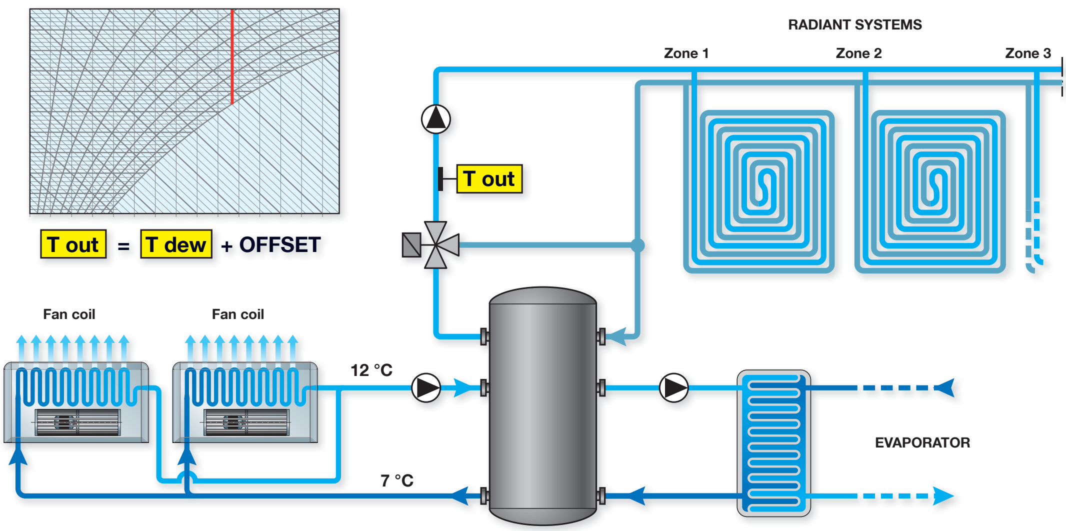
A heat pump, coupled to a fancoil or other ventilation units (typically in-built), can actively control ambient humidity as well as temperature. Inserting a fancoil in the false ceiling together with a radiant system can guarantee the maximum thermo-dynamic efficiency of the radiant system as well as hygrometric control. In fact, as chilled water can already be produced for air-conditioning, no other devices are needed (which in any case consume more energy and can be noisier due to the presence of a compressor, however small).

The picture below shows a “sensitive” underfloor cooling system (with control of the dew-point) and fancoil that controls dehumidification using water at 7°C produced by the multifunction unit.

A few extra points:

1. Humidity acts rather like a perfect gas, so the act of dehumidification (or humidification) can be “precise” but the effects quickly extend to the entire volume of air in the environment.

2. To improve the efficiency of the system, it is NOT necessary for the multifunction unit to constantly produce water at a temperature of 7°C; it is possible even to run the unit at temperatures slightly above the average (nearer those required by the radiant section), changing the set point only when dehumidification is required, e.g. by an advanced control that is also associated with the fancoil.



As shown in the picture, the temperature to the radiant section must be such as to avoid condensing phenomena (and therefore be higher than the dew-point which will be determined, for example, by the fancoil control, by measuring the ambient relative humidity and dry-bulb temperature).

Water temperature and start-up - condensing control

The compressor of the cooling cycle for the heat pumps guarantees high thermodynamic performance as it is a scroll compressor.
These compressors, in the versions with continuous speed variation, are lubricated with a flow of oil guaranteed by a small difference in intake and delivery pressure (and not by centrifugal force, as in the case of compressors that operate at fixed speed).
If this difference in pressure is any less, the oil will stop flowing with the risk of the mechanical components seizing up. In this case, an alarm signalling low delivery pressure stops the unit to prevent unwelcome consequences. Low condensing pressure corresponds, for a fluid, to a low temperature and in fact, if a condensing fluid is chilled too quickly, its pressure drops accordingly.
So if the heat pump is in heating mode and the water in the tank is too cold (the unit has been inactive for a long time or has been started for the first time in the season, or the tank has not been sized correctly), it removes heat too quickly from the condensing fluid.
In the case of units with ON/OFF compressors (constant rotation speed), lubrication can be maintained but a sudden and significant drop in evaporation pressure can also trigger unnecessary defrost cycles and, possibly, trigger the low pressure switch.

The thermal exchange in the condenser, therefore, needs to be reduced; there are two ways of doing this:

- Use a recirculation pump with inverter: this will constantly reduce the load that passes through the condenser of the heat pump, reducing thermal exchange and, therefore, maintaining the same delivery temperature and pressure of the compressor. The pump inverter is controlled directly by the unit control processor which bases itself on the pressure values registered every moment for the condenser. We recommend this solution as it permits accurate control and lower power consumption of the recirculation pump.

- Install a three-way valve between the condenser and storage which, when open, permits bypassing of hot water (from the condenser) for pre-heating the cold load from the storage. In this case the load that passes through the condenser remains constant (it is the temperature that varies), with an effect similar to the first on the cooling fluid. The three-way valve is controlled by a signal emitted by a temperature sensor at the inlet of the condenser, on the water side. This solution permits constant running of the pump.

Estimating the requirement for domestic hot water

The purpose of this section is to outline several "accessible" ways of estimating the amount of energy required for the daily production of domestic hot water.
As specified in the foreword of this publication, the information we provide does not replace the work of the designer but offers a quick way of calculating the quantities concerned.

The same applies if the system is fitted with a boiler the peak power of which avoids any discomfort for the end user, even when the size of storage is limited.

This method is a simple one based on Recommendation R03/3 of the Italian Thermotechnical Committee (CTI, Comitato Termotecnico Italiano). R03/3 has since been replaced by UNI/TS 11300-1:2008 and UNI/TS 11300-2:2008 but we decided to refer to the method proposed by the R03.

The idea is to:

1. Estimate the amount of energy required to produce DHW, taking into account the type of building (intended use and surface area), the number of occupants, the temperatures concerned, etc.

2. Define the minimum power of the multifunction unit (for producing DHW), after setting a "reasonable" time for thermal loading of the storage.

We should point out straight away that the required thermal power calculated in this way can be quite different to that required to heat the building: when selecting the unit we therefore need to prioritize the model able to meet the highest requirement (when the external air conditions are most critical).

RESIDENTIAL buildings

We can estimate the amount of energy required to produce domestic hot water using this equation:

$$Q_{DHW}=V_{DHW} \times \rho_{w} \times C_{S,w} \times N_{dd} \times (T_{w}-T_{0})$$

Where:

Q_{DHW} Thermal requirement for producing DHW (kJ)

V_{DHW} Daily requirement of DHW (litres/day)

ρ_{w} Water density (kg/m³)

$C_{S,w}$ Specific heat of water (= 4186 J / kg°C)

N_{dd} Number of days considered for the calculation (for ease = 1)

T_{W} Temperature for use of DHW in °C; normally 40°C

T_{0} Temperature of DHW from the mains, in °C; normally 15°C

Imagining we "thermally load" the storage within a certain period of time, we can determine the minimum power required for the multifunction unit with this simple equation:

$$P_{DHW}=\frac{Q_{DHW}}{t_{RT}}$$

Where:

P_{DHW} Power required (W) to meet the requirement QDHW, for the time tHP

t_{RT} Operating time in DHW mode for reloading the storage (s)

For conventional values of mains water at 15°C and production at 40°C, we have the following requirements **for residential buildings** (correct for the number of bathrooms and type of system control).

Overall surface area of house (m^2)	Water requirement V_{DHW} (l/m^2 day)	Energy requirement E_{DHW} (kJ/m^2 day)
S < 50	3	314
50 ≤ S < 120	2,5	262
120 ≤ S ≤ 200	2	210
S > 200	1,5	157

The correction factors referred to are the following:

Number of bathrooms	Correction factor Fs
1	1
2	1,33
≥ 3	1,66

Type of control	Correction factor Fs
Autonomous	0,9
Non autonomous	1

Below is an example of a calculation using the proposed method (for a living area of 200 m², with 2 bathrooms and autonomous control):

Input data and values from the tables:

$S_{OVERALL}$ (m^2)	V_{DHW} (l/m^2 day)	E_{DHW} (kJ/m^2 day)	Fs	Fd	T_{RT} (hours)
200	**2**	**210**	**1,33**	**0,9**	**3**

DHW Requirement (l/day)	Energy Requirement (kJ/day)	Power of Multif. Unit (kW)
479	**50274**	**4,7**

The required power must obviously be guaranteed in the critical design conditions.

It should be noted that the calculated power is the daily amount for “loading” the storage already at temperature. When the system is put to “intermittent use” (e.g. only for the weekend), one needs to take into account the time required to bring the entire storage up to temperature, from the equilibrium temperature at the moment the system is started up.

NON-RESIDENTIAL buildings

The energy requirements for non-residential buildings (per person, per day) are given in the tables below.

Type of building	Requirement V_{DHW} (l/person-day)	Requirement E_{DHW} (kJ/person-day)
Hotel - bedroom with shower	60	6280
Hotel - bedroom with bath	120	12600
Hotel - shared bathroom facilities	50	5240
Colleges - other communities	50	5240
Hospitals - shared bathroom facilities	50	5240
Clinics - facilities in bedroom	120	12600
Offices	20	2100
Establishments with showers	40	4190

We then use two multiplication coefficients (the number of people and the rate of occupation) to calculate the final value. The two coefficients that determine the type of system control and number of bathroom facilities are no longer considered.

In the example below, there are 20 people in hotel bedrooms with bath; the occupation rate is 0.8 and the loading time is 4 hours.

Input data and values from the tables:

N° people	V_{DHW} (l/pers. day)	E_{DHW} (kJ/pers. day)	F_{OCC}	T_{RT} (hours)
20	**120**	**12600**	**0,8**	**4**

DHW Requirement (l/day)	Energy Requirement (kJ/day)	Heating Capacity (kW)
960	**201600**	**14,0**

As in the previous case, the required power must obviously be guaranteed in the critical design conditions.

Tables of consumption

The tables below (based on trade articles and technical publications of many manufacturers of hot water storage tanks) give a general idea of the consumption of DHW at a variety of places:

ESTIMATED HOT WATER CONSUMPTION

Type of use	Litres per day	Notes
Schools	5	Per person
Barracks	30	Per person
Industries	20	Per person
Offices	5	Per person
Campsites	30	Per person
Gymnasiums	35	Per user
Laundries	6	Per kg of washing
Restaurants	10	Per meal
Bars	2	Per consumption

The estimated indicative consumption of DHW for common domestic activities is given in the table below:

DHW CONSUMPTION

Type of use	Litres at 40 °C	kWh 10-40 °C
Washing hands	2 ÷ 5	0,07 ÷ 0,17
Washing hair	5 ÷ 15	0,17 ÷ 0,52
Washing dishes by hand	13 ÷ 20	0,45 ÷ 0,70
Shower	30 ÷ 50	1,00 ÷ 1,70
Bath	120 ÷ 150	4,20 ÷ 5,20

The table above allows us - for example - to estimate the peak amount of hot water in the case of 3 people having a shower, one after the other, in a residential home.

Estimating available energy for heating

We will not in this case provide any data to estimate the thermal requirement of a building with approximate calculations. This subject is simply too complex and the variability of the many possible cases is such that the use of any table data could, in all probability, result in grossly incorrect valuations. Let us agree, therefore, that the thermal requirement of a building in critical design conditions is the work of a professional expert and merely a figure for our reasoning.

What we aim to do, however, as any choice of unit has to meet a certain requirement, is provide information to ensure the power required for the comfort of the end user is not insufficient.

To do so, we will consider certain points based on the thermodynamics of the situation, from the properties of the compressors to the intrinsic operation of the units.

Thermal power in relation to air temperature

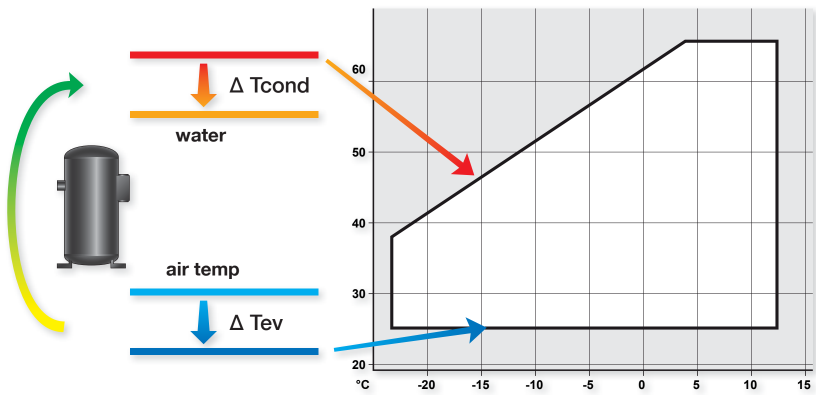
Without going into the mathematics of the cooling cycle, we will simply remember that the load of coolant developed by the compressor depends on the density of the intake gas.
In heat pump or DHW production mode, the compressor takes coolant evaporated in the finned coil, drawing heat from the air outside (the heat can be released back into the environment during compression).
The density of the gas used decreases in proportion to the temperature of the outside air and this inevitably leads to a decrease in available thermal power in relation to air temperature. This has to be taken into account, also because a decrease in air temperature can lead to an increase in thermal power requirement for heating.

One therefore needs to consider the power that can be generated in critical conditions. For example, a drop in air temperature from 7 °C (with relative humidity of 87%) to -5 °C can decrease the thermal power of a heat pump by 25-30%!

With this reasoning, the choice of unit should be very “generous” when the outdoor temperature is above the critical value, so that the thermal power produced increases and the requirement of the building decreases.
The benefit of units with capacity control is clearly very great in this regard, particularly in the case of constant modulation of a unit with "synchronous” compressor (with permanent magnet motor) the electrical efficiency of which is optimal even during regulation.

The maximum temperature of the water produced and of air temperature

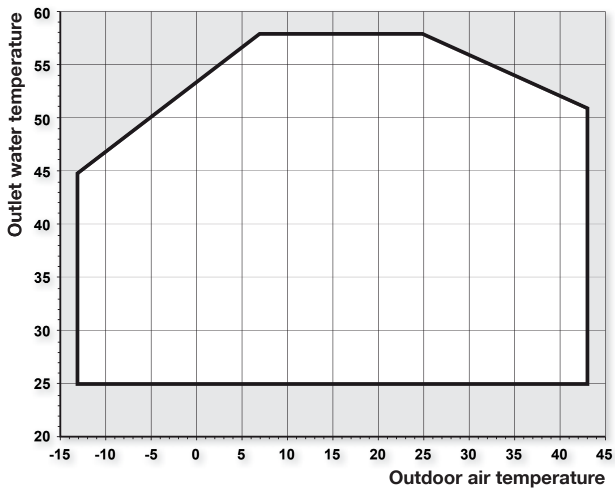
This is an essential point, in particular for the production of DHW and to guarantee the correct supply of energy to the environment (power generated by the units).
The graph below shows how operation of the unit (particularly when heating water up to maximum for heating/DHW) depends on the operational range of the compressor:



The graph on the right-hand side of the image shows the operational range of the compressor: Evaporation temperature forms an abscissa while condensing temperature remains level (the area in the shape is the permitted work range)

In other words, when the outdoor temperature is set, there is a DT that determines the evaporation temperature of the cooling cycle. This, and the work range of the compressor, indicates there is a "limit" to the condensing temperature and therefore also to the temperature of the water produced.

The figure below shows the typical course of the temperature of the water produced in relation to the external air:



Integrated power net of defrost phases

Let us start with a plain statement: air-to-water heat pumps defrost.
The defrost procedure allows the air-to-water unit to prevent the gradual "frosting up" of the evaporator coil which, at a certain temperature and degree of humidity, tends to cover in frost (which can become ice, particularly at the lowest point of the coil). (N.B. water-to-water units do not pose this problem).
Frosting up of the evaporator causes a gradual decline in thermal exchange, and therefore in evaporation temperature (pressure) which, if appropriate measures are not taken, will go down to the threshold at which the minimum pressure switch is triggered or even outside the compressor's work range. Essentially, gradual frosting up causes a gradual decline in the unit's thermal power. Measures therefore have to be taken to remove frost from the coil and ensure an acceptable degree of thermal exchange. One such measure is the defrost phase which involves the inversion of the cooling cycle, turning the evaporating coil into a condensing one, and the energy of hot gas (heated and condensed) breaks the ice down into water which can then be drained from the fin pack.

This operation requires a certain amount of energy which depends on the quantity of ice formed on the external coil. The energy required is taken from the hot water in the system (as the unit is in cooling mode) which will therefore decrease, to a certain extent, in temperature.

We have already mentioned the importance of taking into account the variation of power generated in relation to air temperature; let us now analyse the concept of integrated thermal power.

The unit manufacturers generally use multiplication coefficients for calculating integrated thermal power, starting with "continuous" values at different air temperatures. Here is an example:

	Dry bulb air temperature (°C)			
Multiplication coefficients	**-5**	**0**	**5**	**>5**
Basic control	0,89	0,88	0,94	1,00
Advanced	0,91	0,9	0,94	1,00

Basically, as the defrost phase requires the unit to change work mode, we have to consider that the value of thermal power during the defrost phase is likely to be negative (cooling power). By taking this into account and estimating the duration of the defrost cycles we can estimate the integrated power, which is the continuous thermal power (energy transferred to water) equivalent to the generated power net of the negative power for the defrost cycles.

In general, a defrost cycle involves a pause before cycle inversion (the compressor stops), the time required to run the inverted cycle (negative power), and a pause with the compressor turned off for draining the defrost water. If N_Def is the number of defrost cycles on an hourly basis, we can define (and calculate) the following times:

t_1	Pause before starting operation in chiller mode for defrost
t_d	Time in chiller mode for defrost
t_2	Pause before end of operation in chiller mode
t_{stop}	Total time with compressors turned off
t_{DEF}	Total time of operation in chiller mode (defrost)
t_{Pdc}	Actual time of operation in heat pump mode (within the span of an hour)

For the following equations:

$$t_{DEF} = t_D \times N_{DEF}$$

$$t_{STOP} = (t_1 + t_2) \times N_{DEF}$$

$$t_{HP} = 60 - t_{DEF} - t_{STOP}$$

To calculate the weighted average of the power times:

$$P_{H,\,med} = \frac{P_{heat} \times t_{HP} - P_{DEF} \times t_{DEF}}{60}$$

Where:

$P_{HEAT,\,m}$	Actual thermal power, taking the defrost cycles into account
P_{HEAT}	Thermal power in heat pump mode, in the specified conditions (air and water temp)
P_{DEF}	Cooling power during the defrost cycle (chiller)
t_{DEF}	Total time of operation in chiller mode
t_{PdC}	Actual time of operation in heat pump mode (within the span of an hour)

Here is a numerical example:

Input data

Power values			Defrost cycle times		
P_{HEAT} (kW)	N_Def (cicli/ora)	P_{DEF} (kW)	t_1 (min)	t_d (min)	t_2 (min)
15	2	15	0,5	2	0,5

This figure allows us to determine the following times and therefore estimate the average power net of the 2 defrost cycles/h:

Times on hourly basis			→	Average value
t_{stop} (min)	t_{DEF} (min)	t_{Pdc} (min)		$P_{HEAT,m}$ (kW)
2	4	54		12,5

It corresponds to a decrease of 16.7% in the "continuous" value.

The result, although an approximate estimation, indicates the importance of correct selection, especially when the heat pump is the only source of heat for the system.

Note: This method is obviously simplified; as the thermal power, particularly for cooling, depends on water temperature, the problem requires a full solution and not just simple balancing.

The characteristics of system storage

We have already discussed the importance of the content of water in the system from an inertial point of view. We will now see the influence of the size of storage in relation to the temperatures during the defrost phase.

Below is an example where the minimum temperature (which we will consider homogenous, in the interest of simplicity) at the end of a defrost cycle for a choice of inertial storage is at least equal to the bottom limit (at which thermal exchange or the generated air temperature is kept at a sufficient level).

Note: We will again using a simplified method.

We can use the following values:

P_{DEF}	Cooling power during the defrost cycle (chiller)
t_{def}	Time in chiller mode for defrost
T_{INIZ}	Initial temperature of the fluid (e.g. water)
T_{FIN}	Final temperature of the fluid (e.g. minimum acceptable value)
ρ	Fluid density
C_S	Specific heat of the fluid
ΔT_{DEF}	Decrease in the temperature of the fluid after defrost
E_{DEF}	Energy transferred to the fluid during defrost
V_{H_2O}	Minimum volume of fluid to obtain a final temperature no less than T_{FIN}
$V_{H_2O,\,SP}$	Minimum volume per kW of the thermal power of the HP

We can use this equation:

$$V_{H_2O} = \frac{E_{DEF} \times 1000}{\rho_{DEF} \times C_S \times \Delta T_{DEF}}$$

Where:

$$E_{DEF} = P_{DEF} \times t_{DEF} \times 60$$

$$\Delta T_{DEF} = T_{IN} - T_{FIN}$$

Below is an example in the interest of clarification (assuming the defrost power is more or less the same as the thermal power in the heat pump, if tables indicating the course of cooling power in relation to air and water temperature are not available).

Input data

Power values and times			Temperature values		Physical properties	
P_{HEAT} (kW)	P_{DEF} (kW)	t_{def} (min)	T_{INIZ} (°C)	T_{FIN} (°C)	ρ (kg/m³)	C_S (kJ/kg °C)
45,0	45,0	2,0	50	40	990	4,187

We can now determine the defrost data and minimum content for storage:

Defrost data	
DT_{DEF} (°C)	E_{DEF} (kJ)
10	5400

→

Minimum content	Specific content
V_{H20} (l)	V_{H20} (l/kW)
130	2,9

This is of course only an estimate, so a safety margin is advisable.

Remark: the calculation method we adopt is of course simplified; being the capacity a functions of the water temperature (especially in cooling mode), the problem should be solved by an integral method.

Characteristics and selection of domestic storage

The purpose of this section is to estimate the possible minimum temperature in the storage for a withdrawal of hot water that can be considered (in the design data) as a "peak withdrawal".

Note: We will again using a simplified method.

In the interest of simplicity, let us imagine a certain volume of hot water (that enters the storage at the temperature of the water mains, and leaves at the project temperature) is withdrawn at a given time. In the worst case scenario, the temperature in the storage falls immediately and the multifunction unit has to start up to compensate for this decrease in temperature. In this case, there is a flow of heat into the storage (the power of the multifunction unit) and one out of it (with the DHW); the difference between the two determines the net withdrawal of energy from the tank.

In other words, the following calculations are defined:

V_{DHW} Withdrawal of domestic hot water from the storage, in litres

P_{PdC} Power of the heat pump (or fraction) for the DHW storage, in kW

t_{DHW} Duration of withdrawal of DHW from the storage, in minutes

T_{IN} Temperature at which water enters from the mains (typically 15 °C)

T_{DHW} Temperature at which water leaves the storage (typically 40 °C)

T_{INIZ} Initial temperature of the storage (average) in °C

T_{FIN} Final temperature of storage (average) in °C (minimum acceptable value)

E_{PdC} Energy transferred to the storage from the HP during the withdrawal time, in kJ

E_{DHW} Energy removed from the DHW during the withdrawal time, in kJ

V_{H_2O} Minimum volume in the storage, to obtain a final temperature no less than T_{FIN}

The equation is:

$$V_{H_2O} = \frac{(E_{DHW} - E_{HP}) \times 1000}{\rho \times C_S \times (T_{INIZ} - T_{FIN})}$$

Where:

$$E_{HP} = P_{HP} \times t_{DHW} \times 60$$

$$E_{DHW} = \rho \times V_{DHW} \times C_S \times (T_{DHW} - T_{IN})$$

Below is a numerical example in the interest of clarification:

Input data

Power values and times			DHW temperature values		Storage temp.	
V_{DHW} (litri)	P_{HP} (kW)	t_{DHW} (min)	T_{IN} (°C)	T_{DHW} (°C)	T_{INIZ} (°C)	T_{FIN} (°C)
60	5,0	10	15	40	50	45

We can now determine the input and output energy and therefore the minimum volume of water in the storage:

Exchanged heat	
E_{HP} (kJ)	E_{DHW} (kJ)
3000	6216

→

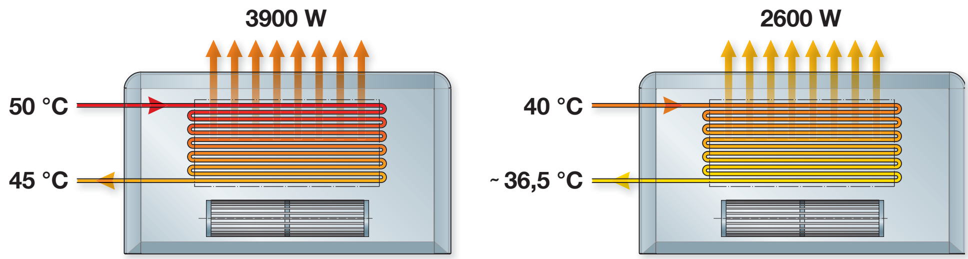
Minimum content
V_{H20} (l)
155

Again, this is of course only an estimate, so a safety margin is advisable.

Properties of the units

This section considers the power transferred to the environment in “critical” conditions, with a numerical example of how the power for heating varies according to the temperature of inlet water (at constant load). We will simply consider fan coils, the manufacturers of which provide methods for quick calculation.

Let us consider a model where water enters at a temperature of 50°C and leaves at 45°C, with an ambient temperature of 20°C; the thermal power would be around 3900 W (with a corresponding load of around 680 l/h). Let us now assume the load remains the same and, due to a defrost with insufficient water content, the temperature at which water enters falls right down to 40°C (at the same load). The power generated therefore amounts to 2600 W.



This simple example proves how important it is to estimate the temperature of the water during critical phases.

Selecting the multifunction unit

Taking into account all that has been said so far, we would now like to suggest how to go about selecting a multifunction unit.

First of all, you need to select a multifunction unit according to the power required in the design conditions of the heating phase.

You then need to check the values of the performance net of the defrost cycles, at air design temperature, the outlet water temperature and the power exchanged by the units at inlet water temperature.

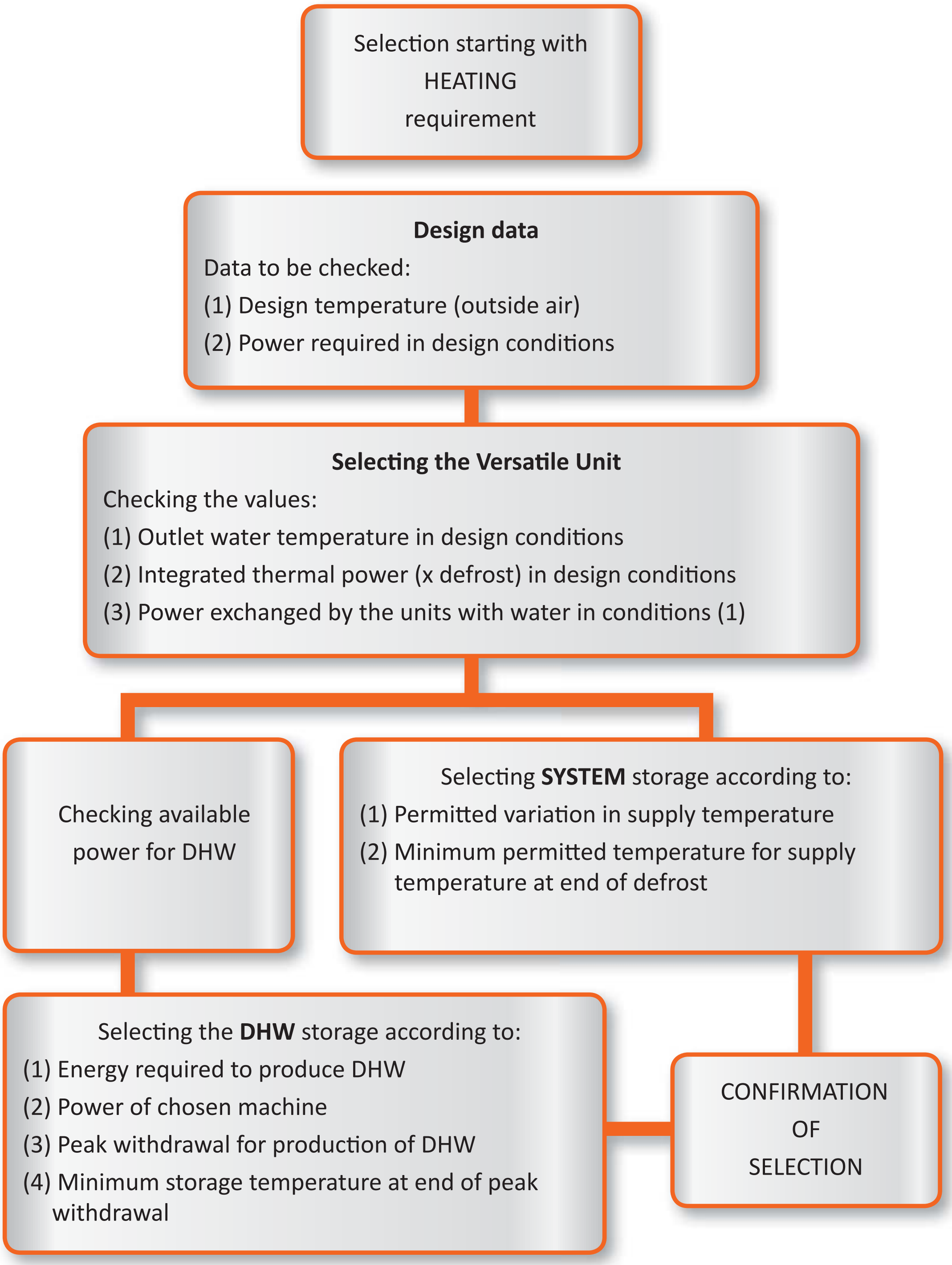
If the values are all positive, you can then check the storage for the production of DHW and for the heating system.

For the DHW storage, check the minimum water content in relation to the energy required for DHW, the power of the choice of unit for DHW, and the minimum temperature the storage has to be at for “peak” withdrawal.

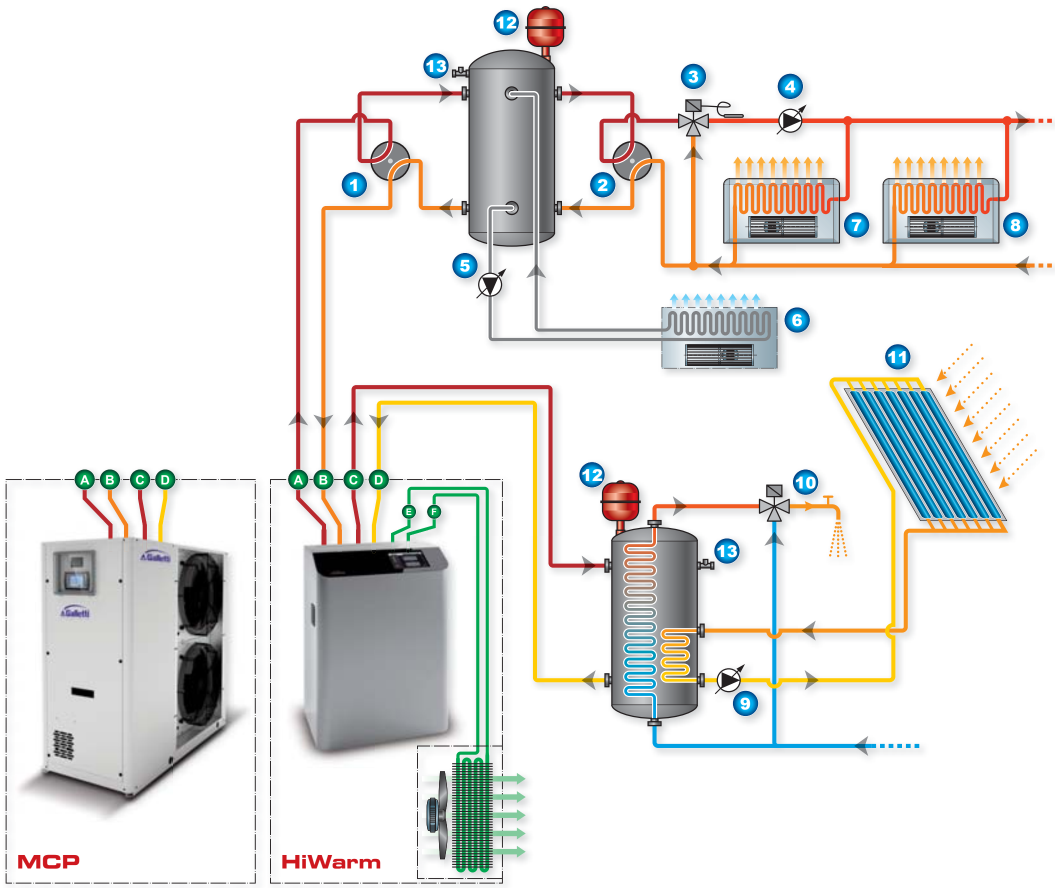
The storage for the heating system is selected on the basis of the inertial effect and its energy reserve function during the defrost phase.

The required size of the unit depends on all the above criteria.

The diagram summarizes this method.



You need to check the amount of DHW and power required, taking into account the type of building and number of occupants, as power requirements can vary considerably (for example, a Class A building will require a different system according to whether there is a single occupant or 4 occupants who each use DHW every morning).



1 - 4 way water valve
2 - 4 way water valve
3 - 3 way water valve
4 - inverter driven water pump (user side)
5 - inverter driven water pump (dehumidifying)
6 - fan coil unit for dehumidifying
7 - fan coil / radiant floor
8 - fan coil user side
9 - inverter driven water pump (solar panel)
10 - Domestic Hot Water
11 - Solar thermal panel
12 - Expansion water vessel
13 - Water safety valve

Conclusions

We believe the information we have given, while not exhaustive or argued in particularly theoretical or mathematical terms, is sufficient to have emphasized the importance of correct integration between a multifunction unit and a heating system.

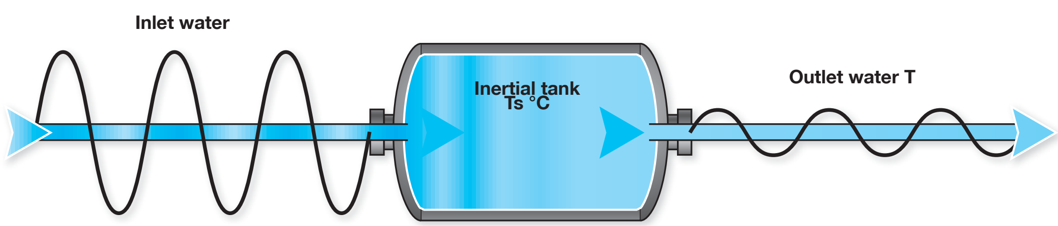
Some of our points may appear difficult to apply in practice or excessively cautious; if so, we would like to apologise and trust the reader understands.

Annex: Inertial effects of system storage

In the case of a unit with ON/OFF compressors (without capacity modulation), the action of the compressors is, in practice, intermittent, as the cooling requirement of the device may not match the cooling load generated by the unit. In the case of systems with low water content, where the effect of thermal inertia is less considerable, it is advisable to check the water content in the section for delivery to the units satisfies this equation:

$$V = \frac{P_F \times \Delta\tau}{\rho \; C_S \; \Delta T \; N_S}$$

V	content of water in unit section [m3]
C_S	specific heat of the fluid [J/(kg°C)]
ρ	density of the fluid [kg/m3]
Δt	minimum times between 2 start-ups of the compressors [s]
ΔT	permitted difference in water T [°C]
P_F	cooling power [W]
N_S	N° of capacity steps

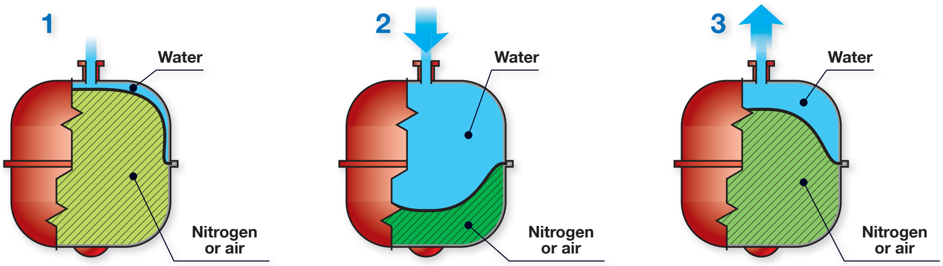


Annex: Sizing and preloading the expansion tanks

As you probably know, the purpose of an expansion tank is to compensate for the thermal dilation of the heat transfer fluid in a closed circuit (which can even be the cause of leaking pipes), as well as compensate for the harmful effects of "pressure surges" (when the flow of liquid in a pipe is stopped suddenly by closing a valve quickly). This annex analyses the closed membrane tank typically preloaded with nitrogen.

The closed expansion tank has a metal outer coating and a rubber internal membrane (made of elastic) connected to the plumbing system. The membrane expands as the volume of water increases in the circuit. This expansion is counteracted by a gas that fills the inside of the metal casing. After expansion of the water in the circuit, the gas which has been compressed pushes the water into the primary circuit.

The picture shows compensation of the volume with expansion of the membrane (compression of the gas).



Below is a common equation (source: ISPESL) for estimating the minimum volume of the expansion tank:

$$V_{VASO} = \frac{V_W \times (E_F - E_i)}{1 - \left(P_{MIN} / P_{MAX} \right)}$$

Where:

E_f and E_i are the coefficients for expansion of the fluid at the final and initial (maximum and minimum operating) temperature; these are given in the table below for water.

P_{max} and P_{min} are the absolute maximum and minimum operating pressure (the minimum pressure is determined by pre-loading of the tank, which must be at least 0.15-0.30 bar more than the static pressure at the point the tank is installed), while Vw is the volume of fluid in the system.

T	E	T	E	T	E
0°C	0,0001	35°C	0,0058	70°C	0,0227
5°C	0,0000	40°C	0,0078	75°C	0,0258
10°C	0,0003	45°C	0,0098	80°C	0,0290
15°C	0,0009	50°C	0,0121	85°C	0,0324
20°C	0,0018	55°C	0,0145	90°C	0,0359
25°C	0,0030	60°C	0,0170	95°C	0,0396
30°C	0,0043	65°C	0,0198	100°C	0,0434

As an example:

T_2=90 °C T_1=15 °C

P_{MIN}=1.5 bar,g P_{MAX}=3.5 bar,g V_w =1200 l

$$V_{VASO} = \frac{1000 \times (0.0359 - 0.0009)}{1 - \left(2.5/4.5\right)} \cong 95\ l$$

NOTE: If there is anti-freeze liquid in the system, you need to take into account that its thermal dilation coefficient is higher than that of water, so the expansion tank will need to be bigger.

Annex: Legionella in heating systems

Extract of the “Linee Guida Trento e Bolzano per la prevenzione e controllo Legionellosi (Trento and Bolzano Guidelines for the prevention and control of Legionella), Official Gazette no. 103 of 05.05.2000”.

PREVENTION AND CONTROL OF CONTAMINATION IN THE WATER SYSTEM

Below are summarized the methods that can be implemented after evaluation of the heating system, water system and environment concerned.
The measures for long-term prevention are associated with the good design of the systems, particularly in hospitals, thermal establishments and care homes for the elderly.

8.1 – THERMAL TREATMENT
The sterilising effect of increasing the temperature has been proven in hospitals and hotels.
Hot water systems kept at temperatures above 50°C are colonised less by Legionella.
Increasing the temperature of hot water is one of the methods currently used to control Legionella in the water distribution system. A temperature above 60°C combats Legionella in proportion to exposure time. (The temperature limits of 48° ± 5° C specified in article 5, section 7 of Presidential Decree no. 412 of 26.8.1993, “apply to heating systems... for the centralized production of hot water... for a multiplicity of facilities in houses...”)
8.1.1 - Thermal shock
The method
Increase water temperature up to 70-80°C for three whole days and leave the water running through the taps for 30 minutes each day. Some people recommend emptying the hot water tanks beforehand and to clean and decontaminate them with chloride (100 mg/L for 12-14 hours).
During the procedure, it is important to check the temperature of the water at the far points reaches or exceeds 60°C; if the temperature is not reached and maintained, the procedure cannot be guaranteed.
After completing this procedure, samples of water and sediments at either end of the system need to be taken for bacteriological testing.
If the testing reveals traces of bacteria, the procedure needs to be repeated until decontamination is conclusive.
After decontamination, microbiological testing should be carried out every so often, following the instructions in section 9.1.4.
Benefits
No particular equipment is required and it therefore can be carried out immediately, a real benefit in the event of an epidemic breakout.
Disadvantages
It requires time and staff, or the installation of remote probes, to check the temperature of the water at the far points and in the tanks, and the time to run the water. It is a systematic yet temporary method as the water system can be recolonised within a variable period of time, from several weeks to several months after the thermal shock if the temperature of the circulating water falls back below 50°C.

8.1.2 - Maintaining a constant temperature of 55-60°C in the system and upstream of mixing with cold water

This method is very efficient but involves high consumption of energy and high costs, not always compatible with the general criteria for energy efficiency. It can also pose problems of safety to users of the water system.

In practice:

in the case of dual-control systems, the first (a thermostat set at 55-60°C) controls storage temperature while the second (a mixer) controls the temperature at which the water is distributed, at 42-44°C.

Due to the temperatures normally used, Legionella cannot form in the boilers but only in the distribution and circulation systems.

To obtain thermal disinfection of these systems:

1. Bypass the mixer with an electric two-way valve controlled by a programmer clock,

2. Set at 60°C (with a thermostat) the temperature at which hot water is produced;

3. Increase the temperature of the bypass valve for half an hour at night when there is least consumption of water, by circulating water at 60°C.

I**n the case of systems where hot water is produced and distributed at 45-48°C**, at a temperature slightly higher than that for use, final adjustment is carried out at the single taps. Due to the relatively low temperatures, Legionella can colonise the boilers and the distribution and circulation systems. There are three reasons thermal disinfection is not practical for these systems:

1. It is only possible to use fixed point control systems with at least two levels: the operating level (45-48°C) and the disinfection level (60°C);

2. It is difficult to control disinfection times as one has to increase the temperature not only of the boilers but also of the distribution systems;

3. Water still has to be distributed at too high a temperature after the disinfection period as there is no control downstream of the boilers. Normally, due to these difficulties, it is more convenient to change the control system and use the one with thermostat and mixer.

Note

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