Module 2
Heat and Cold Energy Demands of Buildings

Part of the SHaKE Educational Package on District Heating and Cooling Systems

Guidebook

Reusable teaching resource for higher education and professional training

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Project

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Guide to using Module 2

This module is part of the SHaKE educational package developed to support the teaching of district heating and cooling (DHC) systems in higher education and professional training contexts.

The module explores DHC systems from the building level to the primary network. It presents different methods for determining space heating, space cooling and domestic hot water (DHW) demands, as well as the main principles of hydronic distribution systems used in buildings.

The module also covers the operation and sizing of key system components, including heat emitters, pipes, pumps, valves, substations and primary networks. Emphasis is placed on practical engineering applications and on understanding how building-side systems interact with district energy networks.

The guidebook is primarily intended as a reusable teaching resource for educators, lecturers and trainers. It can be used to support lectures, blended learning activities, classroom discussions, practical assignments and continuing professional development training. The accompanying presentation slides, question bank, self-check quiz and practical exercise are designed to help educators adapt the content to their own teaching context.

Students and professionals may also use the module for independent study, especially if they already have basic knowledge of thermodynamics, heat transfer and fluid mechanics.

Main topics covered in the module

- Space heating and cooling demand calculation methods
- Heat losses, heat gains and key thermal parameters
- Domestic hot water (DHW) demand and supply systems
- Building-side hydronic systems and heat emitters
- Hydraulic distribution systems, pipes, pumps and control valves
- Substations and control strategies
- Primary district heating and cooling network configurations and operation

Learning outcomes of Module 2

The module is designed to support the development of the following skills and competences.

Upon completion of this module, learners should be able to:

- Estimate space heating and cooling demands using different calculation approaches
- Determine domestic hot water (DHW) demand and select suitable supply concepts
- Analyse and size building-side hydronic systems and heat emitters
- Explain the operation and control of substations and hydraulic distribution systems
- Compare different primary network hydraulic configurations and their operational implications

- Analyse the interaction between buildings and district heating and cooling networks

Estimated workload

This module represents approximately **8 learning hours**, including guided study, independent reading and practical exercises.

Educators may adapt the workload depending on the level of the course, the selected materials and the teaching format. The module can be used as:

- a complete teaching unit,
- a set of selected lecture materials,
- a blended learning component,
- a practical exercise package,
- or a supplementary resource for existing courses.

Activities and Learning Hours

Activity Type	Time Allocation (Hours)	Description
Lectures	4h30	Sequence 1: Description of the building demands, key figures Sequence 2: Description and methodology of the calculation of space heating and cooling. Sequence 3: Description and methodology of the calculation of DHW demands Sequence 4: Description and methodology to design building side hydronic systems Sequence 5: Description of the operation of the substations and their control Sequence 6: Desciption and methodology to design the primary network
Case Studies	4h	Mini project to apply the concepts covered in the sequence 2,3,4 and 5
Self-Study	2h	Preparation for assessment and research for the project
Assessment	30min	Quizz, project
Total Learning Hours	11h	

Target groups

This guidebook is primarily intended for educators, lecturers and trainers who wish to use or adapt the SHaKE materials in their own teaching or training activities.

The primary target groups are:

- higher education lecturers in engineering, building services, energy systems and related fields,

- trainers involved in professional or continuing education on district heating and cooling,
- educators developing blended, modular or practice-oriented learning activities,
- academic staff seeking reusable teaching resources on sustainable district energy systems.

The materials may also be used by students, professionals and independent learners who wish to study selected topics individually. For independent use, basic prior knowledge in the relevant engineering fields is recommended.

Recommended use by educators and trainers

Module 2 can be used flexibly as a complete teaching unit or as a set of selected teaching resources. Educators may combine the materials according to the level of the course, the available teaching time and the intended learning outcomes.

Teaching need	Suggested Module 2 resource
Introducing the topic	Use the module overview and introductory slides to present the scope, key concepts and relevance of heat and cold energy demands in buildings.
Preparing learners before class	Assign selected guidebook sections as pre-reading for flipped classroom or blended learning activities.
Supporting lectures or seminars	Use the presentation slides to explain calculation methods, system components and engineering design principles.
Checking understanding	Use selected questions from the question bank for discussion, short in-class checks, formative assessment or LMS-based quizzes.
Developing applied engineering skills	Use the practical exercise for group work, homework, project-based learning or guided calculation sessions.
Supporting independent review	Direct learners to the module-level self-check quiz after they have studied the relevant materials.
Adapting to local teaching contexts	Select, combine or modify the guidebook sections, slides, questions and exercise according to the curriculum, participant level and professional context.

Recommended pathway for independent learners

Students and professionals using the module independently may follow the sequence below.

Review the introductory presentation

Familiarise yourself with the main concepts, terminology and structure of the module.

Study the guidebook materials

Read the detailed explanations, engineering principles and calculation methods presented in the guidebook.

Watch the module video

Use the short supporting video as an orientation to the module's main topics and learning focus.

Complete the self-check quiz

Test your understanding of the key concepts and technical principles covered in the module.

Attempt the practical exercise

Apply the acquired knowledge through engineering-oriented sizing and calculation tasks related to district heating and cooling systems.

Available supporting materials

The supporting materials include:

- presentation slides for the module subtopics,
- a reusable question bank,
- a module-level self-check quiz,
- a practical engineering exercise,
- and a short supporting module video.

Table of Contents

1. Introduction

The first objective of DHC networks is to supply space heating, domestic hot water or cooling to buildings in order to meet customer requirements. It is therefore essential to assess the space heating, domestic hot water and cooling requirements in connected buildings to design appropriately the DHC including pipes, substations and production units.

2. Space heating demand

The first step is to determine how to calculate the space heating demand in buildings. Depending on the available data, several options are possible: the thermal signature method, the European standard or the building energy simulation.

a. The thermal signature method

The thermal signature method can be used to rapidly determine the space heating needs. This method is adapted to existing buildings when historical measurement data on the power and outdoor temperature are available. The thermal signature method aims to plot the relation between the daily or hourly power needed and the outdoor temperature. As the space heating demand is thermosensitive, the relationship between the power $\dot{Q}$ (W) and the outdoor temperature T_{out} (°C) must be linear (see Figure 1):

$$\dot{Q} = K\,(T_{in} - T_{out})$$

Where K represents the global heat transfer coefficient of the building in (W/K) and T_{in} the indoor temperature of the building often set between 18 and 20 °C.

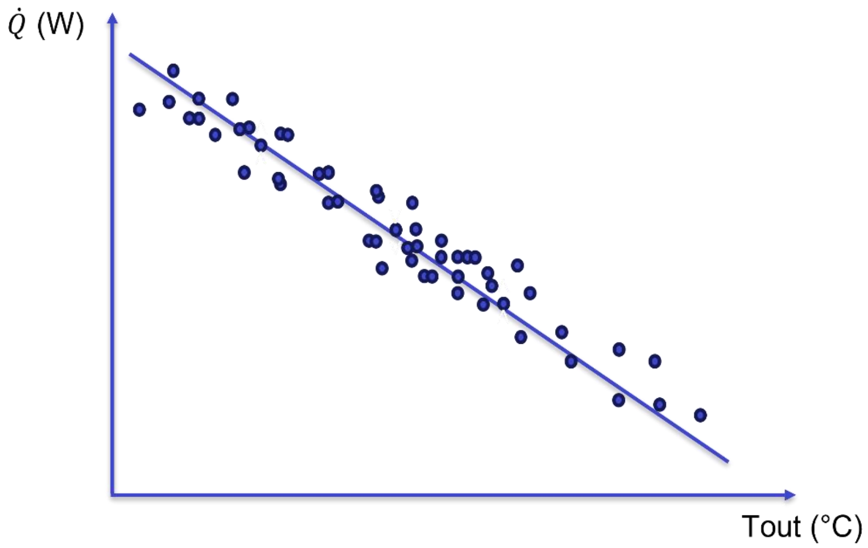


Figure 1 Example of the thermal signature of a building

The thermal signature of a building can also be drawn through the building's consumption (kWh) and the heating degree days (HDD in °C) (see Figure 2).

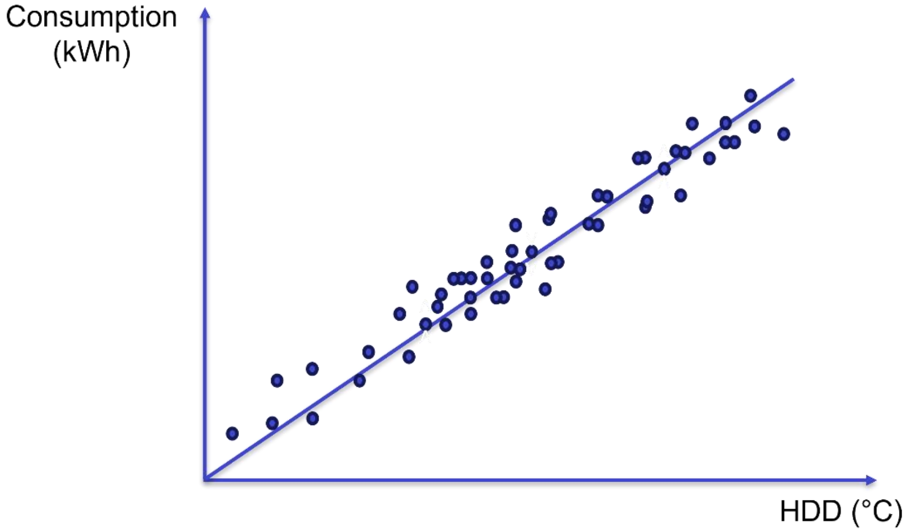


Figure 2 Example of a building thermal signature plot through the consumption and HDD

The heating degree days (HDD) stand for the difference between the daily mean outdoor temperature and the indoor temperature (set point temperature) of the building. The sum of HDD on a year represents the harshness of the climate. Indeed, the higher annual HDD are the colder is the climate. Figure 3 presents the HDD in EU countries in 2022.

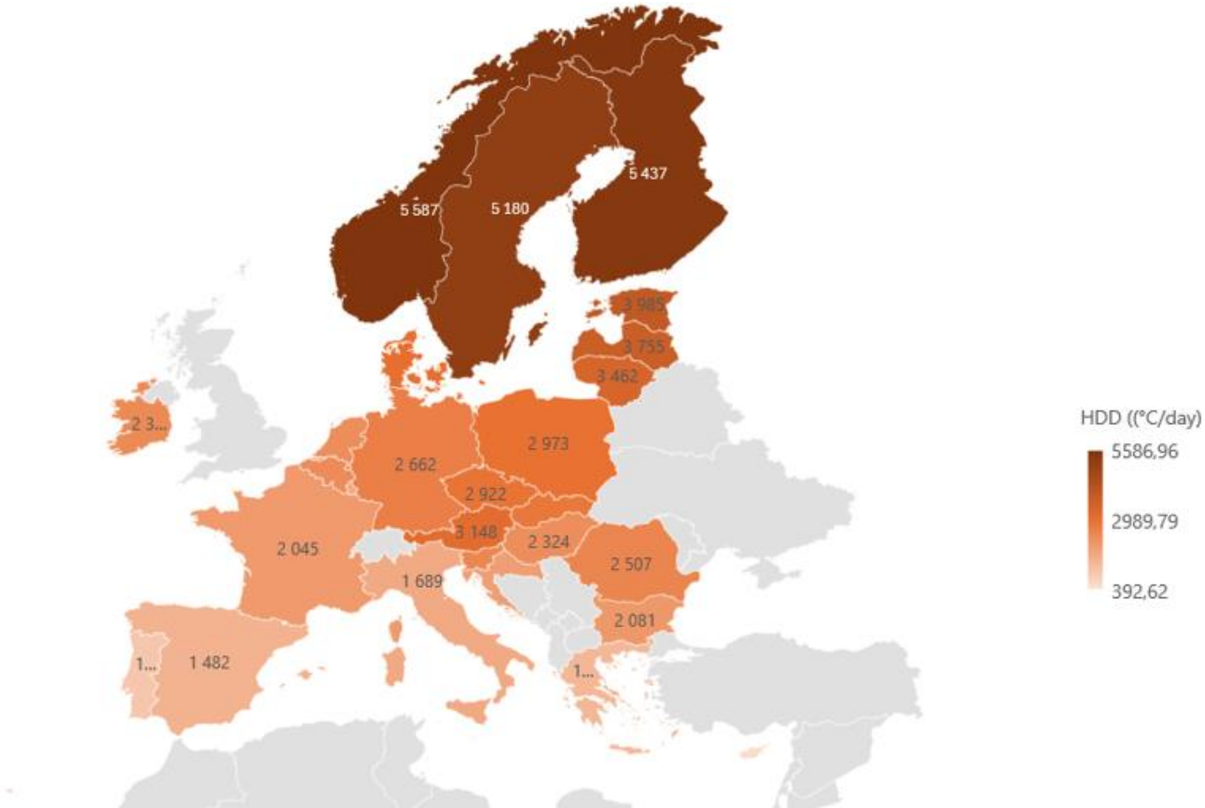


Figure 3 Yearly heating degree days in 2022 for EU countries

The HDD are calculated according to a reference temperature T_{ref} i.e. the indoor set point temperature of the buildings or an average daily outdoor temperature at which it is no longer necessary to heat the building (this temperature is lower than the indoor set point temperature and depends on the building thermal losses and internal gains

– Its value can be assessed as the intersection between the line representing the thermal signature and the x-axis). If during the day the minimal outdoor temperature is higher than the temperature reference, the HDD are equal to 0. Indeed, in this case the outdoor temperature is always higher than the set point indoor temperature, no space heating is needed. For the other case the HHD are calculated as:

$$HDD = T_{ref} - \left(\frac{T_{max} + T_{min}}{2}\right)$$

Where T_{max} is the maximum outdoor temperature of the day and T_{min} is the minimum outdoor temperature of the day.

The thermal signature method can also be used to follow the behavior of buildings through the days and potential dysfunctions. For example, in Figure 4, few points deviate from the reference operation. This could be interpreted as measurement errors or data encoding errors underlying a sensor malfunction.

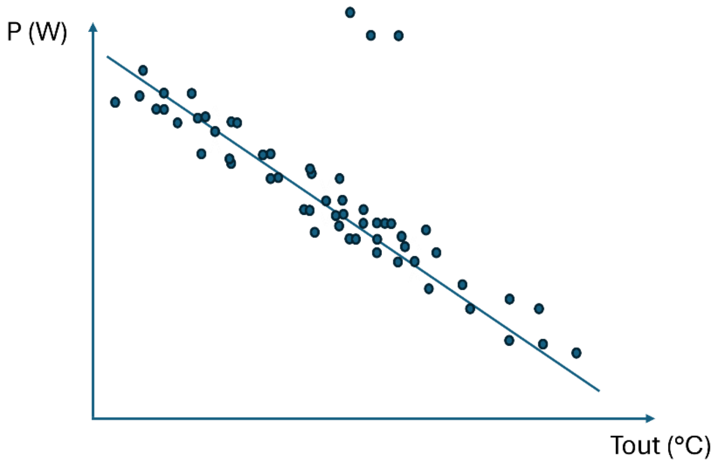


Figure 4 Building thermal signature with deviating points

A wide spread of measurement points (see Figure 5) shows that a potential problem occurs in the control systems (valves, sensors or controllers).

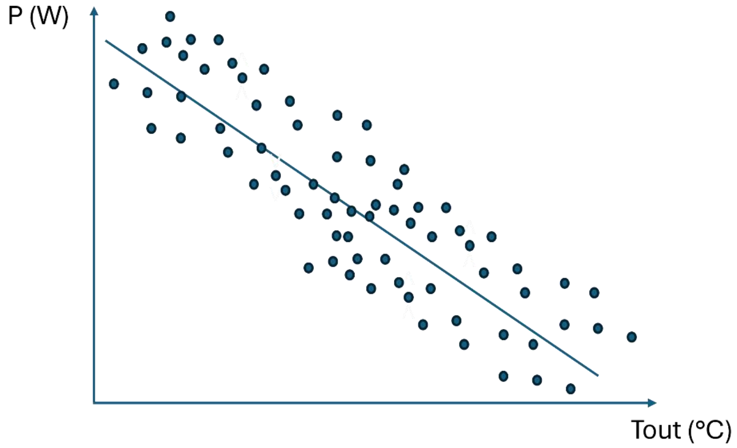


Figure 5 Building thermal signature with widespread points

A progressive drift of measurement points from the reference operation (see Figure 6) stresses a change in the building’s behavior such as a heat supply problem, a leaky building envelope or overheating potentially due to a change in the setpoint indoor temperature.

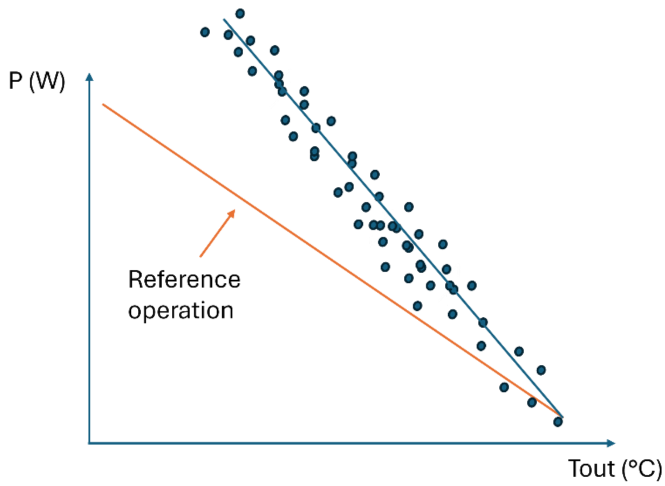


Figure 6 Building thermal signature with a progressive drift

A thermal signature with two separate behaviors (see Figure 7) is the specific sign of a time clock control in the building. A time clock operation consists in modifying the indoor set point temperature during some part of the day (usually when the building's occupation changes). This type of control is often seen in tertiary buildings.

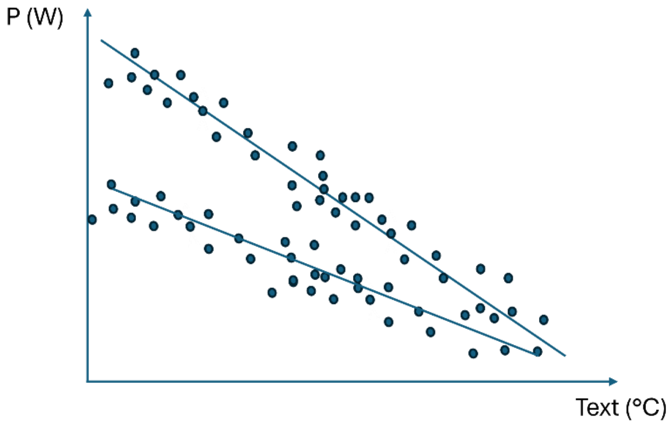


Figure 7 Building thermal signature with two separate behaviors

b. European standard

Consumption data for new buildings and some existing ones are not always available. Other methods must therefore be used to determine the space heating needs.

The space heating design power $\dot{Q}_{sh}$ (W) can be calculated through the European standard EN-12831 [1]. The space heating demand depends on 4 items:

- The transmission losses $\dot{Q}_t$ (W) through the walls, the windows, the ground, etc.
- The ventilation losses $\dot{Q}_v$ (W) through the infiltrations and the mechanical ventilation
- The solar gains $\dot{Q}_s$ (W)
- The internal gains $\dot{Q}_i$ (W) through the lighting, the occupants and electric appliances

$$\dot{Q}_{sh} = \dot{Q}_t + \dot{Q}_v - [\dot{Q}_s + \dot{Q}_i]$$

The European standard aims to determine the maximum space heating power needed in the worst-case scenario. Hence, the solar and internal gains will be set to 0. Therefore, we only need to calculate the transmission and ventilation losses.

i. The transmission losses

For a building where the indoor set point temperature is supposed to be constant, the transmission thermal transfer coefficient $U_{t,tot}$ (W/K) is defined as the ratio between the space heating power and the outdoor indoor temperature difference. The aim of the European standard is to calculate this coefficient thanks to the design data of the buildings.

$$\dot{Q}_t = U_{t,tot}\,(T_{in} - T_{out})$$

The transmission thermal transfer coefficient is the sum of the thermal transfer coefficient of the losses through the wall, the windows, the ground, the non-heated spaces and the thermal bridges (see Figure 8).

$$U_{t,tot} = \sum_n U_{t,n}\, A\, f_n + \sum_i \psi_i L_i + \sum_k \chi_k$$

Where $U_{t,n}$ is the thermal transfer coefficient through a surface (W/K/m²), A the area (m²), f_n a corrective coefficient, ψ (W/m.K) the linear thermal bridge coefficient, L (m) the thermal bridge length and χ (W/K) the punctual thermal bridge coefficient.

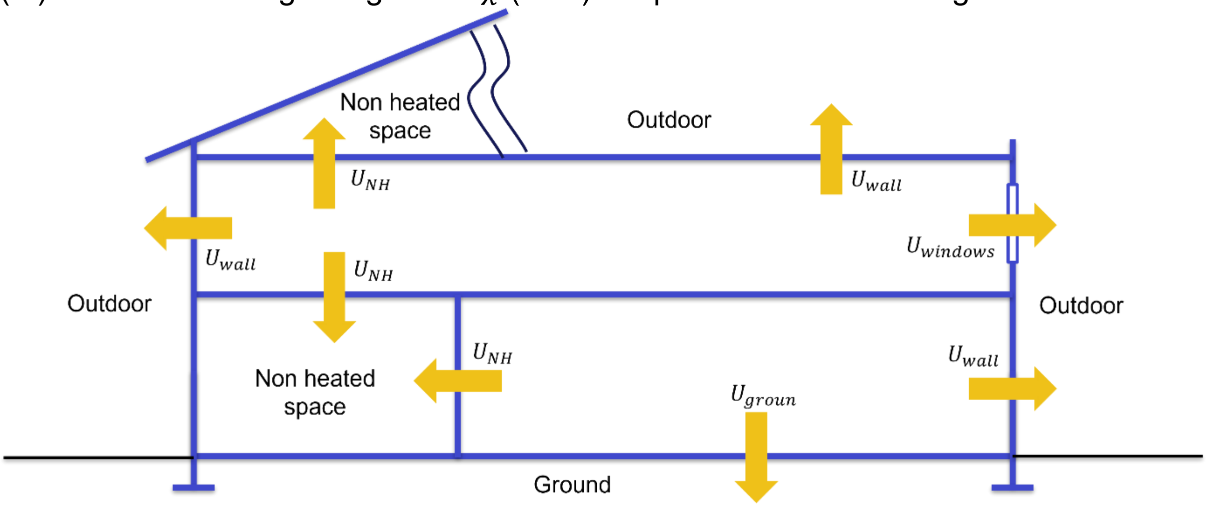


Figure 8 Scheme of the possible transmission losses in a building

1. Transmission through an opaque wall

The transmission through an opaque wall between indoor and outdoor can be determined through the wall composition and the thermoelectric analogy in 1D. The thermal resistance of the opaque wall R_{wall} (the inverse of thermal transfer coefficient U_{wall}) is the sum of the thermal conduction resistances of each layer of the wall and the indoor/outdoor convective thermal resistances (see Figure 9)

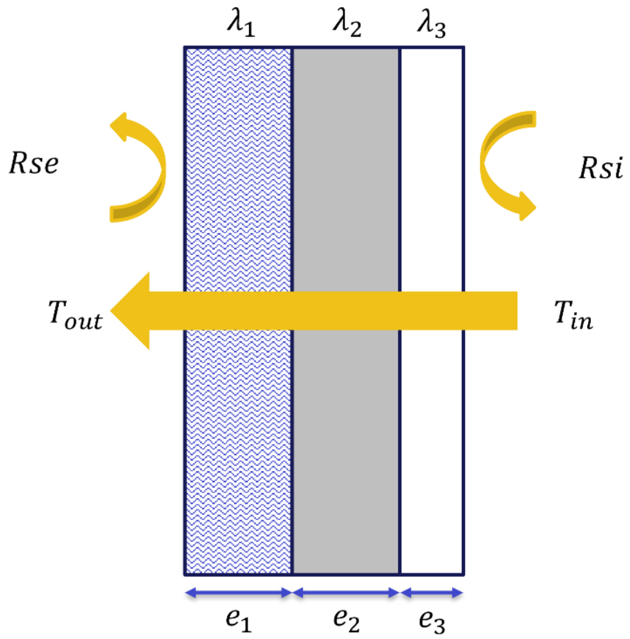


Figure 9 Thermal losses through an opaque wall

The conductive thermal resistance $R_{wall,cond}$ (K/W) can be expressed as:

$$R_{wall,cond} = \frac{e}{\lambda S}$$

Where e (m) is the layer thickness, S (m²) the layer surface and λ (W/m.K) the thermal conductivity.

The indoor and outdoor convective thermal resistances (Rsi/Rse) are defined by the European standard depending on the heat flow direction (see Table 1)

Wall type and flow direction	Rsi	Rse	$Rsi + Rse$
Horizontal flow (>60°) (wall)	0.13	0.04	0.17
Upward flow (roof)	0.1	0.04	0.14
Downward flow (floor)	0.17	0.04	0.21

Table 1 Convective thermal resistance (m².K/W)

Hence the global thermal resistance of an opaque wall is:

$$R_{wall} = \frac{1}{S}\left(Rsi + \sum_i \frac{e_i}{\lambda_i} + Rse\right)$$

$$U_{wall} = \frac{1}{R_{wall}} = \left[\frac{1}{S}\left(Rsi + \sum_i \frac{e_i}{\lambda_i} + Rse\right)\right]^{-1}$$

For the other types of wall, the thermal transfer coefficient can be determined in the same way:

$$U_{roof} = \frac{1}{R_{roof}} = \left[\frac{1}{S}\left(Rsi + \sum_i \frac{e_i}{\lambda_i} + Rse\right)\right]^{-1}$$

$$U_{floor} = \frac{1}{R_{floor}} = \left[\frac{1}{S}\left(Rsi + \sum_i \frac{e_i}{\lambda_i}\right)\right]^{-1} \quad U_{NH} = \frac{1}{R_{NH}} = \left[\frac{1}{S}\left(Rsi + Rsi + \sum_i \frac{e_i}{\lambda_i}\right)\right]^{-1}$$

Where NH refers to a non-heated room.

The heat losses can be expressed as:

$$\dot{Q}_t = [U_{wall} + U_{roof} + U_{floor} * f_{floor} + U_{NH} * f_{NH}]\,(T_{in} - T_{out})$$

With f_{floor} and f_{NH} the corrective coefficients considering the ground and non-heated space temperatures:

$$f_{floor} = \frac{T_{in} - T_{ground}}{T_{in} - T_{out}}$$

$$f_{NH} = \frac{T_{in} - T_{NH}}{T_{in} - T_{out}} + \frac{T_{in} - T_{in}^{*}}{T_{in} - T_{out}}$$

Where T_{ground}, T_{NH} and T_{in}^{*} the temperatures of the ground, the non-heated space and the mean temperature of the inside wall respectively.

2. Transmission through windows

The transmission through a window between indoor and outdoor can be determined through the window composition and the thermoelectric analogy in 1D. The thermal transfer coefficient of the windows U_w is the sum of the thermal transfer coefficient of the glass U_g and the frame U_f weighted by their surface.

$$U_w = \frac{U_g S_g + U_f S_f}{S_g + S_f}$$

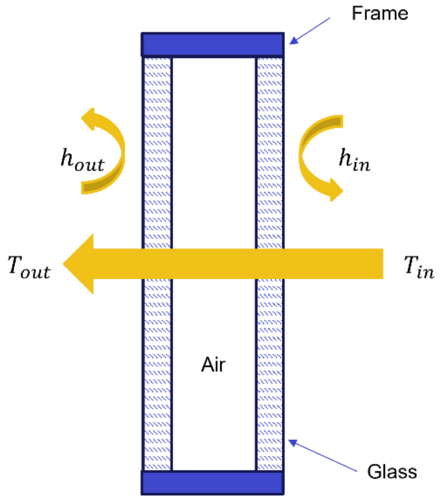


Figure 10 Thermal losses through a window

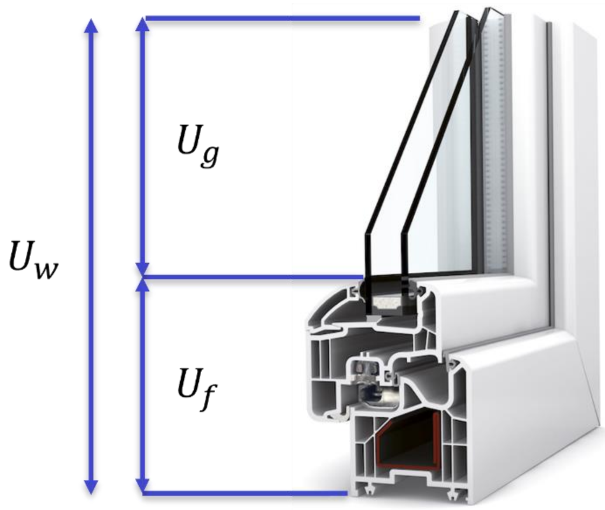


Figure 11 Cross-section of a window

Hence the global thermal resistance of a window is:

$$R_{w,tot} = \frac{1}{S_w}\left(Rsi + \frac{1}{U_w} + Rse\right)$$

$$U_{w,tot} = \frac{1}{R_{w,tot}} = \left[\frac{1}{S}\left(Rsi + \frac{1}{U_w} + Rse\right)\right]^{-1}$$

The heat losses can be expressed as:

$$\dot{Q}_t = [U_{wall} + U_{roof} + U_{floor} * f_{floor} + U_{NH} * f_{NH} + U_{w,tot}]\,(T_{in} - T_{out})$$

3. Thermal bridges

A thermal bridge is a punctual or linear zone in the envelope of a building that presents a variation in thermal resistance (see Figure 12) such as the connections between joinery and an opaque wall, intermediate floors, partition walls, door thresholds, …

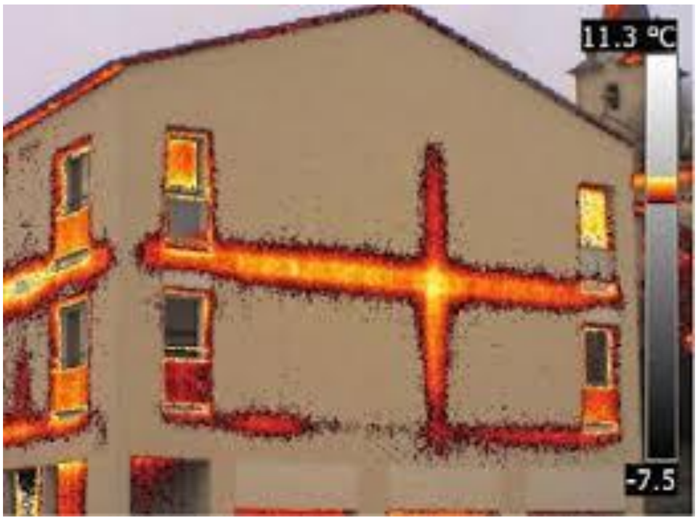


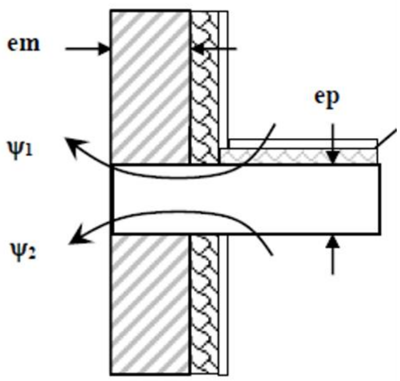
Figure 12 Example of thermal bridges measured by a thermal camera [2]

The thermal effect of a thermal bridge depends on the material, the geometry, the insulation and the type of wall. The heat losses due to thermal bridges $\dot{Q}_{tb}$ (W) can be expressed as:

$$\dot{Q}_{tb} = U_{tb}\,(T_{in} - T_{out})$$
$$U_{tb} = \sum_i \psi_i L_i + \sum_k \chi_k$$

With U_{tb} (W/K) the thermal bridges heat transfer coefficient, ψ (W/m.K) the linear thermal bridge coefficient, L (m) the thermal bridge length and χ (W/K) the punctual thermal bridge coefficient. The values of ψ and χ are determined through numerical simulation. The catalogue of the linear and punctual thermal bridge coefficients can be found in the European standard [1].

Figure 13 shows an example.



Distribution : $\psi_1 = \psi_2 = 50\%\,\psi$

$e_m\ (cm)$	$e_p\ (cm)$		
	15	20	25
20 < e_m< 25	0.67	0.82	0.96
25 < e_m< 30	0.63	0.77	0.90

Figure 13 Thermal bridge coefficients for a connection between an intermediate floor and a wall

ii. Ventilation losses

For a building where the indoor set point temperature is supposed to be constant, the heat losses through the ventilation $\dot{Q}_v\ (W)$ depend on the outdoor indoor temperature difference and the airflow rate q_v (m³/h).

$$\dot{Q}_v = 0.34\ q_v\ (T_{in} - T_{out})$$

The ventilation flow rate is due to:

- The mechanical ventilation imposing regulatory fresh air renewal q_{mv} (m³/h)
- The infiltrations due to leaks in the building envelope q_{inf} (m³/h)
- The natural ventilation due to the wind and thermal draught q_{nv} (m³/h)

$$q_v = q_{mv} + q_{inf} + q_{nv}$$

Figure 14 shows a schematic view of the ventilation heat losses in a building. Airflow rates caused by natural ventilation are related to the occasional opening of doors and windows. They are usually not considered in the heating load calculation.

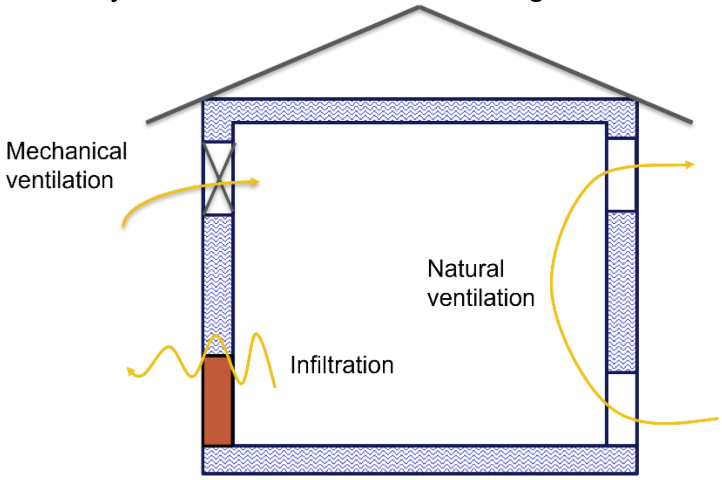


Figure 14 Schematic view of the heat losses caused by the ventilation

1. Mechanical ventilation

In residential buildings, the regulation sets levels of airflow rate for self-regulating systems. The regulation depends on national standards. For example, in France the minimum fresh airflow rate depends on the volume of the buildings or the number of rooms in the buildings. Table 2 gives the French national values for fresh airflow rate.

Table 2 Minimum fresh air flow rate according to the number of rooms in a building in France [3]

Number of rooms	1	2	3	4	5	6	7
Minimum fresh air flow rate (m³/h)	35	60	75	90	105	120	135

2. Air infiltration

A blower door test can be used to measure the air permeability of new buildings. The blower door consists of a fan and a watertight sheet held in place by a frame that fits over the entrance door frame of the building under the test. When the fan is running, the device simultaneously measures the volume flowrate passing through the fan (and therefore through the airtightness defects), and the pressure difference between the inside and outside the building $\Delta p = |p_{in} - p_{ext}|$ (Pa). After a test and several measurement points, the graph is plotted, showing the coefficients K and n of a relationship between two quantities of the form:

$$q_{inf} = K\ \Delta p^n$$

With n a coefficient of which the value is between 0.5 and 1. Based on this equation, two indicators can be calculated:

- $Q_{4Pa,surf}$ (m³/h/m²) the volume flow rate for Δp=4 Pa, divided by the heat loss area of the building
- n_{50} (vol/h) the air renewal rate obtained for Δp=50 Pa

The French thermal regulation RE2020 imposed maximal infiltration based on his two indicators (see Table 3)

Table 3 Maximal infiltration flow rate according to French regulation RE2020

Type of buildings	$Q_{4Pa,surf}$ maximal (m³/h/m²)	n_{50} maximal (vol/h)
Individual house	0.6	2.3
Multi-family house	1	3.9
Tertiary building	1.6	6.2

c. Fast archetype method

For existing buildings, heating consumption data, geometric characteristics and composition of the walls are not always known. To overcome this problem, databases of archetypes on a national scale have been set up to enable comparisons between countries. This was notably the case with the European Tabula and Episcope projects [4] carried out between 2009 and 2016. Their aim is to characterize Europe's residential buildings and their energy consumption to provide practical support for large-scale energy renovation projects.
This program has established, for the thirteen participating European countries, a definition methodology and then a set of building types representative of the buildings encountered in the residential stock. The types are classified by class and by year of construction, according to the specific characteristics of each country. The Tabula project offers a simple way of reading the physical properties of buildings to obtain orders of magnitude of energy demand. A decision-support software package called TABULA webtool has resulted from this project, including national typologies for the 13 participating countries.
For France, the choice was made to use 10 construction periods and 4 classes of dwelling: detached house, semi-detached house, collective building with fewer than 10 dwellings and collective building with 10 or more dwellings (Figure 15).

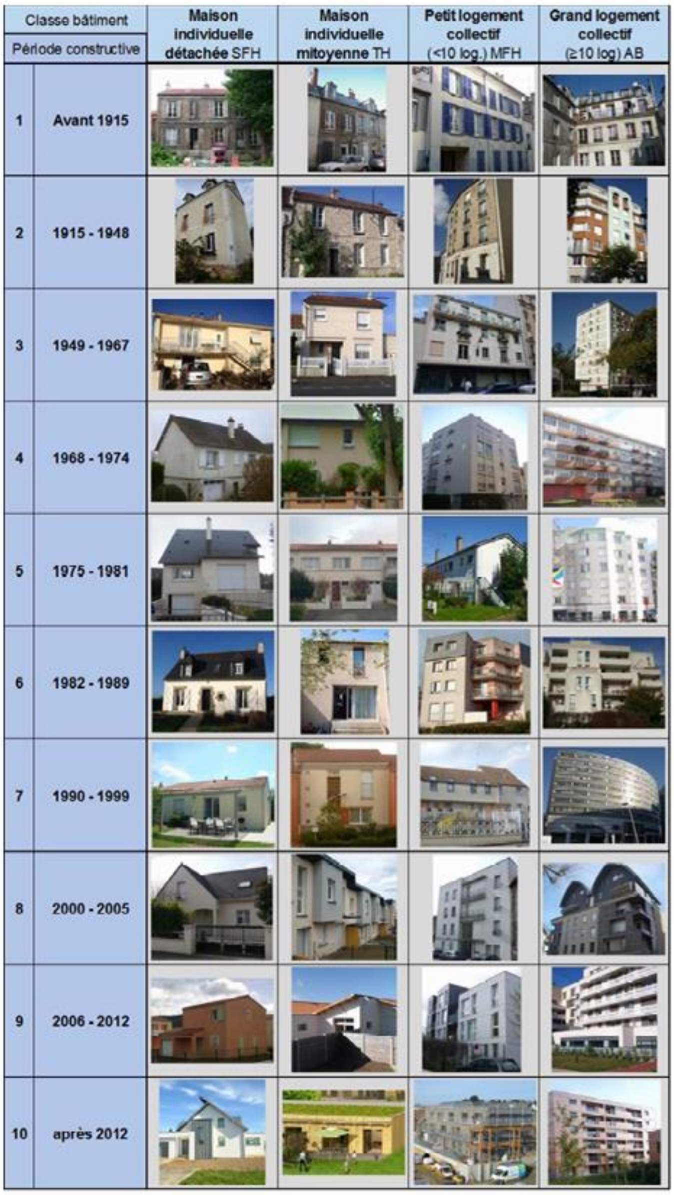


Figure 15 French building archetype according to Episcope/tabula project [5]

For Hungary, the choice was made to use the 5 construction periods and 4 classes of dwelling: small individual house (<80 m²), big individual house (>80 m²), multi-family house between 4 and 9 flats, and multi-family house with more than 10 flats (see Figure 16)

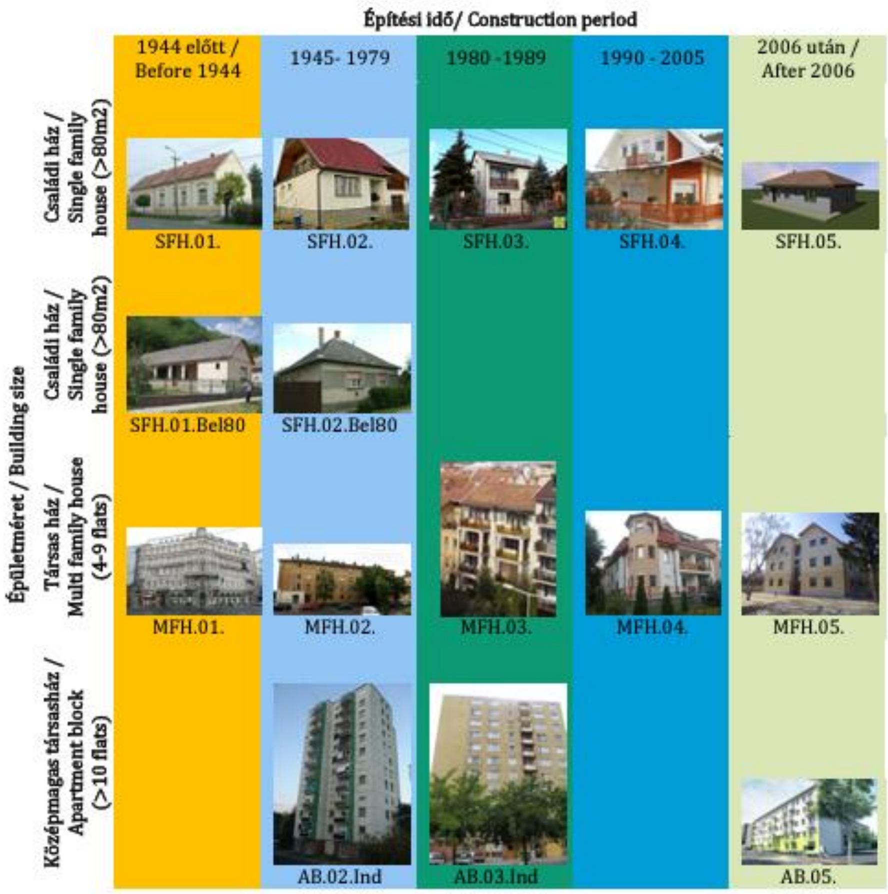


Figure 16 Hungarian building archetype according to Episcope/tabula project [6]

Hence for each archetype in each country, the compositions and thermal characteristics of the walls, windows, roof, … are detailed such as the energy consumption for different levels of envelope refurbishment.

2. Space cooling demand

The first step is to determine how to calculate the space cooling demand for a building. The method described here is inspired by [7].
The space cooling design power $\dot{Q}_{sc}$ (W) depends on 4 items:

- The transmission gains $\dot{Q}_w$ (W) through the opaque surfaces
- The ventilation gains $\dot{Q}_v$ (W) though the infiltration and the mechanical ventilation
- The transmission gains $\dot{Q}_s$ (W) through transparent surfaces
- The internal gains $\dot{Q}_i$ (W) through the lighting, the occupants and electric appliances

$$\dot{Q}_{sc} = \dot{Q}_w + \dot{Q}_v + \dot{Q}_s + \dot{Q}_i$$

The principal difficulty in calculating the space cooling demands is the impact of solar gains. As for the calculus of space heating demand, the space cooling demand must be calculated for the worst-case scenario i.e. when the solar gains are the highest.

a. Transmission through opaque walls

The thermal gains through opaque walls are calculated in the same way as the losses for the space heating demand:

$$\dot{Q}_w = U_{wall} S \,\Delta T$$

Nevertheless, the thermal transfer coefficient of the opaque wall U_{wall} is different to the one used for space heating. Indeed, the internal convection is not the same and the solar gain must be considered. Hence the thermal gain through an opaque wall is [7]:

$$\dot{Q}_w = U_{wall} S \left(T_{eq} - T_{in}\right)$$

With

$$U_{wall} = \frac{1}{R_{wall}} = \left[\left(Rsi + \sum_i \frac{e_i}{\lambda_i}\right)\right]^{-1} \quad with\ Rsi = 0.25\ [m^2.K/W]$$

$$T_{eq} = T_{out} + Rse\ \alpha\ E$$

With T_{eq} (K) the equivalent temperature, E (W/m²) the solar irradiance, α the wall absorptivity. For summer conditions, Rse is equal to 0.06 for vertical surfaces and 0.05 for horizontal surfaces. If the data about the absorptivity factor of the wall is missing, the following values can be used:

- $\alpha = 0.5$ for light-colored walls
- $\alpha = 0.9$ for dark walls
- $\alpha = 0.7$ for medium-colored walls

The solar irradiance depends on the date, the daytime, the localization and the orientation of the walls (see Figure 17 & Figure 18). To calculate the Cooling power needed, the maximum solar irradiance must be choose.

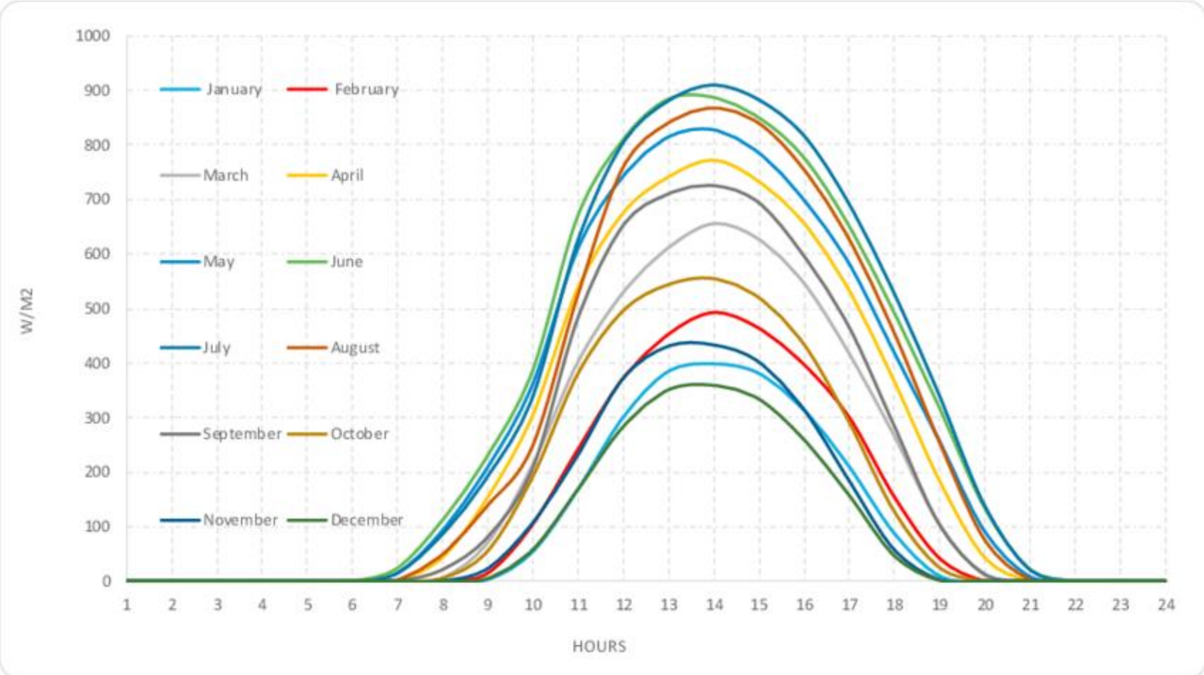


Figure 17 Evolution of solar irradiance according to the date and the daytime

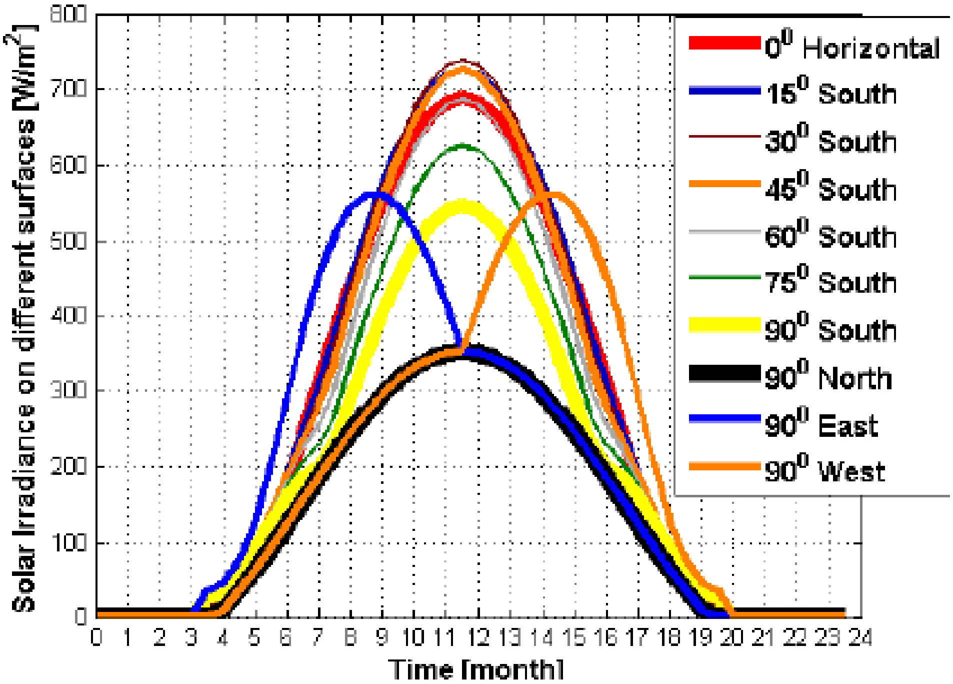


Figure 18 Evolution of solar irradiance according to the orientation

b. Transmission through transparent walls.

The thermal gains through the transparent surfaces are calculated in the way as for the opaque wall. Nevertheless, the solar factor of the windows glass g must be considered. Moreover, the solar irradiance can be reduced if solar shades are present above the window.

$$\dot{Q}_s = g\ S\ E$$

c. Ventilation gains

The ventilation gains are calculated in the same way as the ventilation losses for space heating needs (see paragraph 2.ii)

$$\dot{Q}_v = 0.34\ q_v\ (T_{in} - T_{out})$$

d. Internal gains

The internal gain stands for all the heat inflow due to the occupants, the lighting and electric appliances. For a residential building, the total internal gains can be determined according to the number of occupants in the buildings as shown in Figure 19.

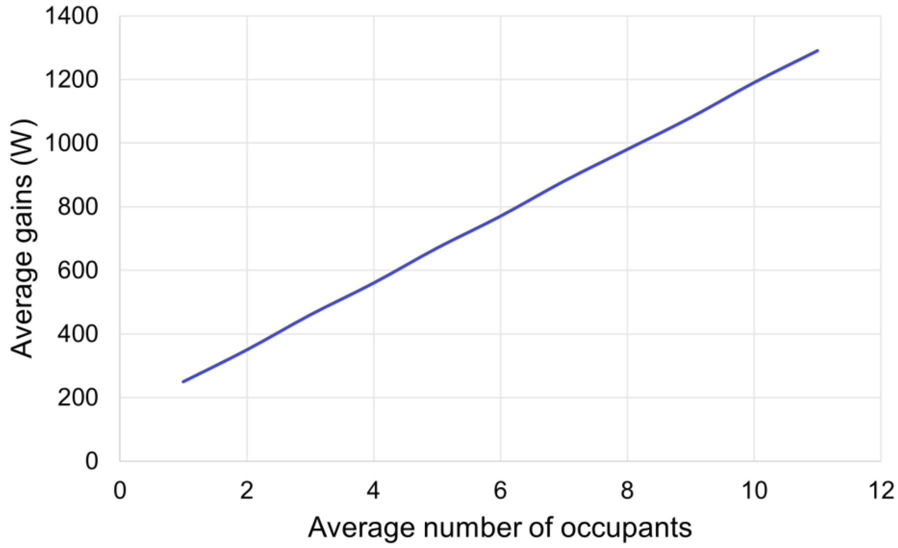


Figure 19 Total internal gains for a residential building according to the number of occupants [7]

3. Domestic hot water demand

A district heating network does not only supply space heating but also domestic hot water (DHW). As space heating demands, the DHW demand can be determined by calculus. The DHW supply must fulfill 2 principal conditions: comfort and health. Indeed, the DHW supply must respect the demands in terms of daily volume, instantaneous flow rate and temperature. Also, the DHW produced must respect sanitary conditions such as avoiding the spread of legionella. Legionella is a deadly bacteria growing in stagnant water between 20°C and 55°C. Hence, the DHW produced must be at least 55°C or 60°C according to the standard and the DHW must always be in motion.
This chapter aims to design DHW production and needs with respect to the constraints detailed above.

a. Domestic hot water consumption

The DHW supply must be at a stable temperature and be immediately available at a sufficient flow rate. DHW consumption is characterized by highly variable volumes drawn and instantaneous flow rate depending on the times and the uses. Nevertheless, DHW consumptions are all labelled by low or zero consumption period followed by one or several periods of varying degrees of consumption. Figure 20 presents examples of DHW consumption patterns for different types of buildings.

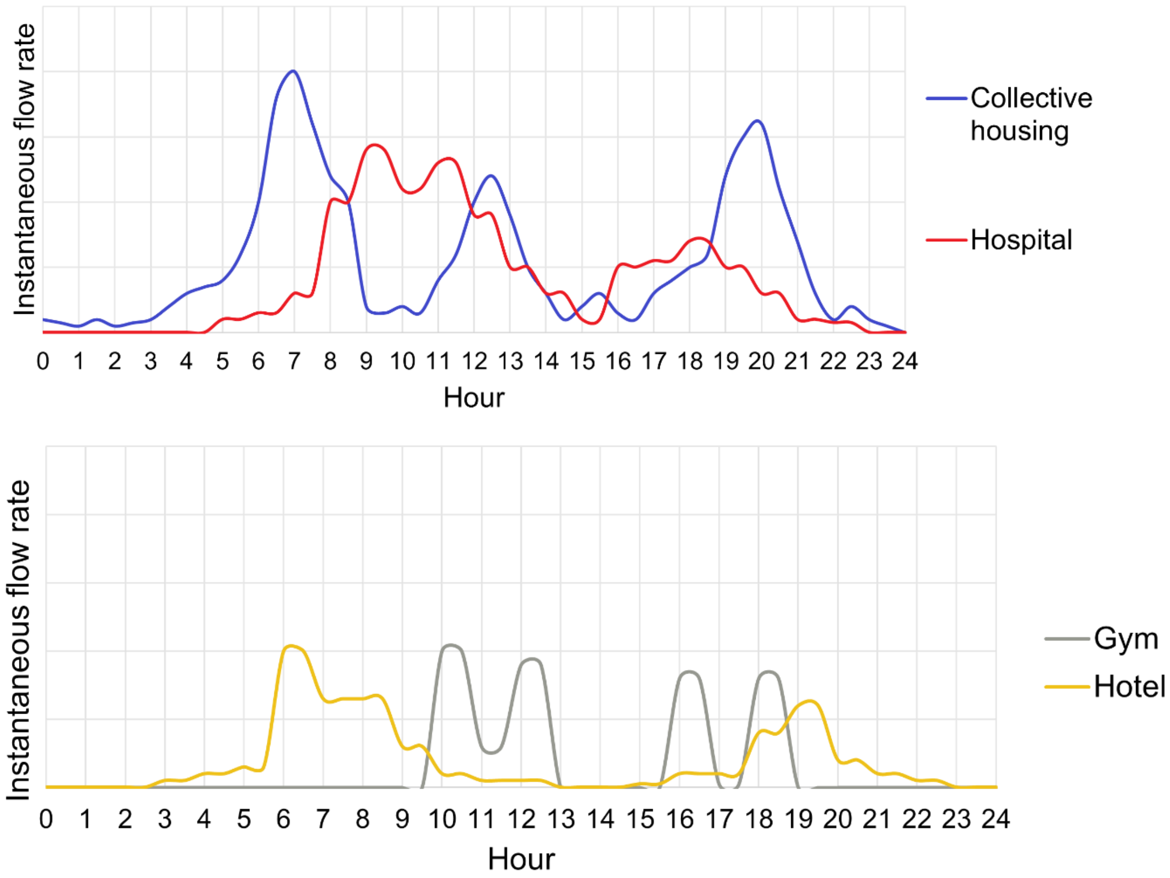


Figure 20 Example of DHW consumption profile for different types of buildings [8]

These figures show that all the patterns are different in terms of daily volume consumed and instantaneous flow rate needed. Moreover, they are characterized by peak and low demand consumption periods but for different hours in a day. Hence the consumption patterns depend on the type of buildings and more specifically on the number and type of equipment (sink, shower bath, dishwasher, …) and their usage.

To size the DHW supply for a building the daily volume consumption pattern can be resumed as 3 separate periods as shown in Figure 21.

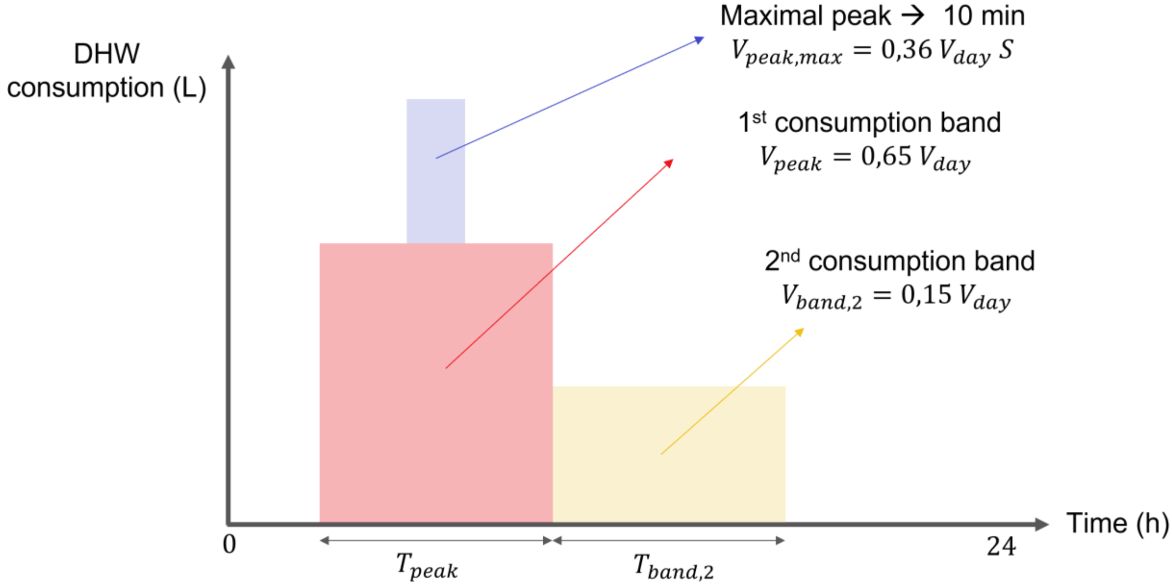


Figure 21 Normalized DHW consumption profile

The maximal peak period is the period where the instantaneous flow rate is higher. This flow rate corresponds to the one used to design the DHW network piping in the building. The maximun peak can last between 2 and 25 min depending on the type of building or the number of dwellings in residential buildings (4.32 min for 10 dwellings, 12.5 min for 100 dwellings, 18.27 min for 200 dwellings). To simplify, for all cases the duration of maximal peak consumption $T_{peak,max}$ is set to 10 min, as it is the duration chosen to size the DHW supply equipment. The DHW volume consumed during the maximal peak $V_{peak,max}$ corresponds to 36% of the daily use V_{day} for a residential building (39% for hospitals, hostels, offices) corrected by a simultaneity factor S.

$$V_{peak,max} = \begin{cases} 0.36 * V_{day} * S & for\ residental\ buiding \\ 0.39 * V_{day} * S & for\ non\ residental\ buiding \end{cases}$$

The simultaneity factor is used to model the probability of all the DHW equipment to be used at the same time. The greater the number of equipment is the lower the probability that all equipment is used simultaneously. The mathematical definition of this factor will be given in the next section. In a same way the maximal instantaneous flow rate can be calculated in L/h:

$$\dot{Q}_{peak} = \begin{cases} 0.36 * V_{day} * S * 6 & for\ residental\ buiding \\ 0.39 * V_{day} * S * 6 & for\ non\ residental\ buiding \end{cases}$$

The second consumption period is the peak one. It takes place during a span T_{peak} calculated according to the amount of equipment in the buildings. The definition of T_{peak} will be presented later. The DHW volume consumed during this period V_{peak} corresponds to 65% of the daily consumption for residential buildings and 78% for other buildings.

$$V_{peak} = \begin{cases} 0.65 * V_{day} & for\ residental\ buiding \\ 0.78 * V_{day} & for\ non\ residental\ buiding \end{cases}$$

The last consumption period corresponds to 15% of the daily consumption for all the cases.

Daily DHW consumption can be determined by knowing all the equipment in the buildings. Table 4 presents the mean daily consumption of classical equipment.

Table 4 DHW daily consumption of several equipment [8]

Equipment	Unit	Daily consumption (L)	Instantaneous flow rate (L/min)
Individual sink	1 hot water tap	40	6
Collective sink	1 hot water tap	120-240	3-6
Bathtub	1 bathtub	120	20
Individual shower	1 shower	80	15
Collective shower	1 shower	300	10
Kitchen sink	1 hot water tap	60	12
Dishwasher	1 meal	4	10
Individual washer	1 washer	50	25
Collective washer	1washer	200-340	25

Hence the daily DHW consumption is the sum of the one of all the equipment present in the building. Nevertheless, sometimes all the equipment is not known, to overcome this hurdle, a module can be determined. A module is a set of drawing points (equipment) likely to be used regularly by a group of identifiable people, examples are given in Table 5.

Table 5 Example of DHW module [8]

Type of buildings	Module	Example	Description	Daily consumption (L)
Residential building	1 Dwelling	1 room flat	1 shower 1 kitchen sink	95
		3 room flat	1 Bathtub 1 kitchen sink 1 Individual sink	180
Hotel	1 Room	Hotel *	1 Shower	90
		Hotel ***	Bathtub	160
Office	10 employees		1 Collective sink	40
Hospital	1 Bed		1 Individual sink 0.1 Bathtub 0.5 Collective sink	112

Hence the daily DHW consumption of the buildings is the sum of the module daily consumption $V_{day,module}$ by the number of modules N:

$$V_{day} = V_{day,module} * N$$

To not overestimate DHW consumption a simultaneity factor S must be calculated. This factor depends on the number of modules (i.e. the number of equipment or drawing points) described by the following equation:

$$S = \begin{cases} \dfrac{1}{(N-1)^{0.3*i}} & for\ residental\ buiding \\ \dfrac{1}{(N-1)^{0.2*i}} & for\ non\ residental\ buiding \end{cases} \quad with\ \ i = \frac{N-1}{N+1}$$

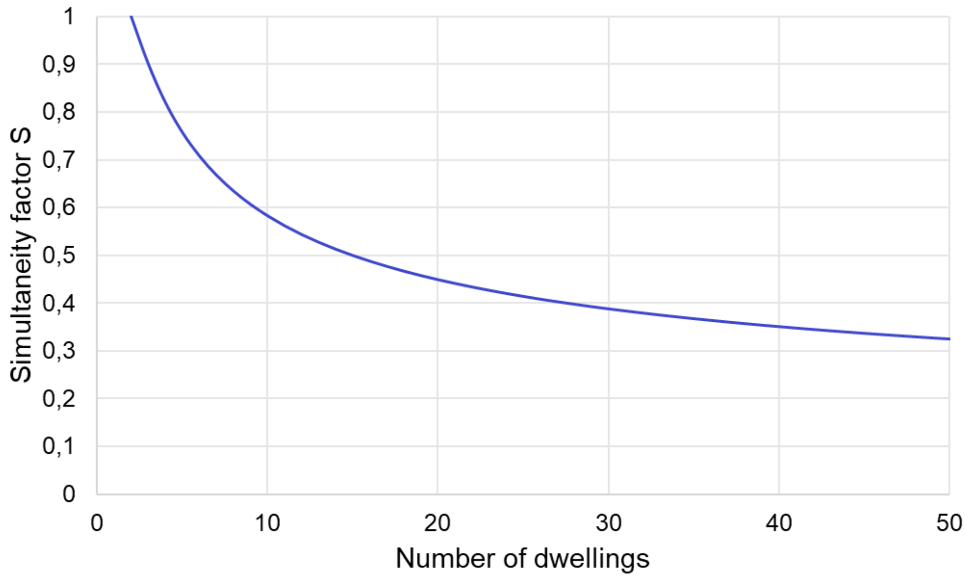


Figure 22 Evolution of the simultaneity factor according to the number of dwellings

The evolution of the simultaneity factor according to the number of dwellings in a residential building is presented in Figure 22.
In a same way the peak duration time T_{peak} (h) can also be calculated as follows:

$$T_{peak} = \begin{cases} \dfrac{N^{0.889}}{(N+1)^{0.65}} & for\ residental\ buiding \\ \dfrac{N^{0.878}}{(N+1)^{0.70}} & for\ non\ residental\ buiding \end{cases}$$

Hence, as shown in Figure 23 the peak duration time rises in respect to the number of modules.

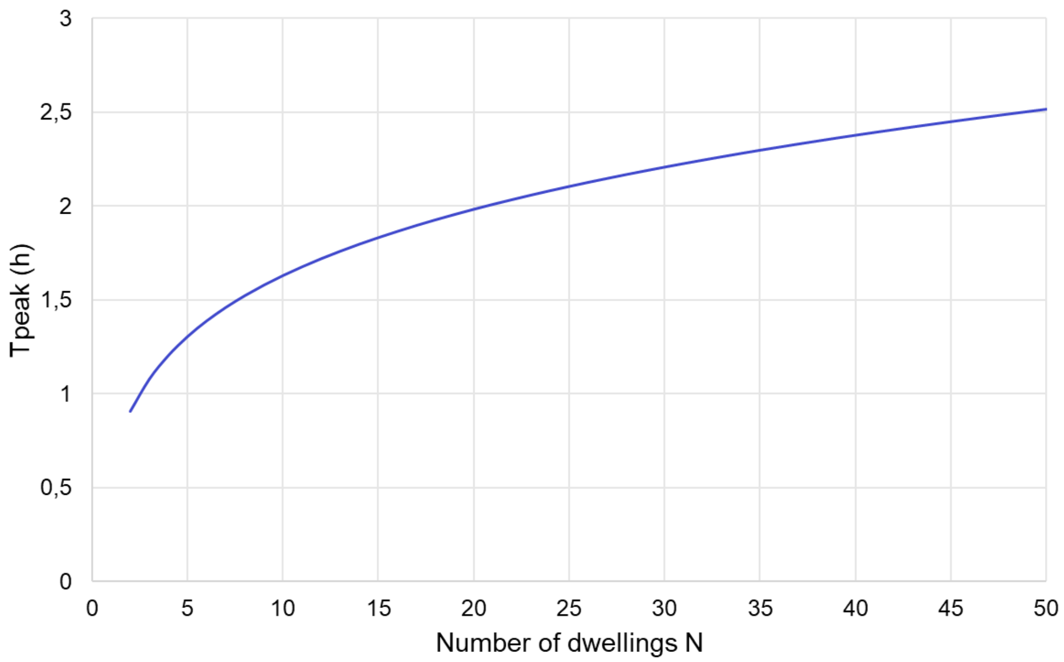


Figure 23 Evolution of the peak duration time according to the number of dwellings

These formulas (simultaneity factor and peak duration time) are not accurate for:

- Sports facility $T_{peak} = 30\ min$ max and $S = 1$

- Hostel and hospital $T_{peak} = 2\ h$ max and $S = 0.25$
- Boarding school $T_{peak} = 1\ h$ max and $S = 1$

b. Domestic hot water supply

Once the maximum instantaneous flow rate and the daily volume consumption of DHW are determined, the sizing and design of the supply can be handled. The DHW energy consumption E_{DHW} (kWh) is expressed as:

$$E_{DHW} = V_{day}\ Cp\ (T_{DHW} - T_{CW})$$

With Cp the water specific heat (kWh/m^3.K), T_{DHW} the DHW supply temperature (°C) (i.e. between 55°C and 60°C) and T_{CW} the cold-water temperature coming for the water networks. The cold-water temperature evolves through time, Figure 24 presents the evolution of cold-water temperature in France.

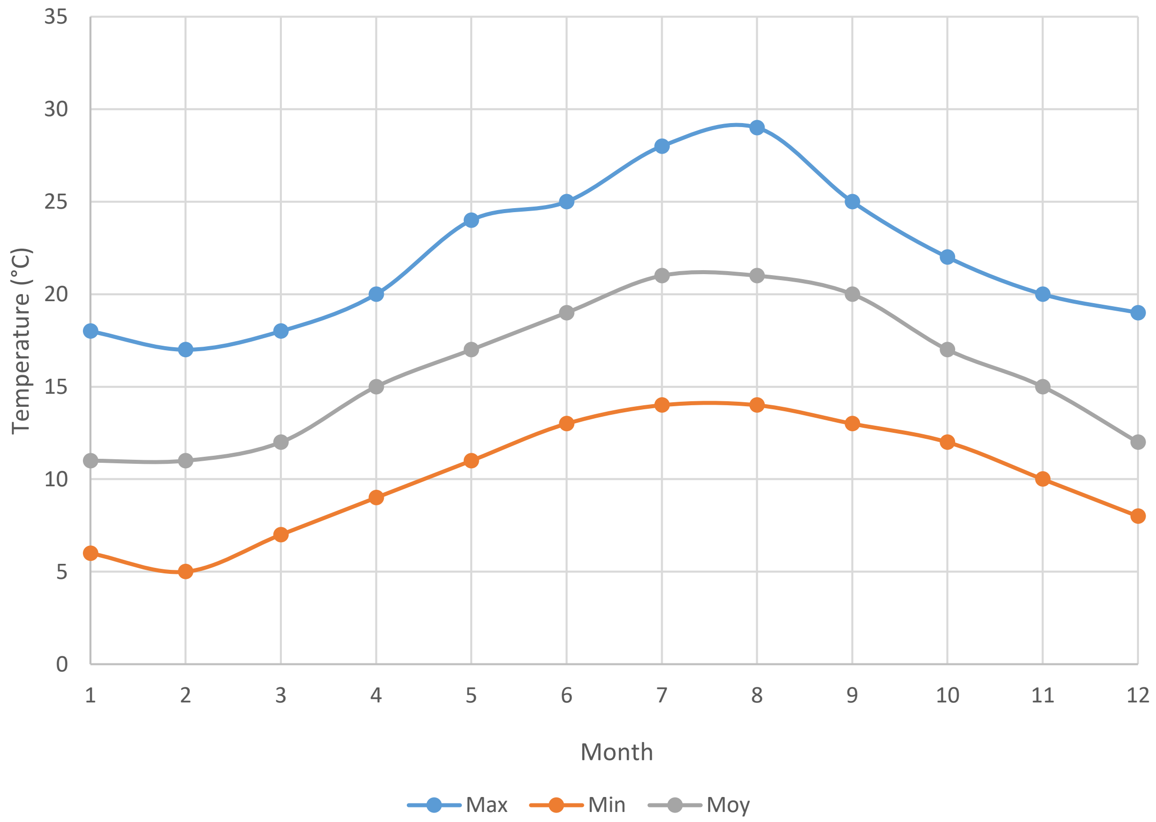


Figure 24 Evolution of the cold-water temperature through the year in French climates (Paris) [8]

Next, the design power of the heat exchanger supplies the DHW must be set. This value depends on the hydraulic scheme of the DHW supply. Indeed 4 ways to supply DHW are used and will be described in the following.

i. Instantaneous supply

The instantaneous supply of DHW consists of supplying DHW with only a heat exchanger between the DHW network piping and the heat production (the primary network for a district heating network) without any storage tank (see Figure 25).

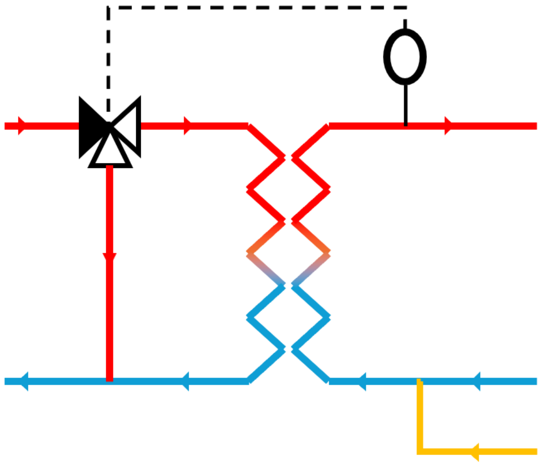


Figure 25 Scheme of an instantaneous supply of DHW

Hence, the heat exchanger is sized to be able to heat cold water at 10°C to 55 or 60°C every time specifically during the maximum daily consumption peak. Consequently, the design power $\dot{Q}$ can be calculated as:

$$\dot{Q} = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{peak,max}}{T_{peak,max}}$$

With $T_{peak,max}$ the duration time of the maximal consumption peak equal to 10 minutes. This DHW supply technique has the advantage of taking up very little space and being inexpensive. But the design power is important and often very superior to the power required to cover the space heating needs of the buildings.

ii. Semi-instantaneous supply

The semi-instantaneous supply consists of using a heat exchanger coupled with a storage tank able to cover totally or partially the DHW volume needs during the maximal consumption peak. A hydraulic scheme is presented in Figure 26.

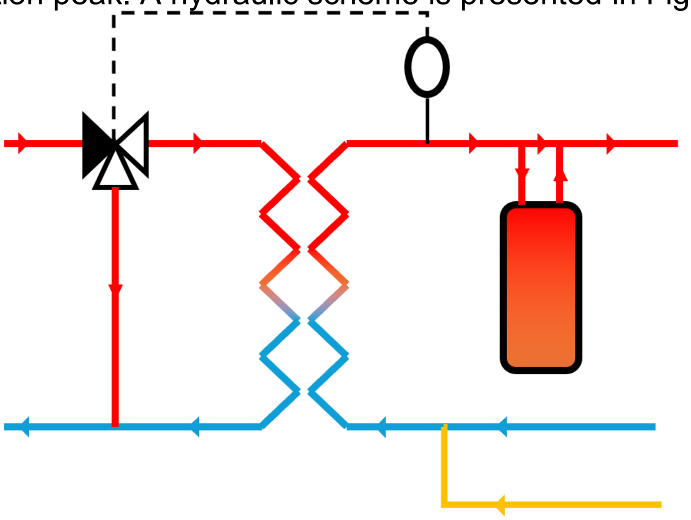


Figure 26 Scheme of a semi-instantaneous supply of DHW

The design power of the semi-instantaneous heat exchanger depends on the size of the storage tank. The design power $\dot{Q}_{SI}$ (W) and the minimum one $\dot{Q}_{SI,min}$ can be expressed as:

$$\dot{Q}_{SI,min} = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{peak} - V_{peak,max}}{T_{peak}}$$

$$\dot{Q}_{SI} = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{peak} - Ca}{T_{peak}}$$

As a result, the volume of the storage tank Ca ranges from 0 to $V_{peak,max}$. This DHW supply technique has the advantage of lowering the required power at the heat exchanger. But the space requirement is higher and it is most costly.

iii. Semi-accumulation supply

The semi-accumulation supply consists of using a heat exchanger coupled with a storage tank able to cover totally or partially the DHW volume needs during the consumption peak and at least the DHW volume needs during the maximal consumption peak. A hydraulic scheme is presented in Figure 27.

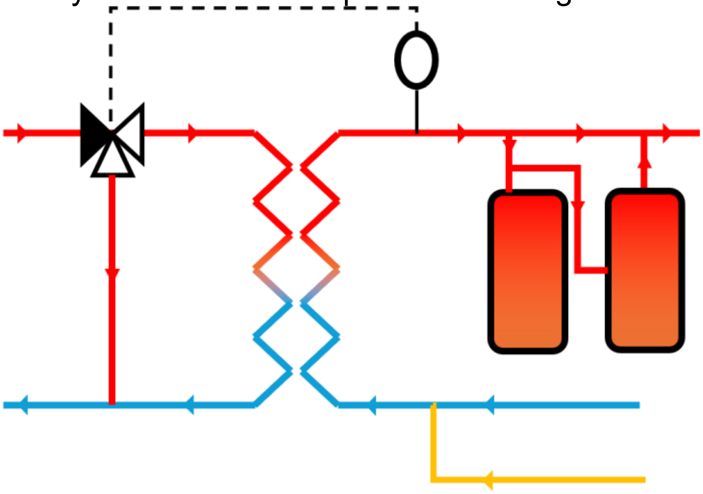


Figure 27 Scheme of a semi-accumulation supply of DHW

The design power of the semi-accumulation heat exchanger depends on the size of the storage tank. The design power $\dot{Q}_{SA}$ (W), the minimum one $\dot{Q}_{SA,min}$ and the maximum one $\dot{Q}_{SA,max}$ can be expressed as:

$$\dot{Q}_{SA,min} = Cp\ (T_{DHW} - T_{CW})\ \frac{0.15 * V_{day}}{T_{peak}} + \dot{Q}_{recirculation}$$

$$\dot{Q}_{SA} = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{peak} - Ca}{T_{peak}} + \dot{Q}_{recirculation}$$

$$\dot{Q}_{SA,max} = \dot{Q}_{SI,min}$$

As a result, the volume of the storage tank Ca ranges from $V_{peak,}$ to $0.5\ V_{day,}$. To avoid stagnant water a recirculation line is used. The recirculation flow rate $\dot{m}_{recirculation}$ is set to maintain at least 55°C in the DHW secondary network. Hence the power needed to recirculation is:

$$\dot{Q}_{recirculation} = Cp\ \dot{m}_{recirculation}(T_{DHW} - 55)$$

This DHW supply technique has the advantages of lowering the required power at the heat exchanger and to be simple. But the space requirement is higher and it is most costly.

iv. Accumulation supply

The accumulation supply consists of using a heat exchanger coupled with a storage tank able to cover totally or partially the daily DHW volume needs and at least the DHW volume needs during the consumption peak. A hydraulic scheme is presented in Figure 28.

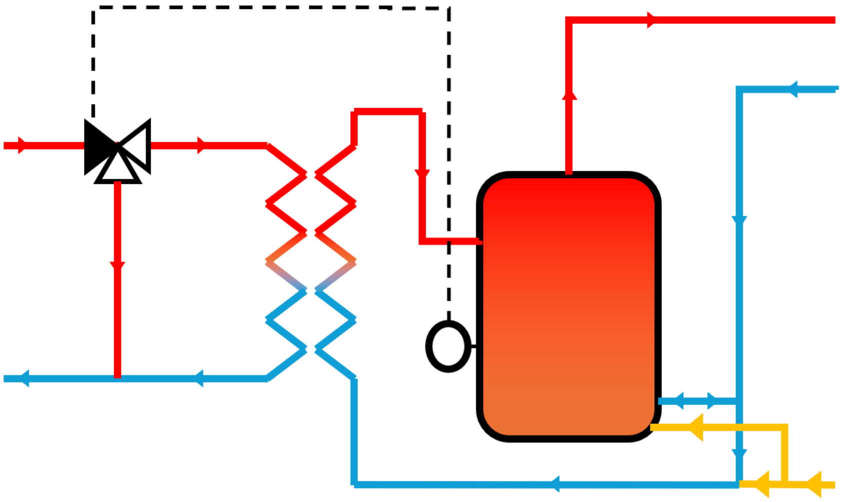


Figure 28 Scheme of an accumulation supply of DHW

In practice, the storage tank is sized to ensure the heating of the tank within 8 hours. The design power of the accumulation heat exchanger depends on the size of the storage tank. The design power $\dot{Q}_A$ (W), the minimum one $\dot{Q}_{A,min}$ and the maximum one $\dot{Q}_{A,max}$ can be expressed as:

$$\dot{Q}_{A,min} = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{day}}{24} + +\dot{Q}_{recirculation}$$

$$\dot{Q}_A = Cp\ (T_{DHW} - T_{CW})\ \frac{V_{day}}{8} + +\dot{Q}_{recirculation}$$

$$\dot{Q}_{A,max} = \dot{Q}_{SA,min}$$

This DHW supply technique has the advantages of lowering the required power at the heat exchanger and to be simple. But the space requirement is higher and it is most costly. Moreover there is a need to add a recycling loop which induces continuous heat losses.

All the equations are summarized in the **Hiba! A hivatkozási forrás nem található.**.

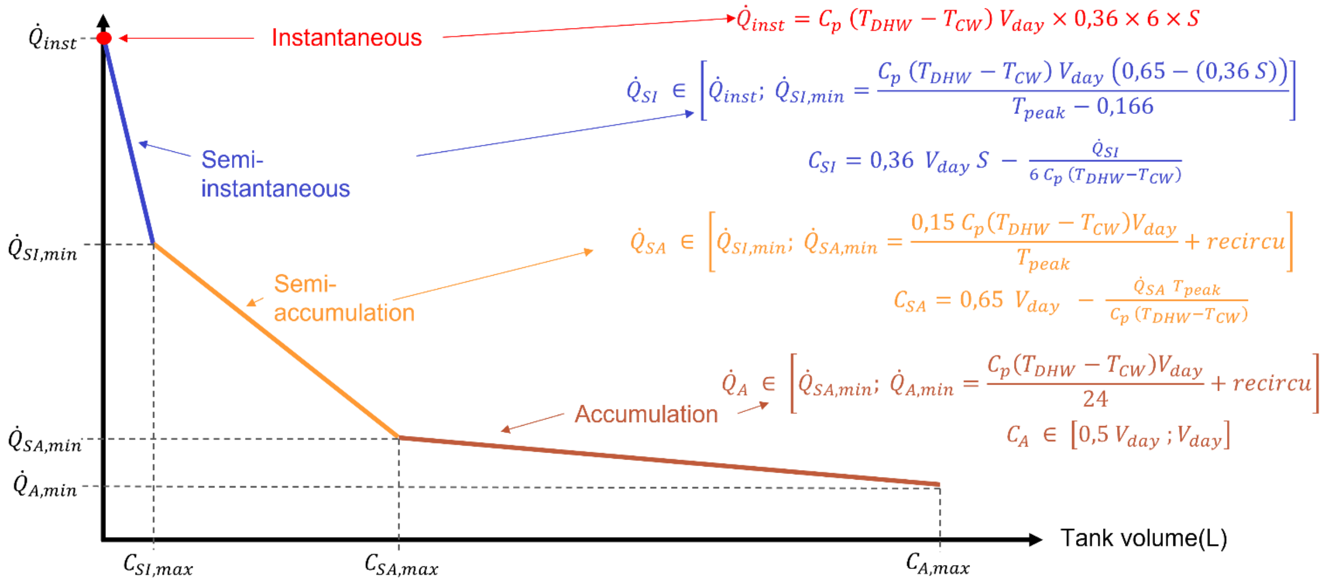


Figure 29 Summary of the different types of DHW supply techniques

4. Building side hydronic systems

This section focuses on all the operations and devices located between the occupied spaces (rooms to be heated or cooled) and the substation. The possible hydronic configurations and systems and their uses are very diverse and numerous. This section is not intended to be exhaustive, only the most common solutions will be discussed. The operation and the sizing of the emitters (radiators, heating and cooling floors, fan coil units), hydraulic network, valves and pumps will be covered. The following sections are based on classical operation in French buildings side hydronic systems and European standards.

a. Heat emitters

In buildings connected to district heating networks, usually three different heat emitters are used: radiators, heating floors and fan coil units. Each of these devices supply space heating needs by heating the indoor air thanks to hot water. Nevertheless, they do not ensure thermal comfort in the same way. Indeed, the radiators and heating floors heat up a room through radiation and convection impacting the wall and air temperature whereas the fan coil unit only heats up the air. This allows the radiator and the heating floor to diffuse more homogeneous and soft heat. Radiator remains the most used heat emitter, it benefits from a low price and an easy installation, but it takes up a significant space in a room and it can't usually be used to provide space cooling. The heating floor is a widespread heat emitter because it does not clutter the space and can be used to provide space cooling. Nevertheless, these emitters are expensive and have a high inertia that could be not suitable in certain buildings. The fan coil unit benefits from the same advantages as the heating floor but it is cheaper, and the energy efficiency is lower. The emitters will affect all the district heating or cooling network operations. Indeed, the choice and sizing of the emitters following the maximal power needed will dictate the supply temperature and the mass flow rate required.

i. Radiators

Radiators are systems that allow to supply heat through hot water circulating inside the elements. It is characterized by their high-water content. It can be built with several materials (cast iron, steel, aluminum, stone, …) and can take a variety of forms (see Figure 29).

Figure 29 Example of radiators

According to the European norm EN 442 [9] the power transferred by a radiator is given by

$$\dot{Q}_{radiator} = UA\,\Delta T^n = UA\left(T_{moy,rad} - T_{indoor}\right)^n = UA\left(\frac{T_{in,rad} + T_{out,rad}}{2} - T_{indoor}\right)^n$$

Where $\dot{Q}_{radiator}$ is the exchanged power (W), UA is the global exchange coefficient (K/W), ΔT is temperature difference (K) between the indoor temperature (T_{indoor}) and the mean temperature of the water inside the radiator $T_{moy,rad}$. The mean water temperature is calculated as the mean temperature between the inlet and outlet water temperatures of the radiator. n is the coefficient characterizing the type of radiator:

- $n = 1.3$ for classic radiators
- $1.1 < n < 1.3$ for convector
- $n = 1$ for fan coil units

According to the previous equation, the power transferred by the radiator depends on the exchange surface, the global exchange coefficient, the water temperature and the indoor temperature. The indoor temperature is set by the occupant. To enhance the heat diffusion, the largest possible exchange surface is required. Hence the radiators have often specific shapes with a lot of pipes and fins (see Figure 31) .

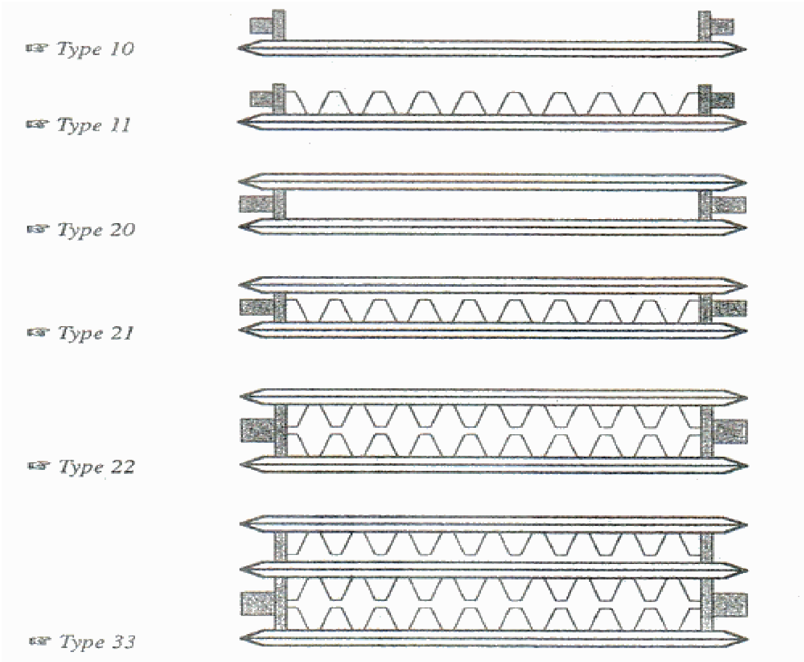


Figure 30 Examples of radiator exchange surface

The power exchanged by a radiator depends on the global transfer coefficient U. This coefficient is calculated through the heat transfer mode. A radiator interacts with its surroundings through natural convection and radiation. Hence a radiator not only heats the indoor air but also the walls allowing to diffuse more comfortable heat. Approximately 30% of heat transfer occurs through radiation and 70% through convection.

A radiator is mostly defined by its regime of temperature and more specifically by the inlet/outlet water temperatures at the design configuration. Hence an 80/60 radiator means that at the design outdoor temperature (-7°C for example in France) water temperature of the radiator is equal to 80°C at the inlet and 60°C at the outlet. A high temperature radiator operates at 80/60 or 75/55 °C, they often correspond to old emitters with a low global transfer coefficient. On the contrary, a low temperature radiator operates at 60/40 or 55/45 °C, they correspond to the new generation of radiators used in new or refurbished buildings. For the same heating power, a low temperature radiator has a larger surface area than a high temperature one (see Figure 31).

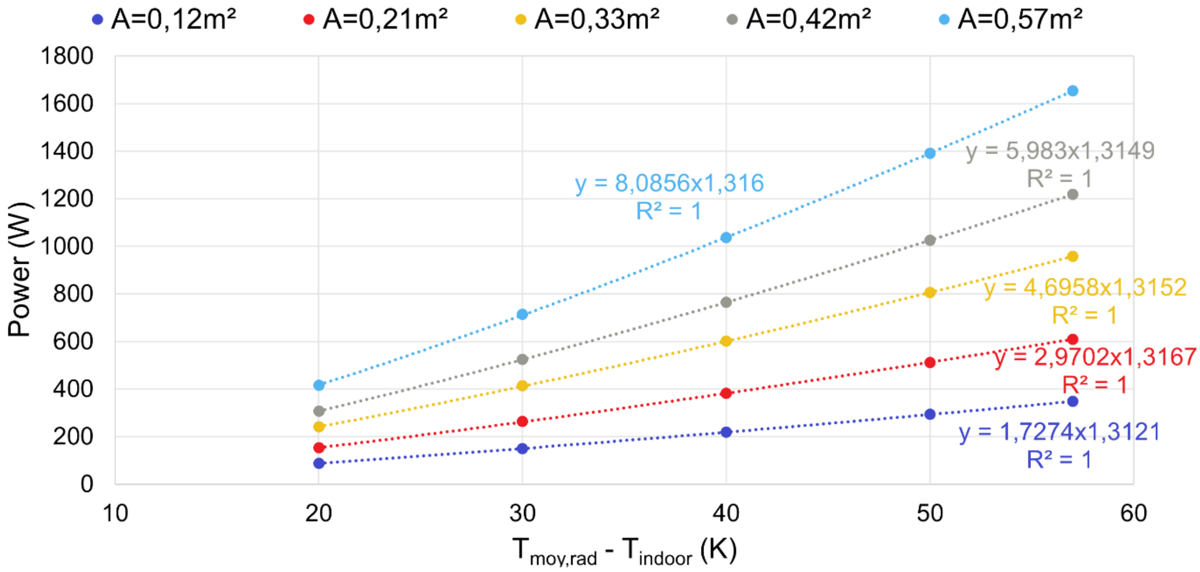


Figure 31 Power supplied by a typical radiator according to the exchange area and the radiator temperature

The choice of the type of radiators according to the heating power needed will impose the maximum mass flow rate of hot water circulating in the radiator:

$$\dot{m}_{rad} = \frac{\dot{Q}_{radiator}}{Cp\left(T_{in,rad} - T_{out,rad}\right)}$$

Usually, a heat emitter is designed to supply the maximum space heating power required by the room where it is installed. In fact, this maximum power is practically never needed and the power delivered by the radiators must be controlled. To achieve the required indoor temperature for different outside conditions the inlet temperature of the radiator is controlled through a supply temperature law.
This control is tuned to reduce the radiator inlet temperature according to the augmentation of the outdoor temperature. Indeed, as the space heating is thermosensitive the higher the outside temperature is the less space heating is needed. Hence as the relationship between the outside temperature and the heating needs is basically linear, the evolution of the set point inlet temperature is also linear. The inlet temperature law is set up through two points: at the outdoor design point the inlet temperature is equal to the design one and when the outside temperature is equal to the required indoor temperature (heating shut off) the radiator inlet temperature is equal to the required indoor temperature. For example, for an outdoor design temperature of -7°C, an indoor temperature of 20°C and an 80/60 radiator:

$$T_{in,rad} = a * T_{outdoor} + b$$

$$\begin{cases} 80 = a * -7 + b \\ 20 = a * 20 + b \end{cases} \Leftrightarrow \begin{cases} a = \dfrac{80 - 20}{-7 - 20} \\ b = 80 + 7 * a \end{cases} \Leftrightarrow \begin{cases} a = -2.22 \\ b = 64.5 \end{cases}$$

The Figure 32 represents two inlet temperature control laws for an 80/60 and an 55/45 radiator.

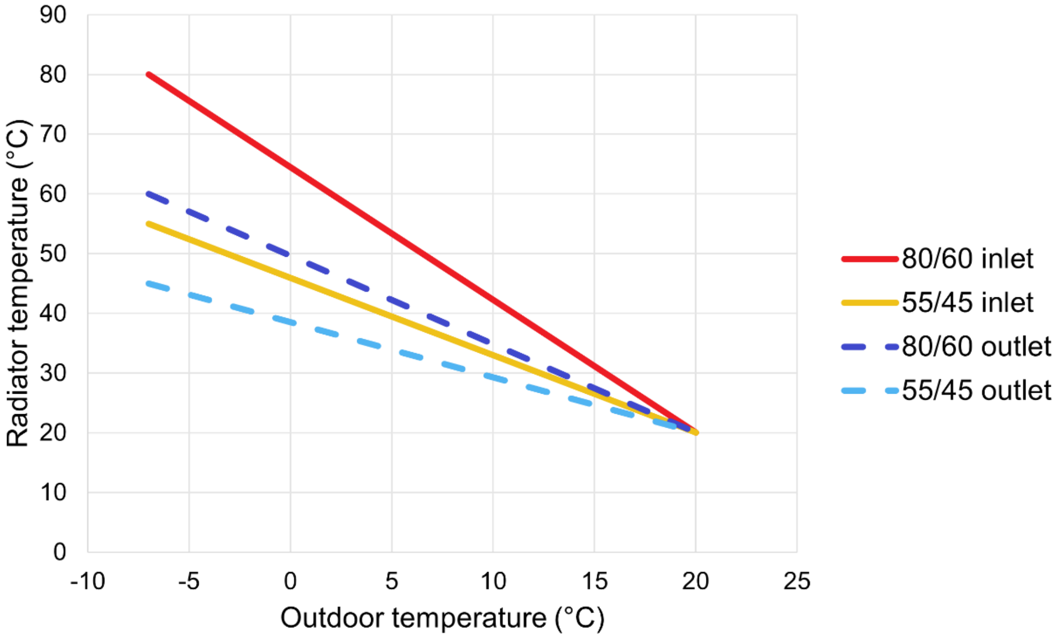


Figure 32 Operating radiator supply temperatures

The inlet temperature is not the only control parameter. As the space heating power required does not only depend on the outdoor temperature. Indeed, on a small scale the heating gains or losses due to the occupant behavior or solar gains affect the space heating power needed regardless of outdoor temperature. Hence, to ensure thermal comfort the water mass flow rate passing through the radiator is also controlled thanks to 2-way or 3-ways valve (see Figure 33)

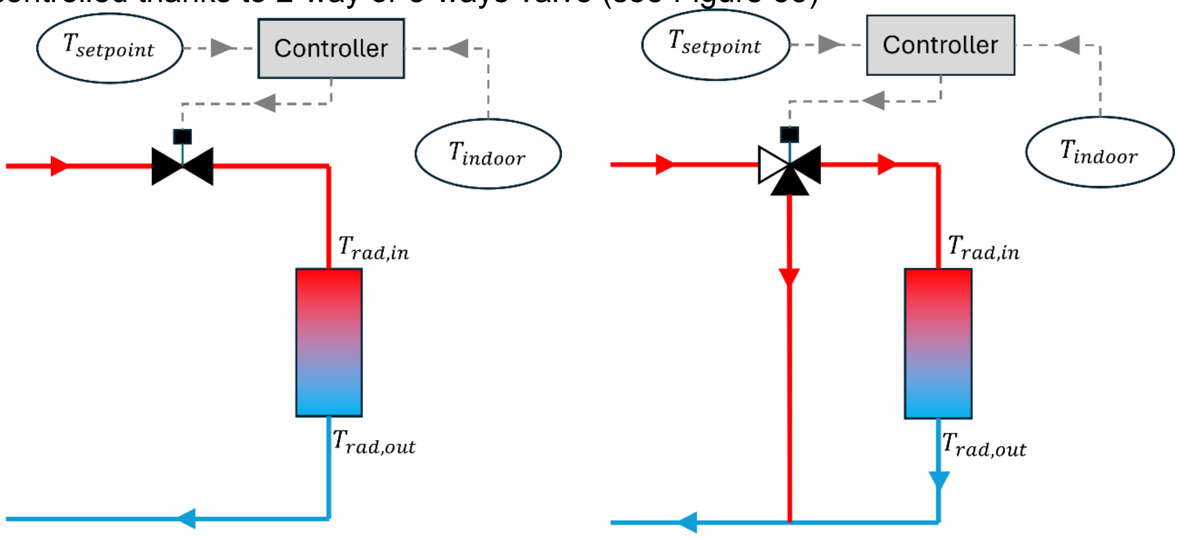


Figure 33 Hydronic regulation of radiators (on the left side two-way valve, on the right three-way valve)

Therefore, the actual indoor temperature of a room is measured and compared to a set-point temperature. According to the difference between the set-point and the measured temperature the valve will open or close to control the flow rate. Hence the indoor set-point temperature will be reached according to the following equation:

$$\dot{Q}_{radiator} = UA\left(\frac{T_{in,rad} + T_{out,rad}}{2} - T_{indoor}\right)^{n} = \dot{m}_{rad}\,Cp\,(T_{in,rad} - T_{out,rad})$$

ii. Heating floor

Heating floors are systems that allow to supply heat through hot water circulating inside the floor. In order to homogenize the heat on the floor the pipes are arranged in a snail shape (see Figure 34)

Figure 34 Example of a heating floor

A heating floor is composed of 4 layers as shown in Figure 35.

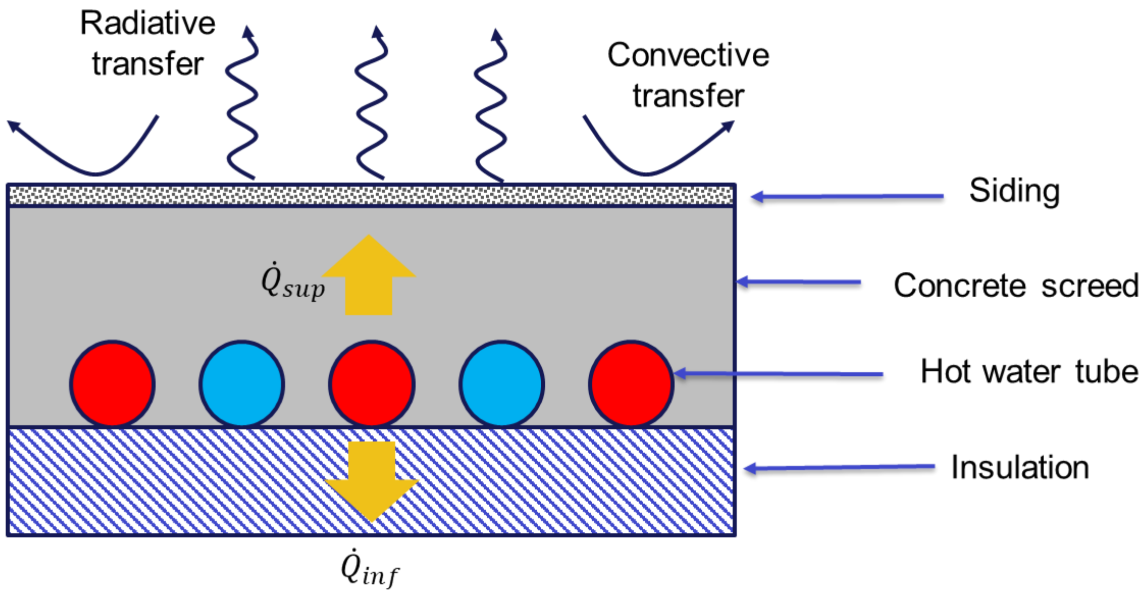


Figure 35 Scheme of a heating floor

The pipes that carry the hot water are embedded in a concrete screed. An insulation layer is installed beneath them to reduce heat loss downward and instead direct more heat into the room being heated. The Figure 36 represents the thermoelectric scheme of the heating floor:

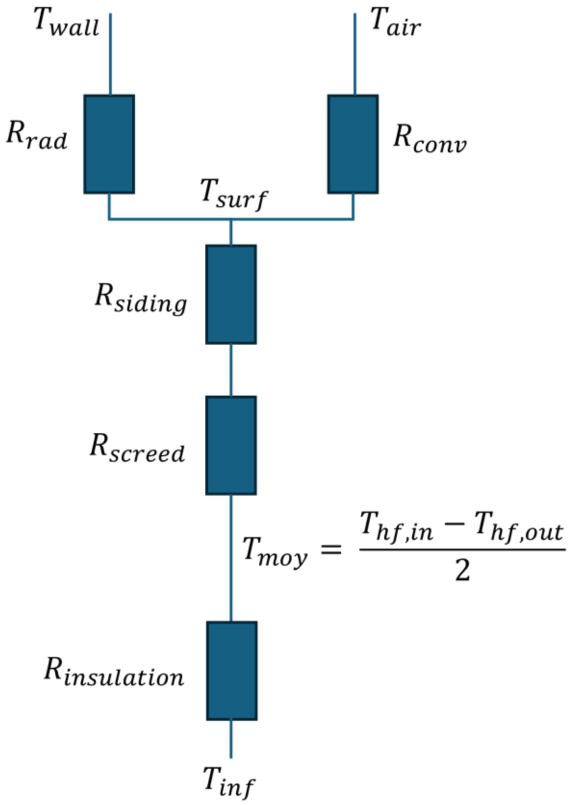


Figure 36 Thermoelectric scheme of a heating floor

According to the previous figure, the power exchanged by the heating floor can be expressed as:

$$\dot{Q}_{hf} = \dot{m}_{hf}\, Cp\, (T_{hf,in} - T_{hf,out}) = \dot{Q}_{sup} + \dot{Q}_{inf}$$

Where $\dot{Q}_{hf}$ stands for the power exchange by the heating floor (W), $\dot{Q}_{sup}$ and $\dot{Q}_{inf}$ stand for the heat flux going to the room and under the floor respectively. $\dot{m}_{hf}$ represents the water flow rate in the heating floor (kg/s), Cp the specific heat (KJ/K/kg) and $T_{hf,in}$ and $T_{hf,out}$ represent respectively the inlet and outlet water temperatures in the heating floor.
The power effectively used for the space heating is $\dot{Q}_{sup}$. $\dot{Q}_{inf}$ is considered as losses in this case, it can be expressed as:

$$\dot{Q}_{sup} = \frac{T_{moy} - T_{surf}}{R_{screed} + R_{siding}}$$

$$\dot{Q}_{inf} = \frac{T_{moy} - T_{inf}}{R_{insulation}}$$

Where T_{moy} is the mean temperature of the water on the heating floor (K), T_{surf} the temperature at the surface of the heating floor (K), T_{inf} the temperature of the element under the insulation (K). R_{screed}, R_{siding} and $R_{insulation}$ stands for the thermal resistance (K/W) of the screed, the siding and the insulation respectively.
The power exchanged by a heating floor depends on the global transfer coefficient U. This coefficient is calculated through the heat transfer mode. A heating floor interacts with its surroundings through natural convection and radiation. Hence a heating floor not only heats the indoor air but also the wall allowing to diffuse more comfortable heat. Approximately the

heat transfers through radiation represent 70% and through convection 30% of the total exchanged heat.
Contrary to the radiator, the heating floor benefits from its high surface. Hence, the heating floor can operate at lower temperature than the radiators. Moreover, due to health consideration the surface temperature cannot exceed 28°C. So according to the European standard EN ISO 13370 [10], the temperature regime of the heating floor is 35/30 (i.e. $T_{hf,in} = 35°C \ and \ T_{hf,out} = 30°C$) at the design outdoor temperature.
As for the radiator, the inlet temperature and the flow rate are controlled to fulfill the power and indoor temperature needs. The inlet temperature is controlled in the same way as the radiator (see Figure 37)

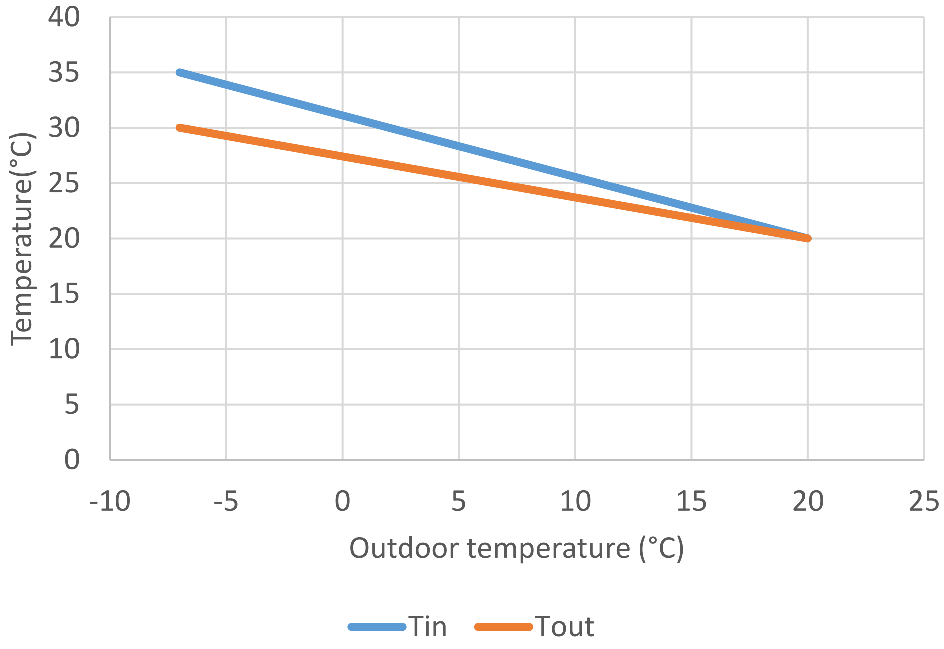


Figure 37 Operating heating floor temperature

The flow rate is also controlled to achieve the required indoor temperature and to respect the surface temperature constraint. Usually a 3-way valve is set at the inlet of the heating floor to monitor the flow rate and to cool down the inlet temperature with the outlet flow to respect the constraint (see Figure 38).

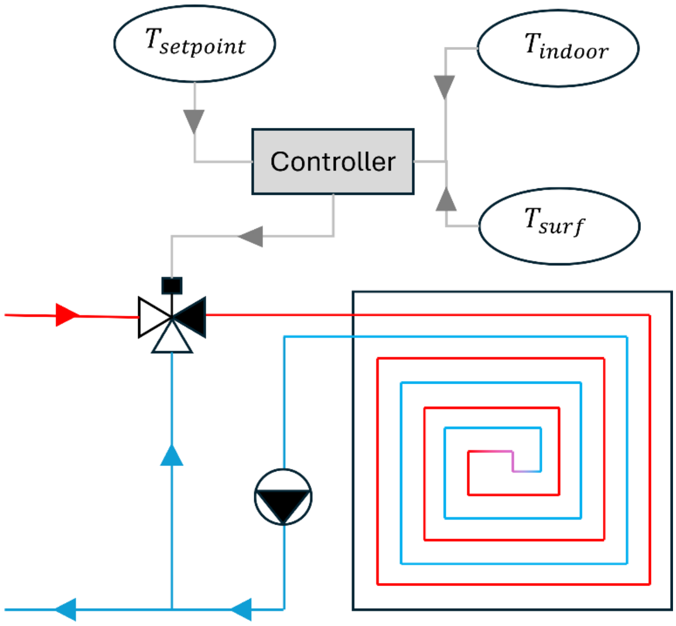


Figure 38 Hydraulic regulation of heating floor

iii. Fan coil unit

Fan coil units are systems that allow to supply heat through hot water circulating inside a heat exchanger that transfers the heat to indoor air moved through a fan.

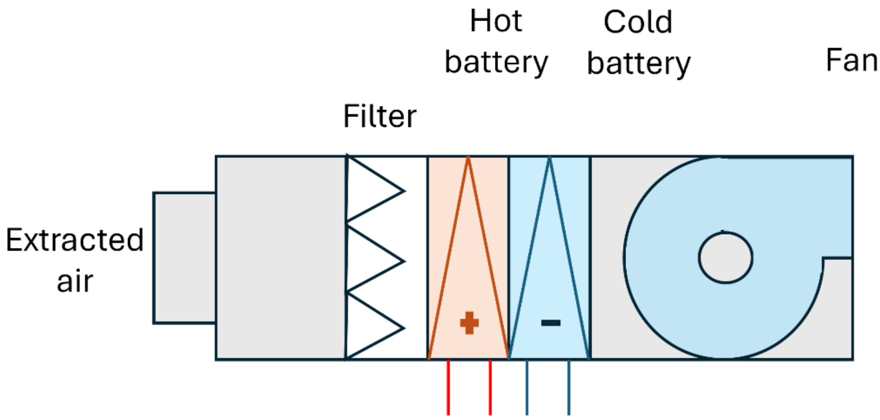


Figure 39 Scheme of a fan-coil unit

A fan coil unit is composed of a fan, one or two heat exchangers, a condensate collection tray and a filter (see Figure 39). It can deliver either heating or cooling to a room using the same heat exchanger or two separate ones. Space heating can also be provided through electric resistance. Compared to a radiator, the heat exchange is better. Indeed, the fan coil units work in forced convection leading to a better heat convection coefficient. Hence the temperature regime is 50/40 for an indoor air of 20°C to blow air up to 35°C (see Figure 40)

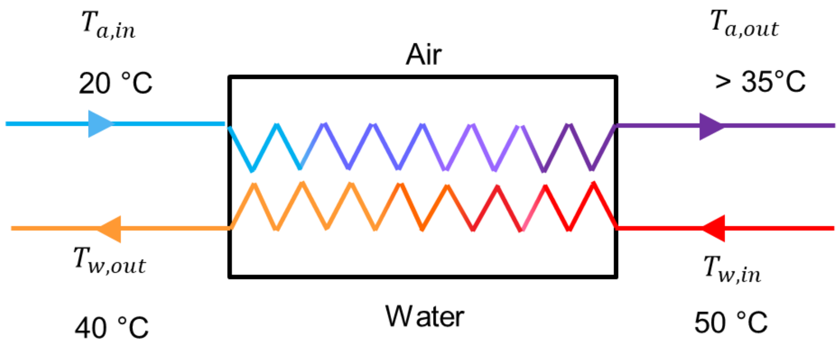


Figure 40 Scheme of the heat battery

The equations governing the power transfer are the equation of a classical heat exchanger in counterflow.

$$\dot{Q} = \dot{m}_a Cp_a \left(T_{a,out} - T_{a,in}\right)$$
$$\dot{Q} = \dot{m}_w Cp_w \left(T_{w,in} - T_{w,out}\right)$$
$$E = \frac{\dot{Q}}{min(\dot{m}_a Cp_a;\ \dot{m}_w Cp_w)\left(T_{w,in} - T_{a,in}\right)}$$

If only one heat exchanger is used to provide space heating and cooling the hot water inlet and outlet temperature must be redefined. Indeed, in cooling mode the temperature regime is 7/12°C i.e. a temperature difference of 5°C, leading to an oversizing of the heat exchanger in heating mode. Hence, to avoid underperformance in heating mode, the temperature regime must be redesigned to match the same temperature difference as in cooling mode. The fan coil units must operate with a temperature regime of 50/40°C or 35/30°C in heating mode.

As for the radiator, the inlet temperature and the flow rate are controlled to fulfill the power and indoor temperature needs. The inlet temperature is controlled in the same way as the radiator (see Figure 41)

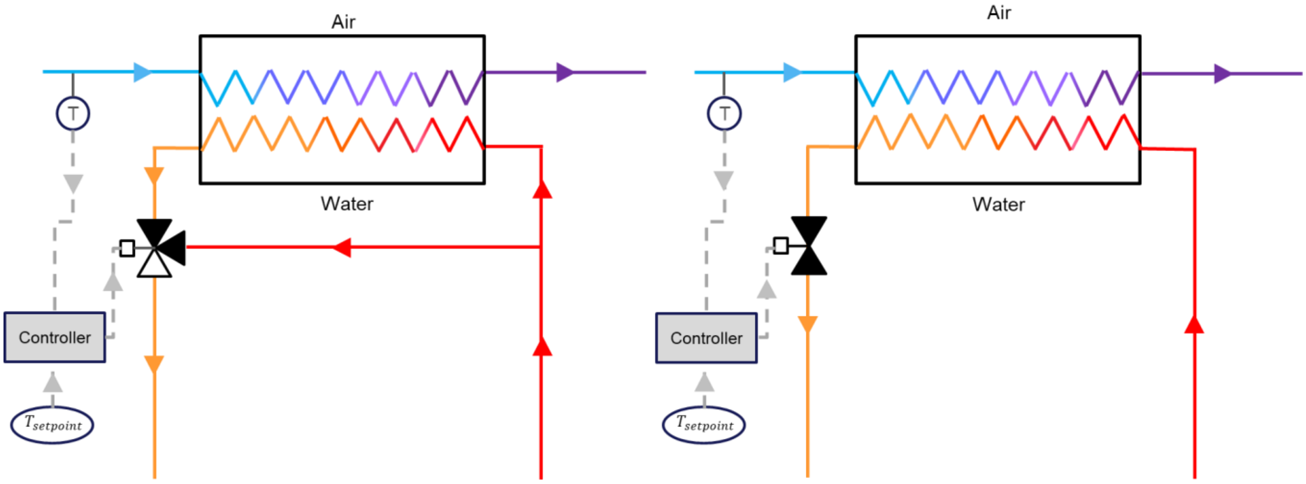


Figure 41 Hydraulic regulation of a fan coil unit (three-way valve configuration on the left and two way-valve configuration on the right)

b. Cold emitters

In buildings connected to district cooling networks, usually three different cold emitters are used: cooling ceilings, cooling floors and fan coil units. Each of these devices supplies space cooling needs by cooling the indoor air thanks to cold water. These three types of terminal units differ in their ability to manage condensate and in their heat transfer capabilities. Indeed, the cooling ceilings and floors cool down a room through radiation and convection impacting the wall and air temperature whereas the fan coil unit only cools down the air. Consequently, thermal comfort is managed differently since the body exchanges heat both by radiation with the walls and by convection with the air. Moreover, the cold ceilings and floors require higher water temperatures to prevent condensation, which limits the available cooling capacity but results in more homogeneous and soft cooling. Since fan coil units (FCU) can recover condensate, they typically operate at temperature regimes of 7/12°C, allowing them to transfer more cooling capacity. All these emitters are reversible. Lastly, FCU is also cheaper and easier to install.

i. Fan coil units

In cooling mode, the fan coil units operate in the same way as in heating mode. Indeed, through an air/water heat exchanger the indoor air is cooled down by cold water with a 7/12°C temperature regime.

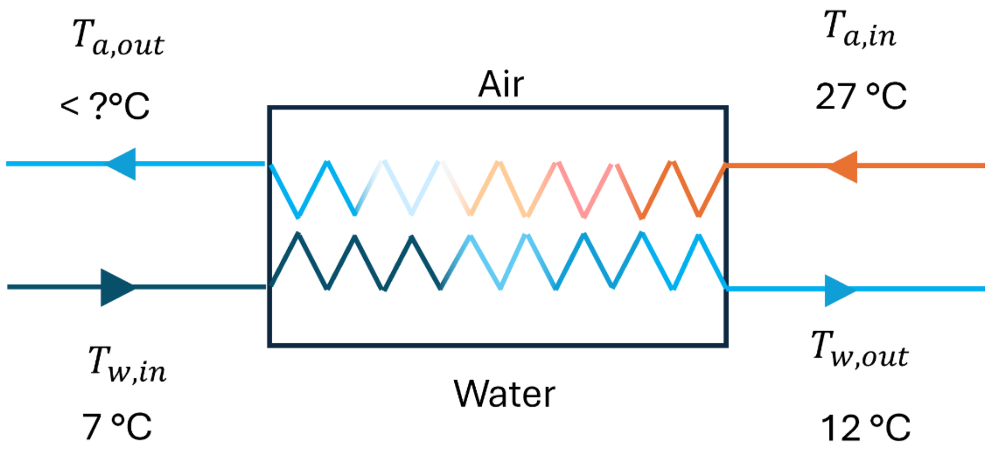


Figure 42 Scheme of the cold battery

Nevertheless, in cooling mode as the indoor air contains water a condensation is possible. Indeed, if the air is refreshed below the dew point temperature the air condenses and power supplied by the cold water is used to cool down the air (the desired service) and the air humidity is partially condensed (a useless service). A psychometric chart is used to determine the dew point temperature and the condensate volume (see Figure 44)

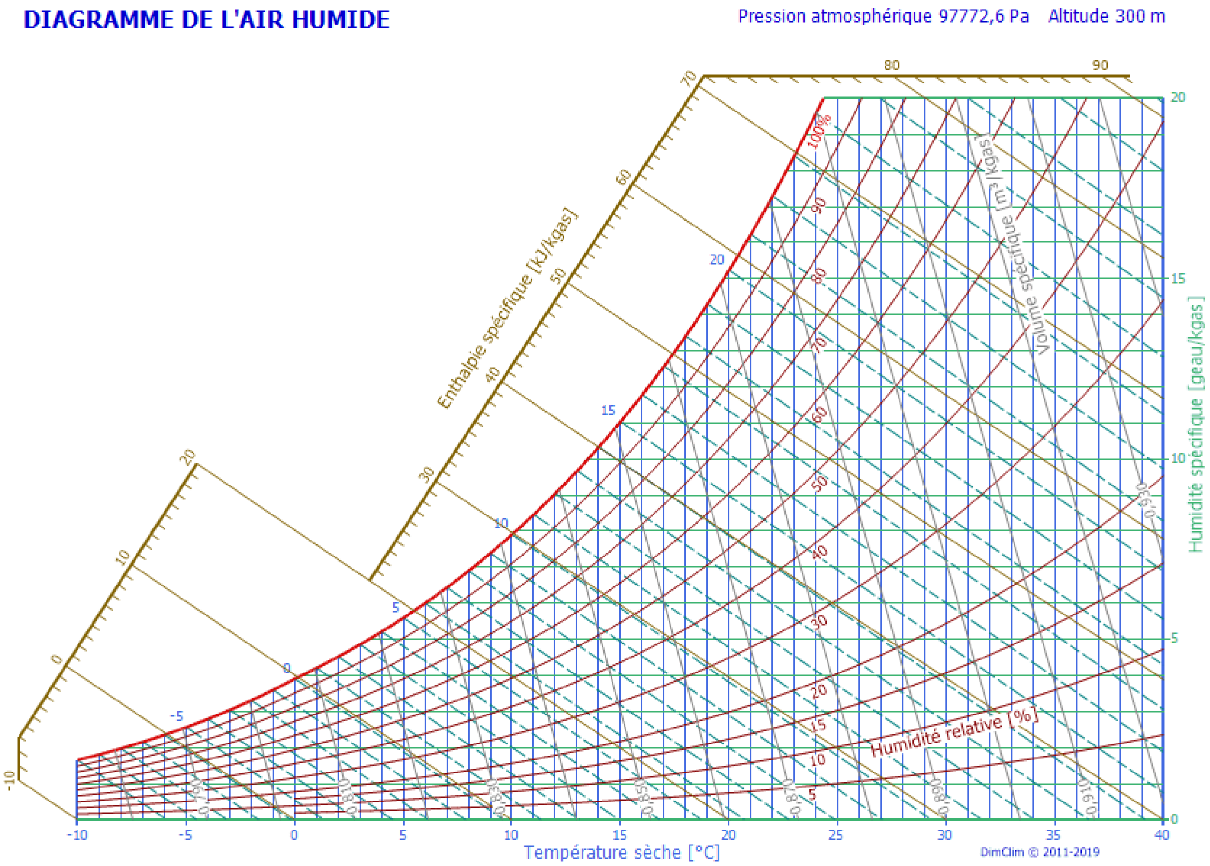


Figure 43 Psychometric chart of the humid air

A lot of parameters can be read through this graph. The x-axis represents the dry bulb temperature θ_s [°C] (i.e. the temperature measured through a simple thermometer), the y-axis stands for the air humidity ratio r [kg_{water}/kg_{dryair}]:

$$r = \frac{m_{water}}{m_{dryair}} = 0.622\frac{p_v}{p - p_v}$$

The air humidity ratio is the ratio between the vapor and dry air masses. It can be calculated through the partial vapor pressure p_v [Pa] and the total pressure of the moist air p [Pa].
On the Figure 43, the diagonal brown line represents the specific enthalpy h [kJ/ kg_{dryair}]:

$$h = h_{dry\ air} + rh_{vapour} = \ Cp_{dry\ air}\ \theta_s + r\left(L_v + Cp_{vapor}\ \theta_s\ \right)$$

where $h_{dry\ air}$ [kJ/ kg_{dryair}] is the specific enthalpy of dry air, h_{vapor} [kJ/ kg_{dryair}] is the specific enthalpy of vapor, $Cp_{dry\ air}$ and Cp_{vapor} [kJ/ K.kg_{dryair}] the thermal capacity of respectively dry air and vapor, and L_v [kJ/ kg_{dryair}] the latent heat of vaporization.
The red line stands for the relative humidity φ [%] the ratio of the vapor partial pressure in the air p_v and the saturation vapor pressure at the same temperature $p_{v,s}$:

$$\varphi = \frac{p_v}{p_{v,s}}$$

The diagonal grey line represents the specific volume v [m^3/ kg_{dryair}].
The other variables are determined by graphic construction on the psychometric chart. The dew point temperature θ_{dp} [°C] is determined by the intersection of the horizontal line from the indoor condition (dry temperature and humidity) and the saturation curve (see Figure 44). The wet bulb temperature θ_h [°C] is the temperature measured by a thermometer whose bulb is covered with wet cotton and placed in a stream of air (2 m/s). On the psychometric

chart it can be read by following the dash lines closed to isenthalpic lines up to the saturation curve (see Figure 44).

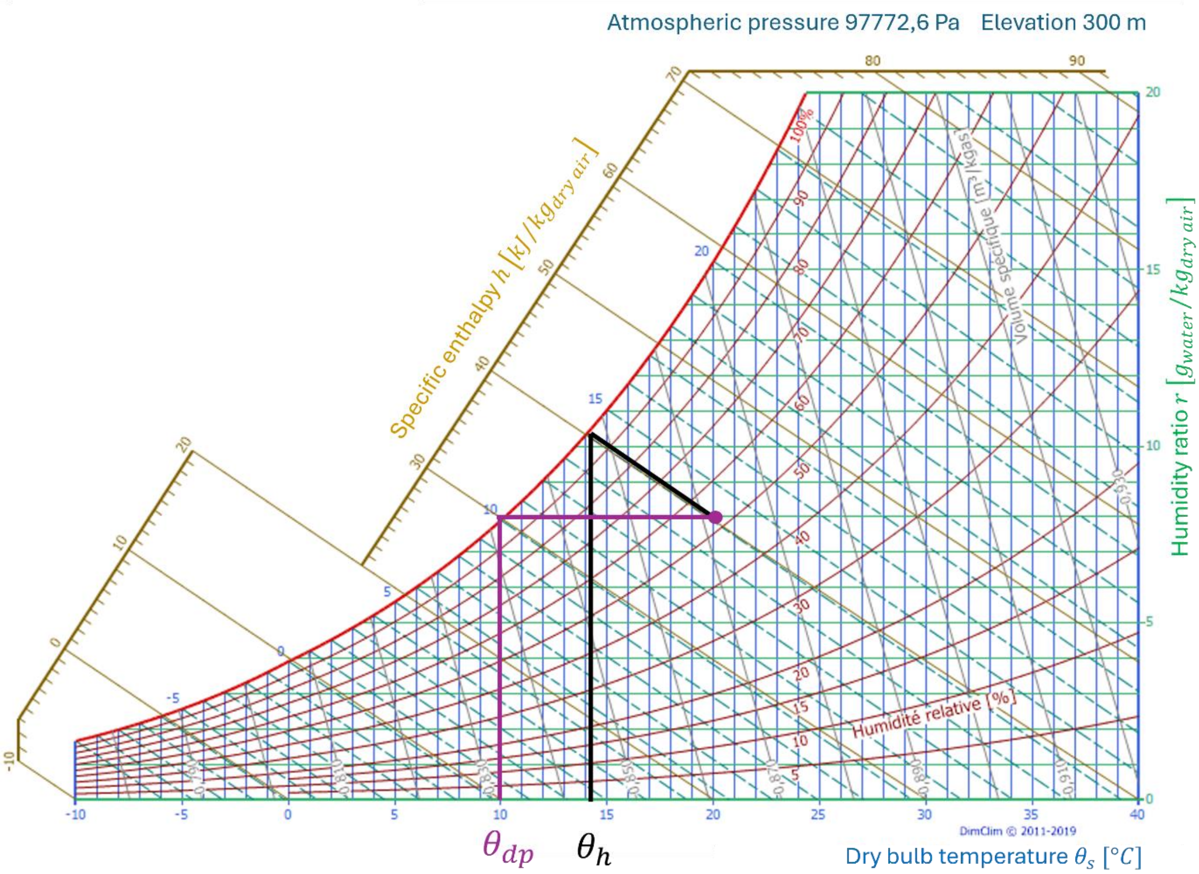


Figure 44 Psychometric chart of the humid air

The equations governing the cold heat exchanger of the fan coil are the same as for the hot heat exchanger except for the power exchanged in the air side:

$$\dot{Q} = \dot{Q}_{sensible} + \dot{Q}_{cond}$$

$$\dot{Q} = \dot{m}_{air}\, Cp_{air}\, (T_{a,in} - T_{a,out}) + \dot{m}_{cond}\, L_v$$

$$\dot{m}_{cond} = \dot{m}_{air}\, (r_{a,in} - r_{a,out})$$

With $\dot{Q}_{sensible}$ and $\dot{Q}_{cond}$ (W) respectively the sensible and the condensation power, $\dot{m}_{cond}$ (kg_{water}/s) the condensate flow rate and $r_{a,in}$ and $r_{a,out}$ [kg_{water}/kg_{dryair}] the humidity ratio respectively at the inlet and the outlet of the heat exchanger.

ii. Cooling floor and ceiling

The cooling floor and ceiling are constituted in the same way as a heating floor (see Figure 45 & Figure 46). The cold water circulating in the tubes will transfer its energy to several layers and the room will be cooled down through radiation and natural convection.

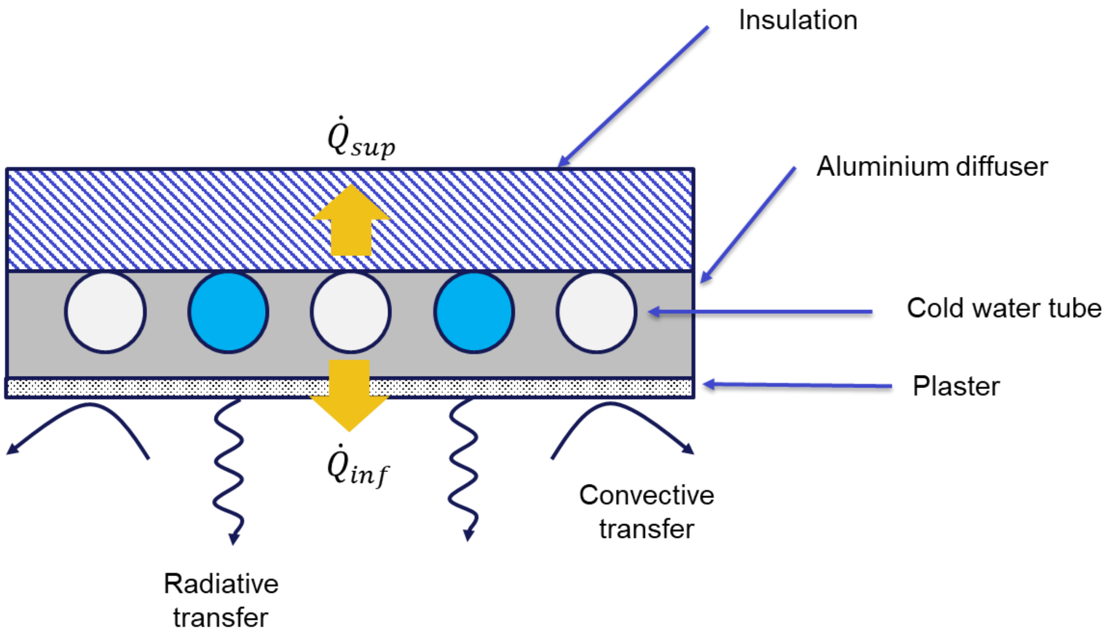


Figure 45 Scheme of a cooling ceiling

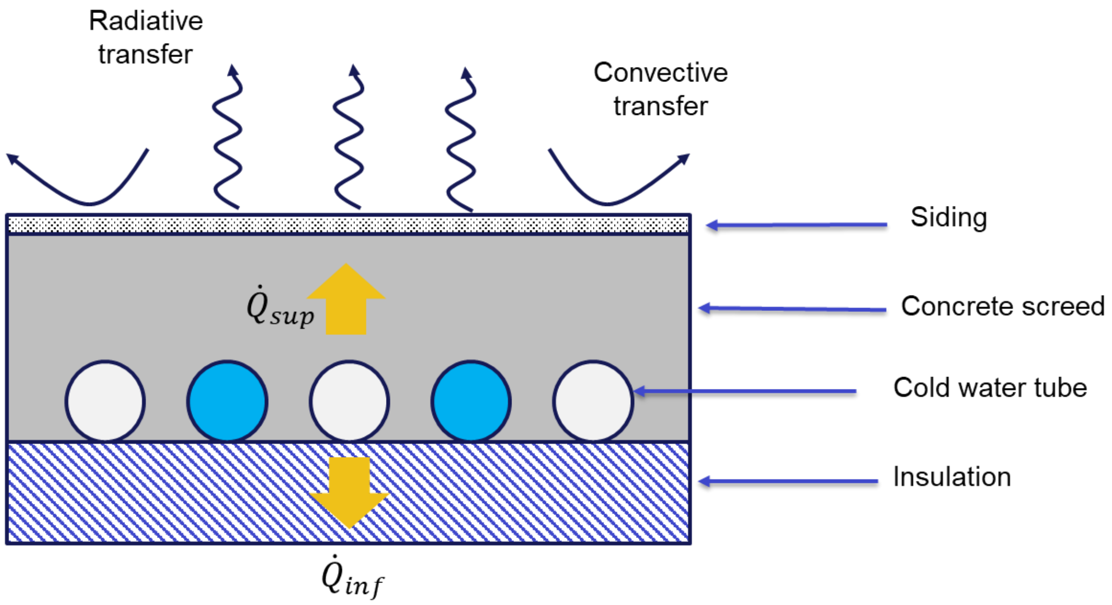


Figure 46 Scheme of a cooling floor

Hence the equations modelling thermal transfer are similar. Only the value of the transfer coefficient will change; the radiative heat transfer coefficient can vary with the emissivity of the surface and its temperature and the convective heat transfer coefficient is different between a ceiling and a floor. As the fan coil unit, the indoor air can condense on contact with the cooling floor or ceiling. But the condensation must be prevented with these cold emitters to avoid getting water on the floor. Hence the cooling floor and ceiling are monitored to limit their surface temperature to 18°C maximum in temperate climates.

c. Distribution

This section deals with hydraulic distribution in the secondary side of a district heating and cooling networks. Usually, this part of the network is not often covered because the DHC network operator does not usually have control over it. Nevertheless, the secondary distribution systems still have a great impact on the overall DHC network operation. This section aims to understand how the hydraulic distribution of the secondary side works. These hydraulic systems are composed of three key elements: pipes, valves and pumps.

i. Pipes

A pipe in the building side hydronic systems aims to distribute the hot or cold water from the substation to the emitters. It is composed of steel or PVC tube and thermal insulation (see Figure 47 & Figure 48)

Figure 47 Example of a hot water pipe in a building hydronic system

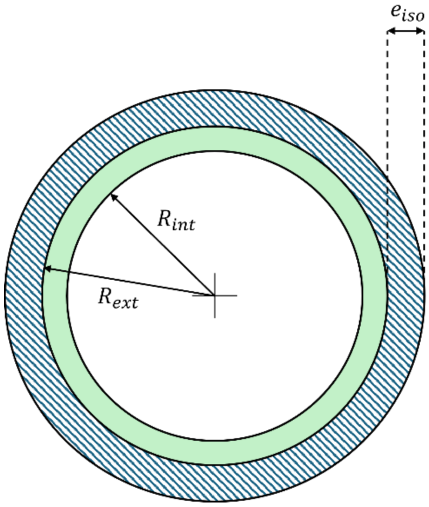


Figure 48 Scheme of a pipe

A pipe must respond to several constraints:

- Having the smallest diameter possible to reduce costs
- Minimizing the heat losses using a sufficient insulation thickness
- Minimizing the pressure losses to reduce the pump consumption

All these constraints are considered to design a pipe. The first point is to calculate the right diameter, knowing the maximal needed mass flow rate in the pipe. Indeed, this parameter will affect the cost, the heat losses and the pressure losses. In the European standard two equivalent methods are described [11] . The first one consists in limiting the velocity of the water inside the tube to $1\ m/s$. This limit allows to reduce the noise of the water circulation in the pipe to maintain good acoustic comfort in the buildings and to avoid possible vibration of the pipe. Hence the following equation is used to assess the pipe interior diameter:

$$\dot{m}_w = \rho\, v\, S = \rho\, v\, \frac{\pi\, {D_{int}}^2}{4}$$

$$D_{int} = \sqrt{\frac{4\,\dot{m}_w}{\pi \rho\, v}}$$

With $\dot{m}_w$ the mass flow rate (kg/s), ρ the density (kg/m^3), v the velocity (m/s), S the pipe cross-section (m^2) and D_{int} the pipe inside diameter (m)

Limiting the water velocity enables also to reduce the pressure losses according to the Darcy-Weisbach equation:

$$\Delta p = \Lambda \frac{L}{D_{int}} \frac{\rho v^2}{2}$$

With Δp the pressure losses (Pa), Λ the friction factor and L the pipe length (m).

Hence the second method to design the pipe diameter is to restrain directly the pressure losses instead of the velocity. Usually, the linear pressure losses must be lower than $100\ pa/m$. Therefore, the diameter is calculated through the following equation (Darcy-Weisbach):

$$\frac{\Delta p}{L} = \Lambda \frac{8}{\rho \pi^2} \frac{\dot{m}_w^2}{D^4} \Leftrightarrow D = \left(\frac{\Lambda \mathrm{L}}{\Delta p} \frac{8}{\rho \pi^2}\ \dot{m}_w^2\right)^{1/4}$$

With

$$\begin{cases} \Lambda = \dfrac{64}{Re} & if\ Re < 2000 \\ \Lambda = 0.3164\ Re^{-0.25} & if\ Re > 4000 \end{cases}$$

With Re the number of Reynolds

In practice, the inside diameter of the pipes is normalized. The real pipe is chosen by selecting the normalized pipe with the inside diameter just above the calculated one to be always inferior to the design constraint. Table 6 presents the normalized pipe dimension.

Normalized diameter	Inside diameter (mm)	Outside diameter (mm)
DN 6	5	10
DN 8	8	13
DN 10	12	17
DN 15	15	21
DN 20	20	27
DN 25	26	34
DN 32	33	42
DN 40	40	49
DN 50	50	60
DN 60	60	70
DN 65	66	76
DN 80	80	90
DN 90	90	102
DN 100	102	114
DN 125	125	139
DN 150	150	168

Table 6 Relationship between the nominal diameter DN of a pipe and the inside and outside diameter

Then according to the pipe's outside diameter the thermal insulation thickness can be determined according to the European standard EN 15316-3 According to the wanted class of insulation the standard proposes the linear thermal conductance to achieve (see Figure 50).

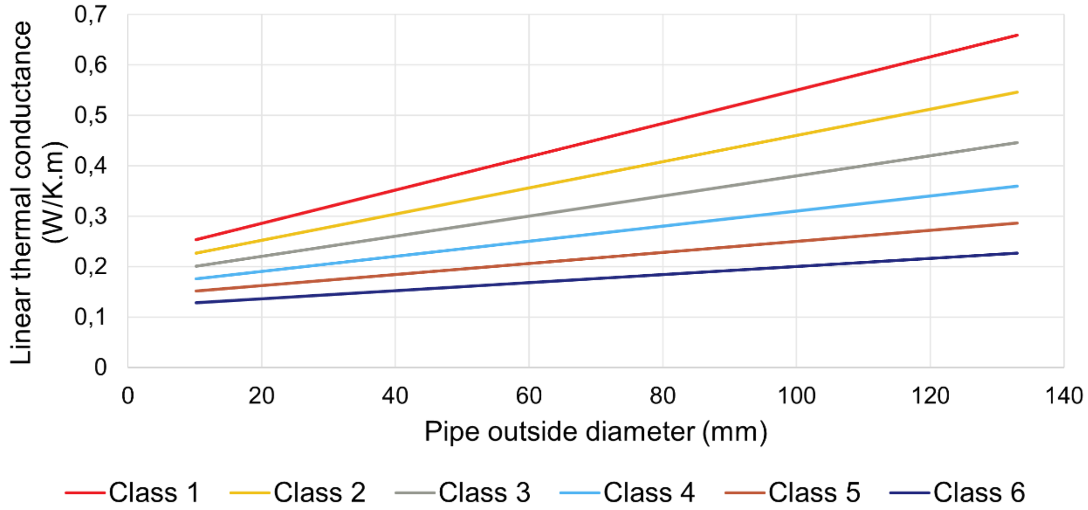


Figure 49 Evolution of the pipe linear thermal conductance according to the pipe outside diameter and the insulation class

The class of insulation corresponds to a level of heat loss, the higher the class is the less important heat losses will be (see Figure 50).

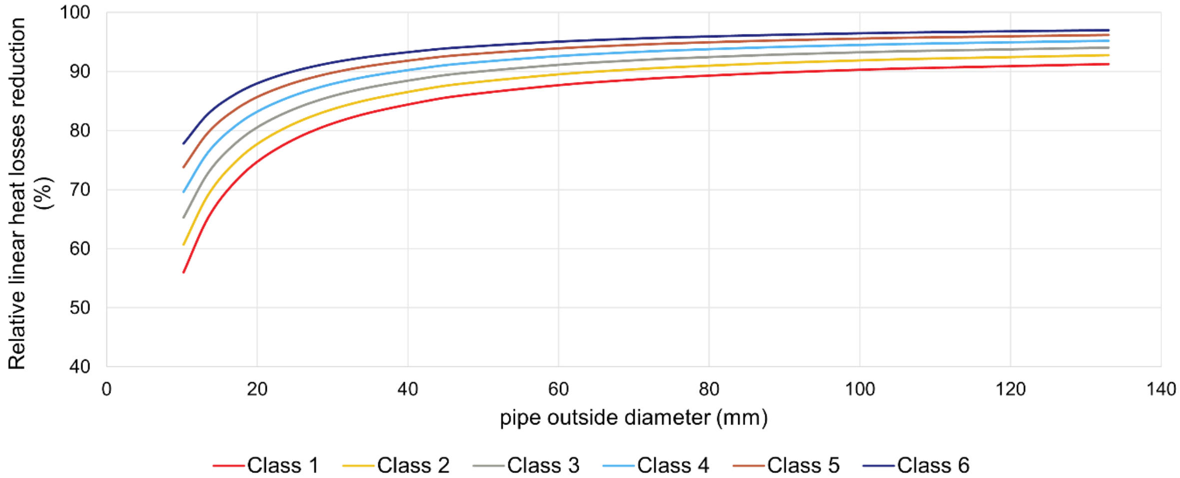


Figure 50 Evolution of the relative

Knowing the outside diameter of the pipe and the linear thermal conductance to achieve the thermal insulation thickness can be assessed according to the following equation, neglecting convective heat transfer inside the pipe, since the convective heat transfer coefficient for water under forced convection is much higher than that for air under natural convection:

$$\frac{1}{U_l} = \frac{ln\left(\frac{e_{iso} + R_{ext}}{R_{ext}}\right)}{2\pi\lambda_{iso}} + \frac{1}{h\ 2\pi\ (e_{iso} + R_{ext})}$$

With U_l the linear thermal conductance $(W/K.m)$, e_{iso} the thermal insulation thickness (m), R_{ext} the pipe outside radius (m), λ_{iso} the insulation thermal conductivity $(W/K.m)$ and h the thermal convection coefficient outside the pipe $(W/K.m^2)$.

Once all the geometrical dimensions of the pipes in the building side hydraulic network are determined, the real pressure losses at the design point must be calculated. Indeed, it is important to check if the required mass flow rate is reached in all the emitters to avoid thermal discomfort. Thus, the most disadvantaged emitter (i.e. the emitters with the most cumulated pressure losses) risks to operate with less flow rate than required at design point. This is called the hydraulic unbalancing (see Figure 52)

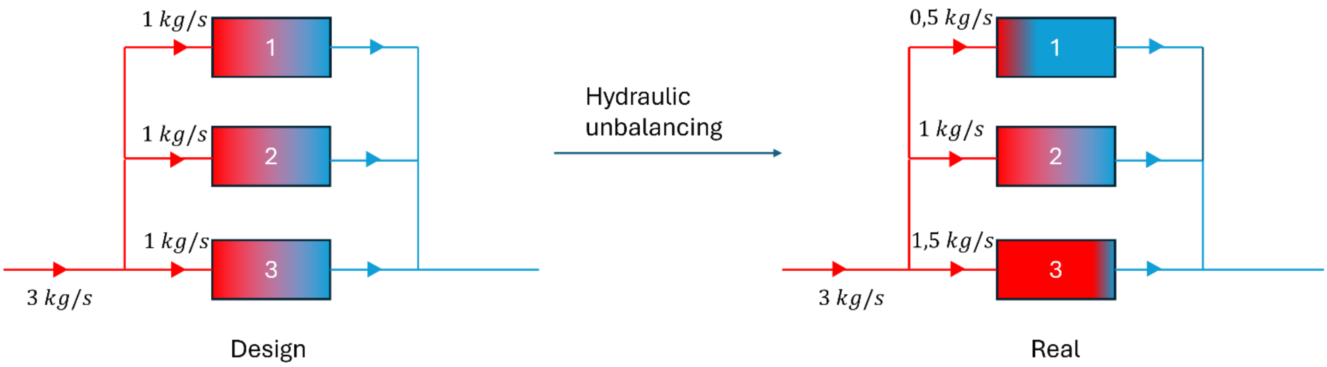


Figure 51 Difference between balanced and unbalanced hydraulic network

The pressure losses of all the different paths can be calculated through the hydraulic electric equivalence:

$$\Delta p = \Lambda \rho \frac{v^2}{2} = \frac{8 \Lambda}{\rho \pi^2} \frac{\dot{m}_w^2}{D^4} = Z \dot{m}_w^2$$

With Z the hydraulic resistance ($pa.s^2/kg^2$).

To achieve the hydraulic balance, some valves must be added to obtain the design flow rate.

ii. Valves

The valves are a device allowing to modify the flow rate of a fluid by adding or deleting pressure losses. To ensure hydraulic balancing static valves are added in the distribution circuit to increase pressure losses in specific parts of the network. These types of valves are not intended to be controlled. They only ensure the hydraulic balance at the design point. For example, on a network where all the emitters must be supplied with the same flow rate, the static valves must be placed to equalize the pressure losses in all branches (path from the pump to the emitters back and forth).

Once the hydraulic balancing is achieved other types of valves must be installed to control the flow rate passing through the emitters for non-design conditions. These valves are named control valves. They are monitored by an actuator whose modify the size of the orifice through which the fluid passes (opening rate). In this way, the pressure drop across the valve is modulated with the result that the flow rate through the valve is controlled (see Figure 52).

A control valve can be modelled by:

$$\dot{m} = Kv_s \sqrt{\Delta p} * opening$$

With Kv_s the coefficient representing the mass flow rate passing through the valve for a fully open valve ($opening = 1$) for a $\Delta p = 1$ bar. The coefficients Kv_0 and Kv represent the same parameter but respectively for a fully closed valve and for an opening rate between 0 and 1 excluded.

$$\frac{\dot{m}}{\sqrt{\Delta p}} = \begin{cases} Kv_s & if \ \ opening = 1 \\ Kv & if \ opening \neq 1 \quad for \ \ \Delta p = 1 \ bar \\ Kv_0 & if \ \ opening = 0 \end{cases}$$

The control valve can have different characteristic curves, i.e. different relations between the valve coefficient Kv and the opening rate. A linear valve is characterized by a linear relationship such as:

$$\frac{Kv}{Kv_s} = opening$$

The second possibility is the equal percentage characteristic. Here the opening variation corresponds to equal percentage Kv variation according to the Kv value before the variation:

$$\frac{(Kv - Kv_0)/(Kv + Kv_0)}{(Kv_s - Kv_0)/(Kv_s + Kv_0)} = opening$$

In practice the relation between the valve coefficient Kv and the opening rate is quadratic.

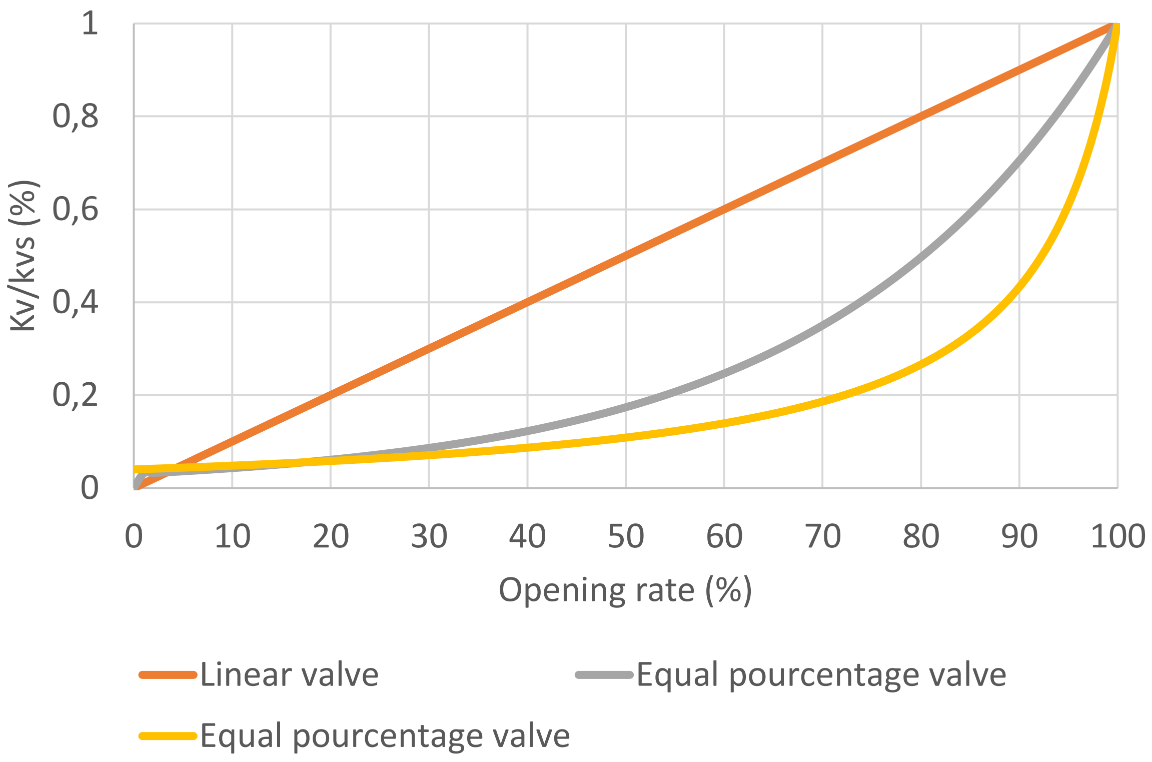


Figure 52 Valve characteristics

To be effective a valve must add a significant pressure loss in comparison to the total pressure losses of the networks regulated by the valve. This effectiveness is called valve authority:

$$a = \frac{\Delta p_v}{\Delta p_v + \Delta p_c}$$

With a the authority, Δp_v the pressure losses due to the valve fully open (Pa) and Δp_c the pressure losses of the network to regulate (Pa). The higher authority is the more the valve will operate according to its specifications. When the authority is close to 0, the valve will operate as an on/off valve (see Figure 53)

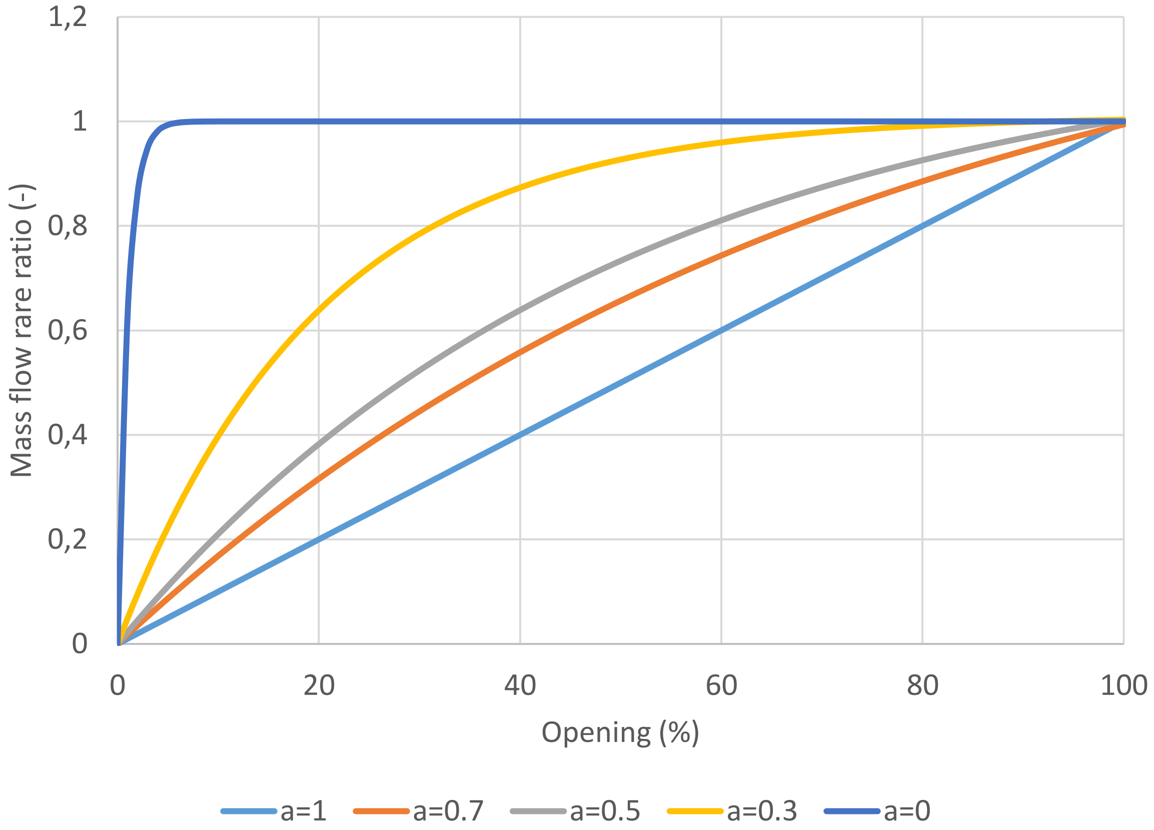


Figure 53 Effect of valve authority on the valve's characteristics

In a heat exchanger or in a radiator, the exchanged power is not proportional to the flow rate. Hence, to have more accurate and stable control a non-linearity must be introduced in the control system (valve or controller). As the controllers used for the emitters are often proportional ones (so linear) the valve characteristics must be nonlinear. Two choices are possible a linear valve with low authority or an equal percentage valve with high authority. In practice an equal percentage valve with an authority between 0.3 and 0.7 is preferred.

iii. Pump

A pump is a hydraulic device that enables the circulation of fluids. In the building side hydronic network only, centrifugal pumps are used. A centrifugal pump accelerates the fluid by imparting a rotational movement and therefore a certain amount of hydraulic power. The pump must be sized to supply the required maximal flow rate and at the same time to overcome the total pressure losses of the network. The electrical power consumed by the pump is expressed as:

$$P_{pump} = \frac{\dot{V}\,\Delta p_c}{\eta}$$

With P_{pump} the electrical power consumed by the pump (W), $\dot{V}$ the volumetric flow rate (m^3/s), Δp_c the total pressure losses (Pa) and η the pump yield.
On the building side network, three types of pumps are used (see Figure 54):

- Pump with constant flow rate
- Pump with variable flow rate and constant speed rotation
- Pump with variable speed rotation

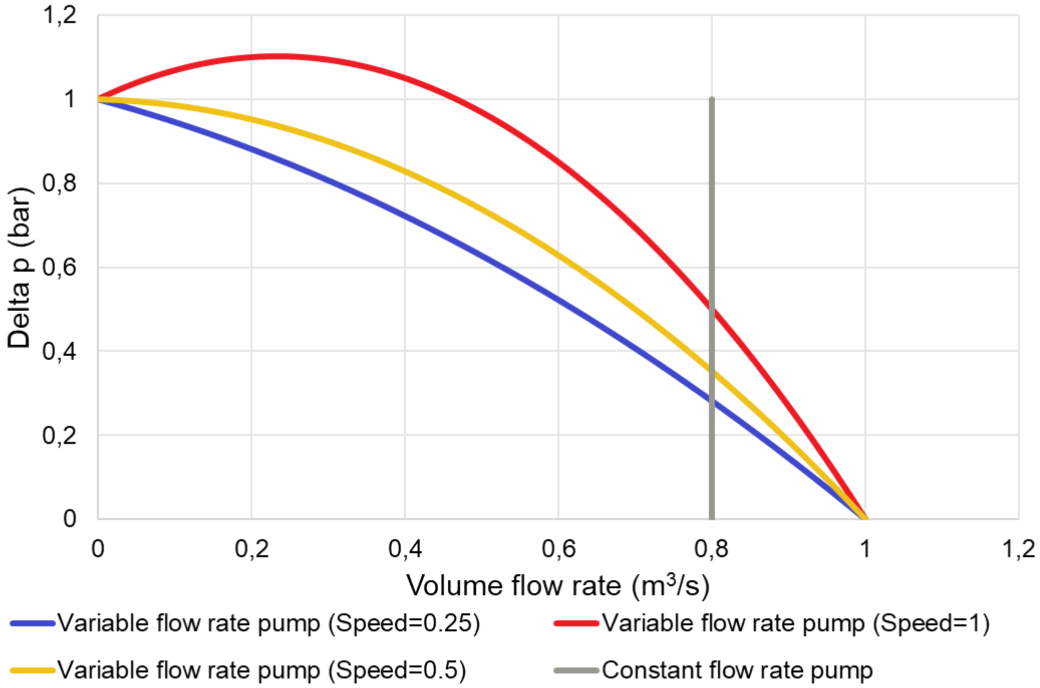


Figure 54 Example of pump characteristics

A constant flow rate pump supplies the same flow rate to the network regardless of the pressure losses to overcome. The rotational speed of the pump is adapted to supply the same flow rate at any time. This system is still used in building's side distribution network coupled with 3-way valves. Indeed, this association is easy to install and balance but leads to poor hydraulic performance of the pump.

A variable flow rate pump at constant speed pump operates on a unique characteristic curve where the pressure losses Δp_c depends on the flow rate through a polynomial equation such as:

$$\Delta p_c = A\,\dot{m}^2 + B\,\dot{m} + C$$

Where A, B and C are parameters given by the pump's manufacturer. The whole network will operate at the intersection of the pump curve and the network curve (the evolution of all the pressure losses linear and singular according to the flow rate) (see Figure 55)).

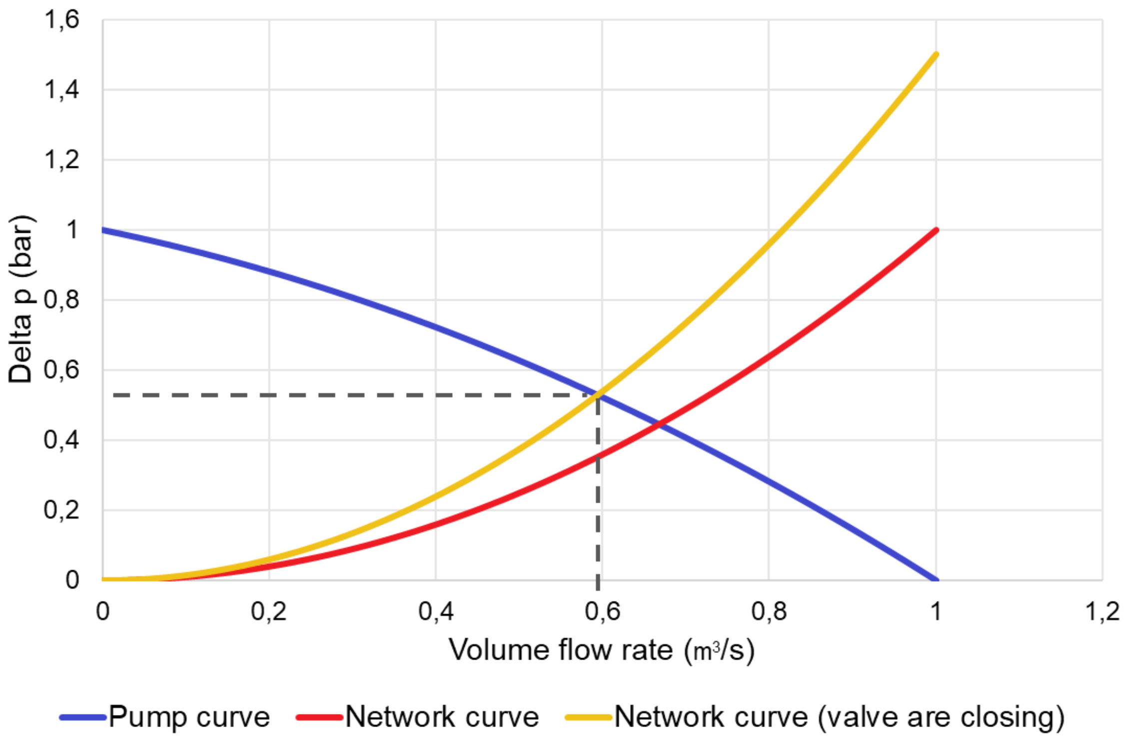


Figure 55 Example of pump & network characteristics

To improve the efficiency of the system, a variable speed pump can be installed. In first approximation in every pump, the flow rate is proportional to the speed rotation, the pressure losses are proportional to the square of the speed rotation, so the electrical power is proportional to the cube of the rotational speed. Moreover, the pump yield depends on the rotational speed. Hence, modifying the rotational speed of a pump can lead to an improvement of the pump efficiency for the same hydraulic performance (see Figure 56). In fact, the pump can operate following different polynomial curves as defined before (see Figure 56).

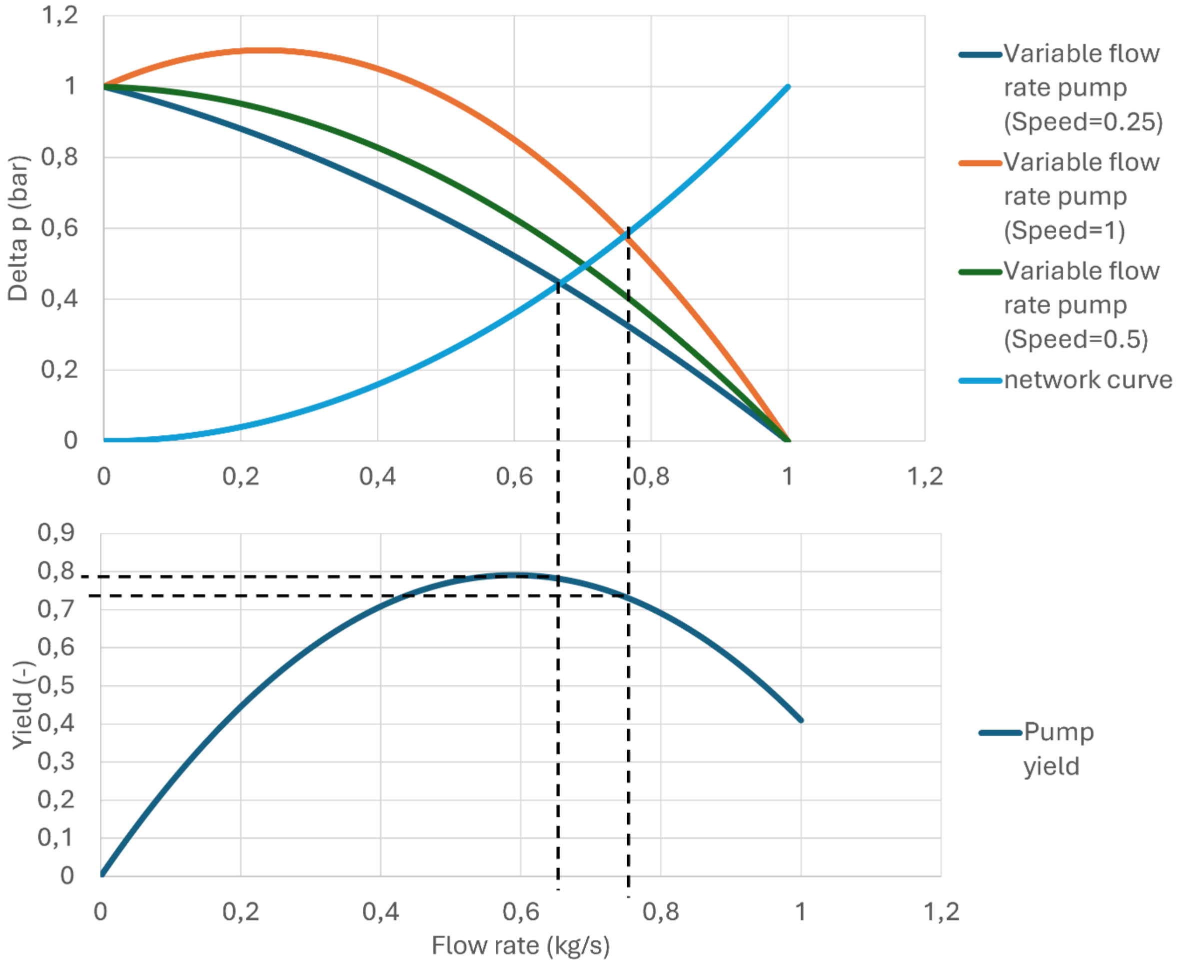


Figure 56 Example of the pump speed effect on the yield

5. Substation and control

This part deals with the substation and its control. The composition and the operation of the substation are described as well as the most common architecture of substations.

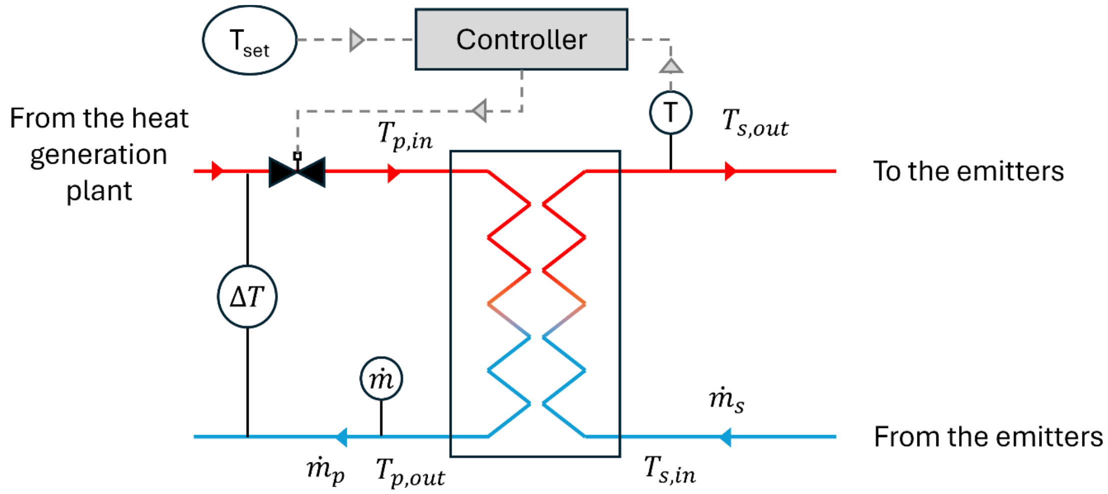


Figure 57 Descriptive scheme of a substation

The substation is the element at the interface between the primary and secondary network. It enables the two networks to be hydraulically separated. Hence the primary and secondary networks can operate at different temperatures and pressure levels. The substation is equipped to control the power and the temperature supply to the secondary network and to evaluate the thermal consumption. A substation is composed of 4 key elements (see Figure 57):

- One or two heat exchangers …
- A control valve on the primary side to monitor the flow rate
- A secondary temperature sensor and a controller to monitor the valve
- A heat meter to measure thermal consumption

a. Heat exchanger

In a substation, heat exchangers are the key elements. Indeed, they allow to separate hydraulically the primary and secondary networks and to exchange to required power. One or several heat exchangers can be used in a single substation to supply space heating, domestic hot water or to improve the performance of all systems. Indeed, the efficiency of the heat exchangers is the principal parameter to take into consideration to reduce the primary return temperature and so to improve the district heating network performances. Usually, plate heat exchangers (see Figure 58) are used in counterflow configuration because they are more compact and more efficient than other types of heat exchangers.

Figure 58 Example of a plate heat exchanger used in substations

A heat exchanger is supposed to have a perfect yield i.e. all the power given off by one fluid is recovered by the other. Hence the power exchanged by a heat exchanger is written as:

$$\dot{Q} = \dot{m}_p Cp \left(T_{p,in} - T_{p,out}\right) = \dot{m}_s \, Cp\left(T_{s,out} - T_{s,in}\right)$$

A heat exchanger is sized to be the most efficient possible i.e. to have an efficiency close to 1. Efficiency E is defined as:

$$E = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{\min(\ \dot{m}_p Cp\ ;\ \dot{m}_s Cp)\, \Delta T_{max}}$$

Where $\dot{Q}_{max}$ (W) is the maximum power the heat exchanger can exchange if the exchanged surface is infinite and ΔT_{max} (°C) the largest temperature difference between the two fluids. Consequently, the efficiency can be expressed as:

$$E = \frac{\dot{Q}}{\min(\ \dot{m}_p\ Cp\ ; \dot{m}_s\ Cp\ )\ \left(T_{p,in} - T_{s,in}\right)}$$

In design condition, the aim is to have the highest efficiency possible typically around 0.95-0.9. Indeed, the closer the efficiency is to one the smaller the pinch is. The pinch represents the smallest temperature difference between the two fluids. Usually in counter flow heat exchangers used in DH networks the pinch is the difference between the temperature at the outlet of the primary side and the inlet temperature at the secondary side.

$$Pinch = T_{p,out} - T_{s,in}$$

Usually, the DH network operators design the heat exchanger to have a pinch equal to 1 (i.e. to have the primary return temperature as low as possible). As the secondary temperatures are imposed by the emitters and so perfectly known, assuming a pinch equal to 1, a primary supply temperature set by the operator and the maximum power needed, the heat transfer coefficient UA can be calculated through:

$$\dot{Q} = UA\,\Delta TLM$$

$$\Delta TLM = \frac{T_a - T_b}{ln(T_a/T_b)}$$

$$T_a = max\left(T_{p,in} - T_{s,out}; T_{p,out} - T_{s,in}\right)$$

$$T_b = min\left(T_{p,in} - T_{s,out}; T_{p,out} - T_{s,in}\right)$$

The surface heat transfer coefficient U can be set thanks to the Dittus-Boettler correlation:

$$U = \left(\frac{1}{h_p} + \frac{e}{\lambda_a} + \frac{1}{h_s}\right)^{-1}$$

$$Nu = 0.023\,Re^{0.8}\,Pr^{0.4} = \frac{hd_h}{\lambda_w}$$

$$Re = \frac{\rho\,v\,d_h}{\mu} \qquad Pr = \frac{Cp\,\mu}{\lambda_w}$$

With h_p and h_s respectively the convective heat transfer coefficient of the primary and the secondary sides (W/K/m²), e the thickness of the plate (m), λ_a and λ_w the thermal conductivity respectively of the plate and the water (W/K/m), Nu the Nusselt number, Re the Reynolds number, Pr the Prandtl number, d_h the hydraulic diameter (m) (here the space between two plates), ρ the density (kg/m3), v the velocity (m/s) and μ the dynamic viscosity (Pa.s)

b. Control system

During operation, the power and the secondary supply temperature required are not constant. Indeed, it evolves generally according to the outdoor temperature. Hence, a constant mass flow rate at the primary side will imply an augmentation of the primary supply temperature as the required power declines (for the same primary supply temperature). This effect is to be avoided to not degrade the overall performance of the DH network. Consequently, the primary mass flow rate must be monitored according to the secondary side's needs. This is made possible by driving a control valve located on the heat exchanger primary side through a controller, supply temperature setpoint curve and a sensor measuring the secondary supply temperature (see Figure 59 & Figure 60).

Figure 59 Control valve [12]

Figure 60 Temperature sensor [13]

According to the difference between the measured and the setpoint secondary supply temperature, the controller orders the closing or the opening of the control valve.

c. Energy meter

An energy meter is a component used to measure the heat consumption of a substation (see Figure 61).

Figure 61 Energy meter [14]

The energy meter is composed of 2 temperature sensors placed respectively on the primary side inflow and outflow, a fluid flow meter and a calculator.

$$\dot{Q} = \int_0^t \rho Cp \, \Delta T \, dV$$

This device is used to bill heat to customers. Hence, the accuracy of the sensors must be controlled. According to the European standard EN 1434-1 [15] the maximum measurement uncertainties associated with the sensors are for energy meter of class 2 (for household use and outdoor installation):

$$E_f = \pm \left(2 + 0{,}02 \, \frac{q_{max}}{q}\right) \text{ in } \% \;\; for\ the\ flow\ meter\ of\ class\ 2$$

$$E_t = \pm \left(0{,}5 + 3 \, \frac{\Delta T_{min}}{\Delta T}\right) \text{ in } \% \; for\ the\ temperature\ sensors$$

d. Substation architecture

Once all the components are designed, the global substation architecture must be set. As a substation commonly supply space heating and domestic hot water at least two heat exchangers are used in a single substation. Hence different substation hydronic architectures are widely used.

i. The one heat exchanger configuration

The one heat exchanger used two heat exchangers but only one in the interface between the primary side and the secondary side. A scheme of this architecture is presented in Figure 62.

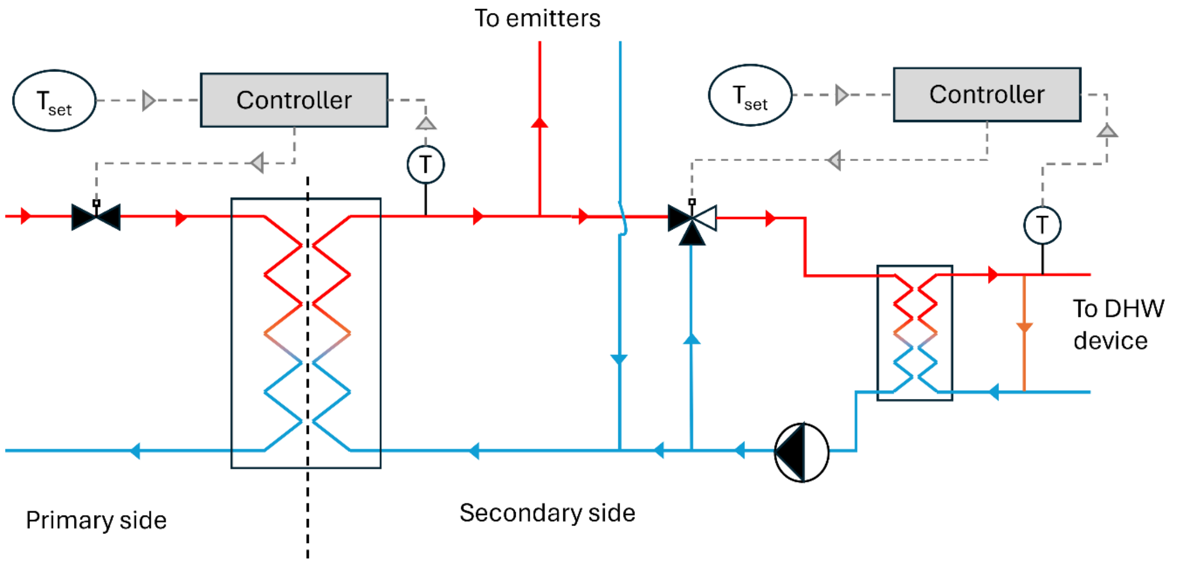


Figure 62 Scheme of a substation in one heat exchanger configuration

This configuration is not widely used and is less efficient than the other configurations presented below. Indeed, despite its simplicity this configuration is supplied with a temperature always equal or superior to 60°C to supply DHW. That's led to higher primary return temperatures (see Figure 63)

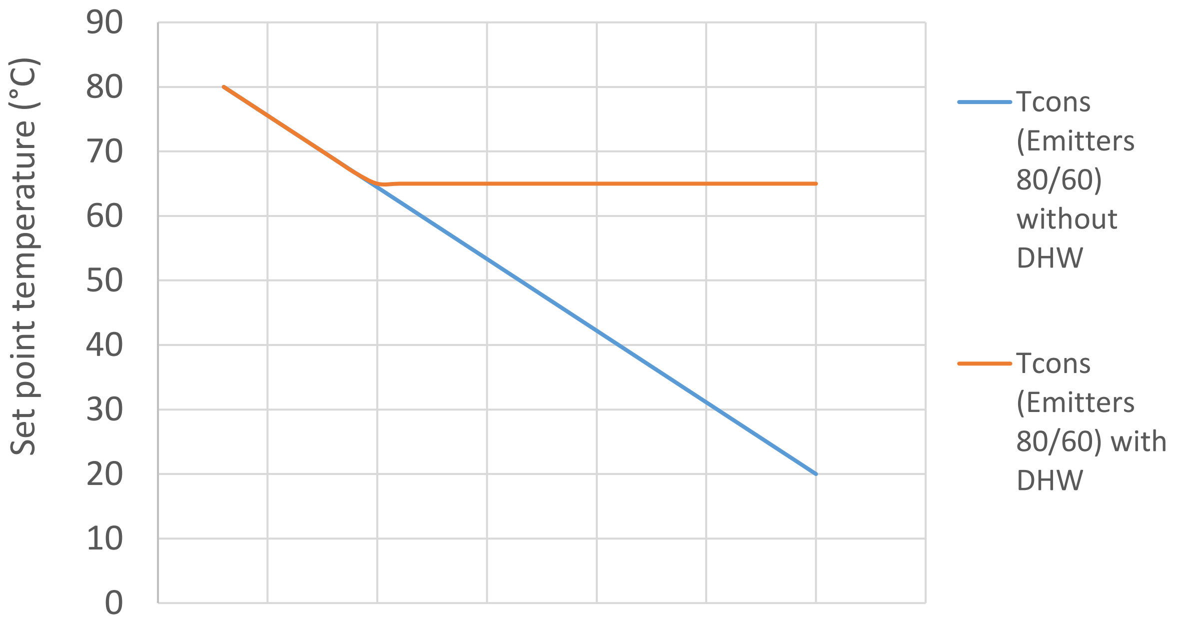


Figure 63 Secondary supply temperature setpoint according to the outdoor temperature for the one heat exchanger substation

ii. The parallel configuration

A substation in parallel configuration is the most widely used architecture. Two heat exchangers operate respectively for space heating and DHW, they share the same supply line at the same temperature (see Figure 64).

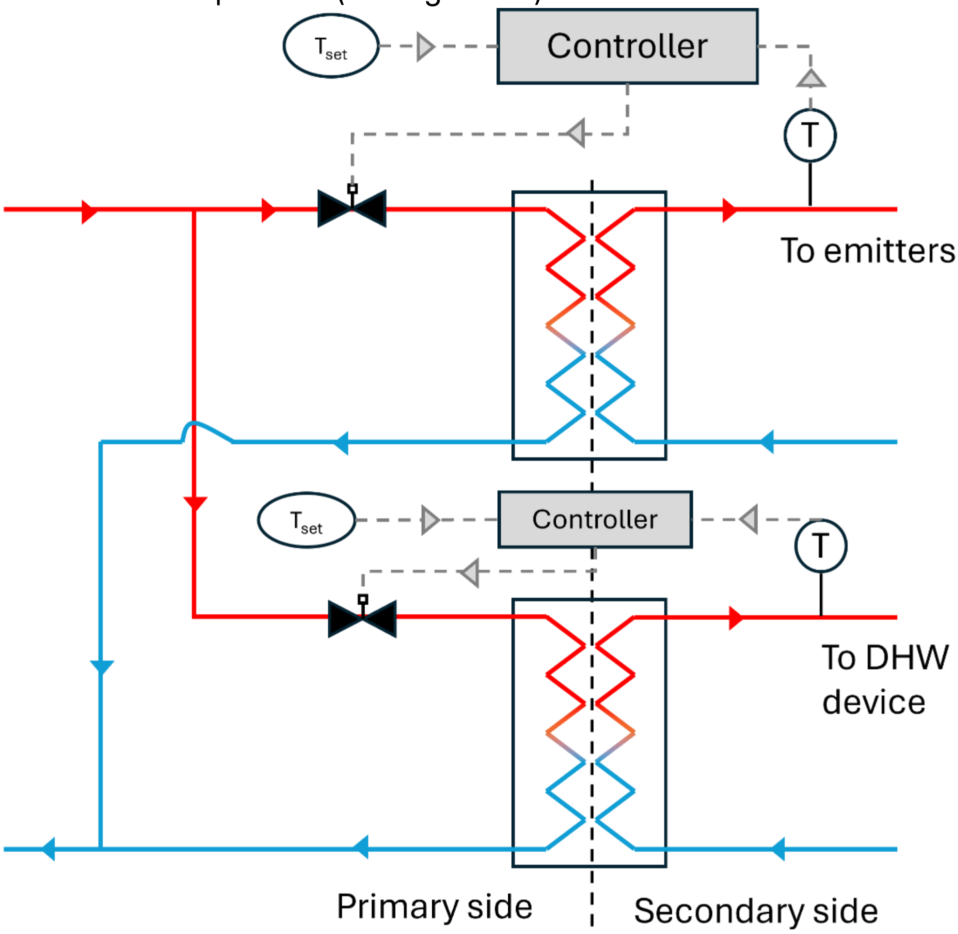


Figure 64 Scheme of a substation in parallel configuration

Each heat exchanger is coupled with its own control system operating with different supply setpoint secondary temperatures (see Figure 65)

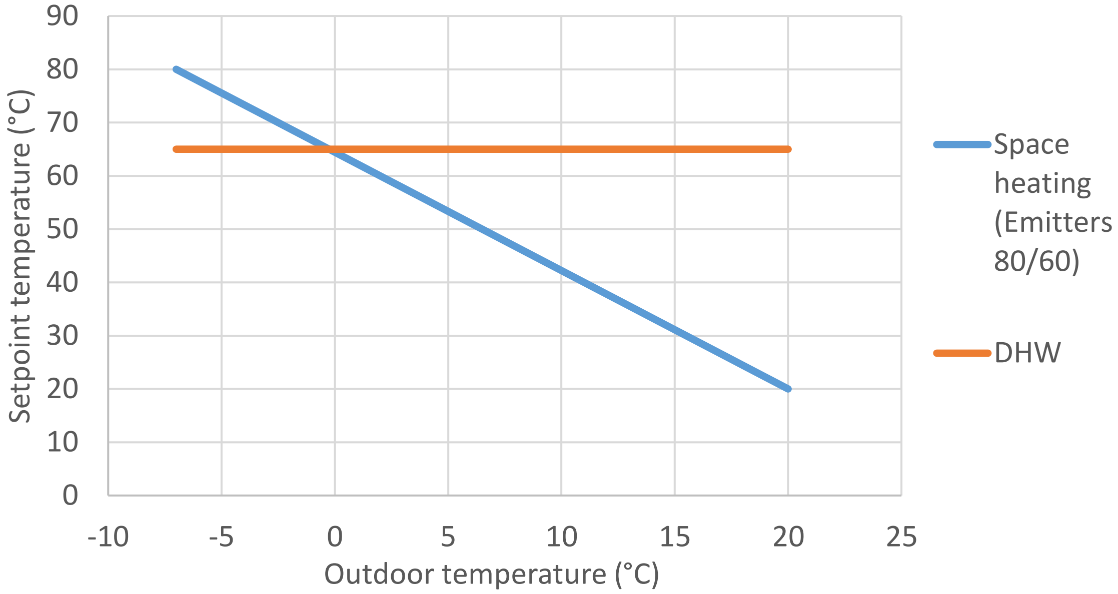


Figure 65 Secondary supply temperature setpoint according to the outdoor temperature for the parallel substation configuration

This configuration is simple to control, to install and is not expensive..

iii. The series configuration

A substation in series configuration is composed of two heat exchangers respectively for space heating and DHW. This configuration enables to use the primary return temperature of one heat exchanger to supply completely or partially the needs of the other heat exchanger. (see Figure 66).

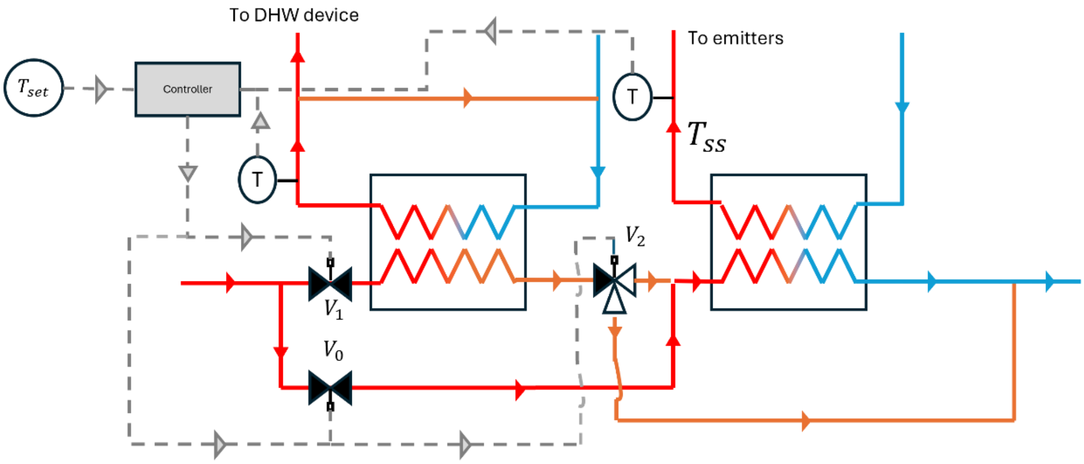


Figure 66 Scheme of a substation in series configuration

Depending on the temperature needed for space heating, the DHW heat exchanger supplies or is supplied by the space heating heat exchanger. Indeed, the DHW supply temperature is around 60-55°C. So, if the required temperature of the space heating emitters is inferior to the DHW supply temperature, the primary return temperature of the DHW heat exchanger is able to supply the space heating one. On the contrary, if the required temperature of the space heating emitters is higher than the DHW one, the space heating heat exchanger return temperature can supply the DHW one.

Based on the scheme presented on Figure 67, the series configuration operates as follows:

- Simultaneous DHW and space heating demand:

- If $T_{ss} < T_{ss,set}$ the valve V_1 is open, firstly the valve V_2 opens (the space heating heat exchanger is supplied), if it is not sufficient the valve V_0 opens
- If $T_{ss} > T_{ss,set}$ the valve V_1 is open, firstly the valve V_0 closes, if it is not sufficient the valve V_2 closed (the space heating heat exchanger is bypassed)

- Only space heating demand: the valve V_1 is fully closed, the valve V_0 opens
- Only DHW demand: the valve V_0 is fully closed, the valve V_1 opens

The series configuration is more complex to install because of the complexity of the control system.

iv. The two-stage configuration

A substation in two-stage configuration is composed of three heat exchangers, two in parallel respectively for space heating and DHW and a third one connected to the return line of the two other heat exchangers used to preheat the DHW. This configuration enables to lower the overall primary return temperature and so to improve the efficiency of the DH network.

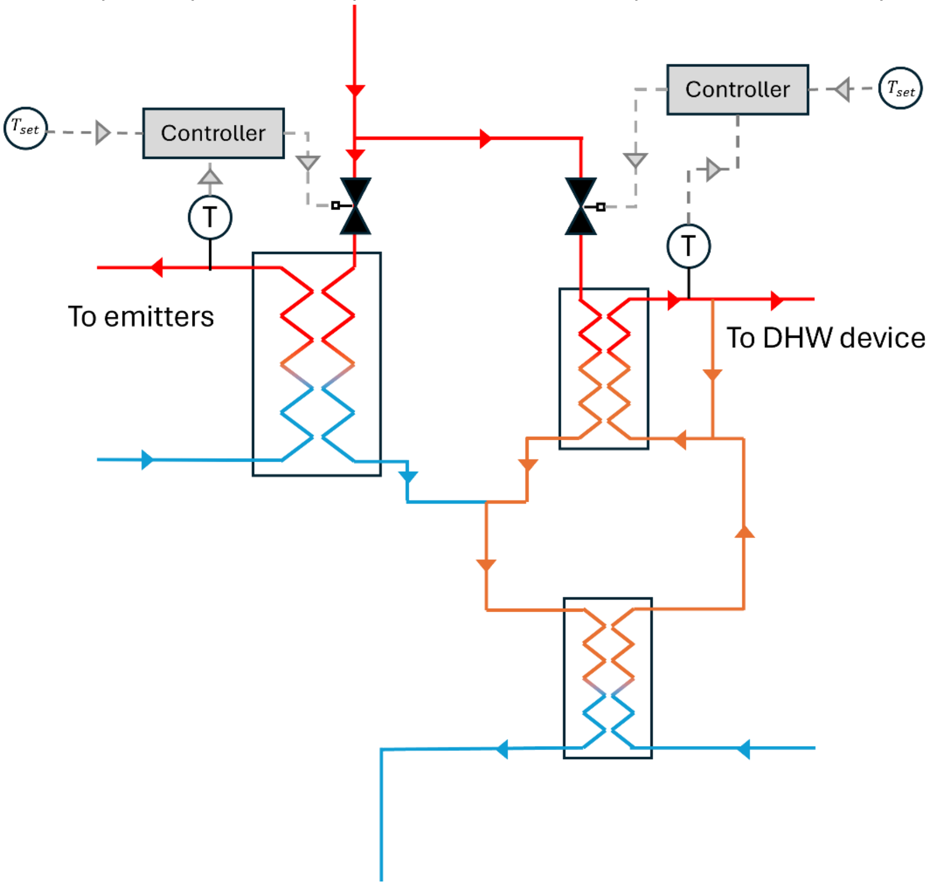


Figure 67 Scheme of a substation in 2-stageconfiguration

The two-stage configuration benefits the advantages of the parallel configuration and the series configuration. Indeed, this configuration is easy to install and control and reduces the primary return temperature. However, the parallel configuration is generally preferred, as the efficiency gain does not justify the additional cost associated with using a third heat exchanger.

6. Primary network

Once the secondary network and the substation are designed, the primary network must be sized. This section aims to give information about the different possibilities to build the primary network and how to design the pipes.

a. Heat load curves

A DHC network is often located in urban areas and is connected to different types of buildings such as residential buildings, schools, offices, shops, hospital, … The space heating consumption of each type of building differs by the occupation hours, DHW use, peak consumption, …

Residential building heat consumption is characterized by a daily occupation, a non-negligible DHW consumption and two consumption peaks, one in the morning and the other in the evening (typical load profile in Ile de France in a cold winter week) (see Figure 68).

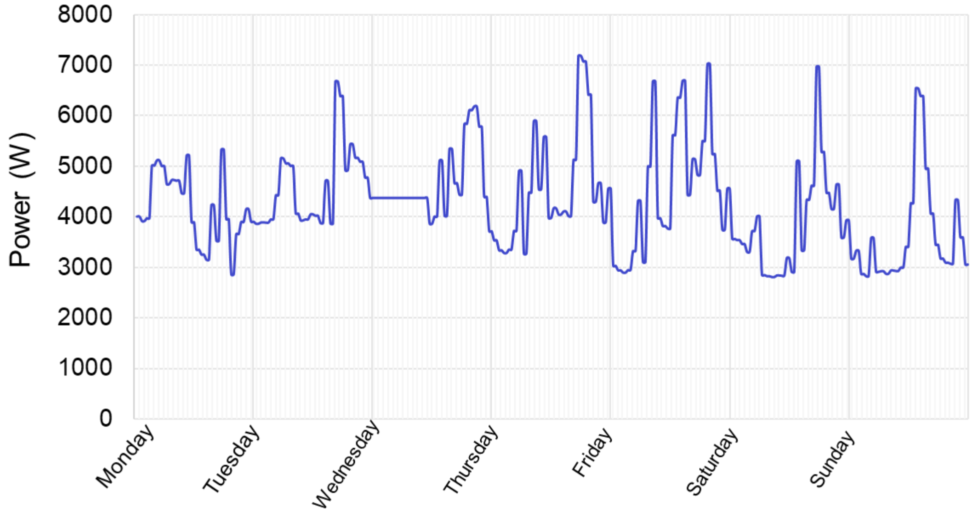


Figure 68 Typical residential building consumption pattern

School heat consumption is characterized by an occupation only during work time and a morning consumption peak due to night setback control (reduction of the indoor set point temperature during the inoccupation period (see Figure 69).

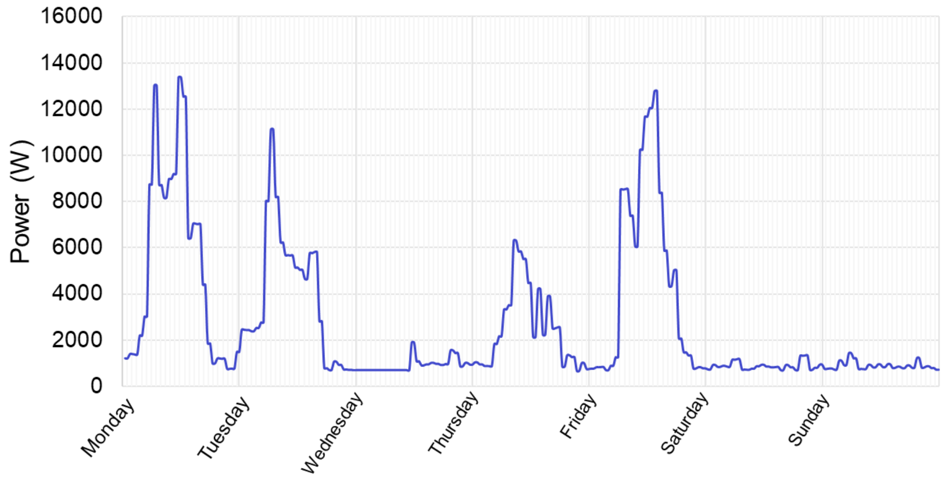


Figure 69 Typical school consumption pattern

Gymnasium heat consumption is characterized by an occupation only during days, a morning and early afternoon consumption peak and an important DHW consumption (see Figure 70).

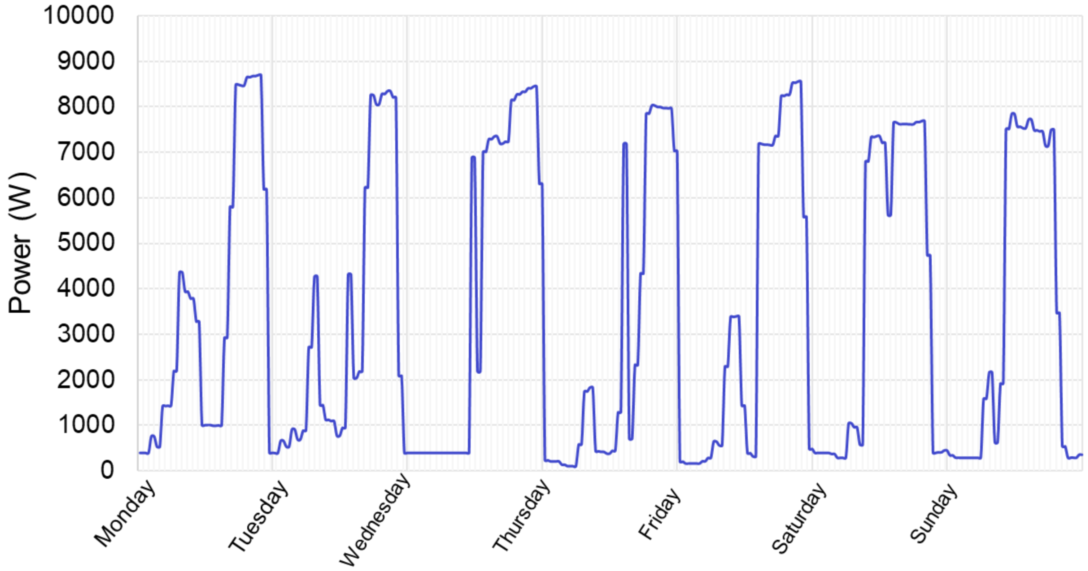


Figure 70 Typical gymnasium consumption pattern

Office buildings heat consumption is characterized by an occupation only during workdays and a morning consumption peak due to night setback control (see Figure 71).

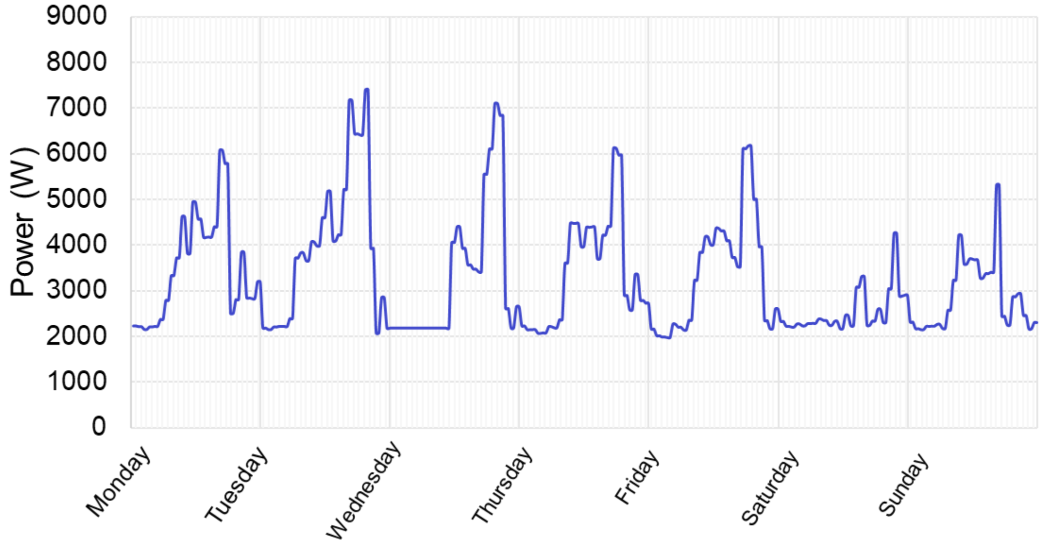


Figure 71 Typical office building consumption pattern with a weekend set back

The difference in heat load patterns implies that the consumption peak in all the substations in the DHC networks does not happen at the same time. Hence the maximum total power needed by all networks is inferior to the maximum power of each substation. This ratio is called the simultaneity factor S:

$$S = \frac{\max \dot{Q}_{tot}}{\sum_i \max \dot{Q}_i}$$

In concrete terms, a low simultaneity factor reduces the peak power that needs to be covered by the DHC network. This means that the size and the cost of the heat generation systems can be reduced (in comparison to individual systems) while improving the performance and sustainability of the heating network.

To size the heat generation systems, a classical tool is used called the heat duration curve. This tool allows to calculate the number of hours per year that the demand exceeds a certain power level. With a heat duration curve (see Figure 72) the peak demand (few hours, high power) can be distinguished from the base load (lower power, many hours). Usually in DHC networks the peak demand is ensured by gas boilers or backup units and the base load by renewables or waste heat. Hence this tool makes it easier to choose the best energy mix with the most efficient and cheap systems according to the power required and the operating hours.

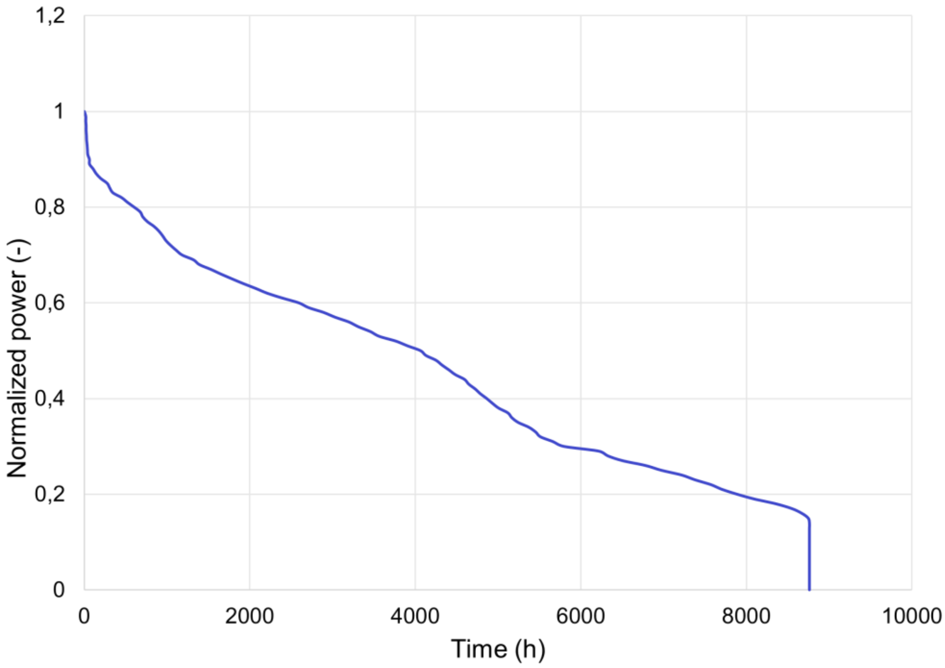


Figure 72 Example of heat load curve

b. Primary pipes

The basic function of primary pipes carries the hot water from the heat production plant to the substations and carries cold water in the other way. Nevertheless, according to the strategies the conception of the primary network can be more complicated by using several pipes with different temperature levels.

Usually, the basic strategy with only two pipes is used to provide DHW and space heating. These pipes are set in parallel, one for the hot supply and one for the warm return (see Figure 73). This is the cheapest and simplest structure. The supply temperature is set to reach the highest temperature needed in a substation. If the DHC networks are also connected to substations with a need of a lower temperature, the substation will always be correctly provided but the performance of the network will not take profit of low return temperatures.

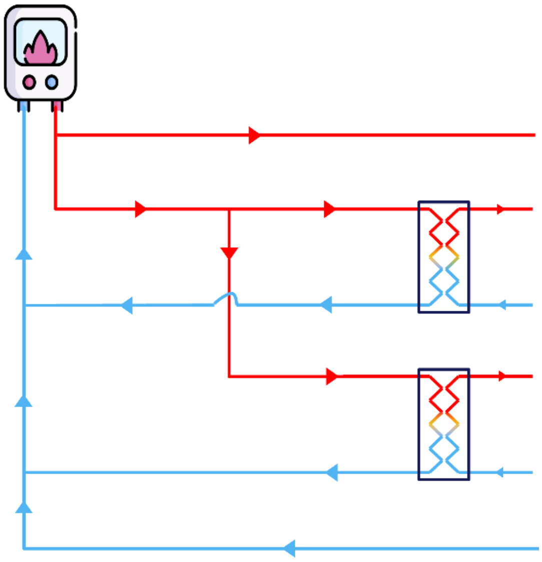


Figure 73 Two pipes configuration

To improve the DHC network performance, a third pipe set in parallel can be added to the previous scheme to supply heat at a different temperatures. For example, one pipe to supply space heating or DHW and another separate pipe to supply substation only connected to building with low temperature emitters (see Figure 74). This configuration is most expensive in terms of installation but allows the heat losses through the pipes to be reduced.

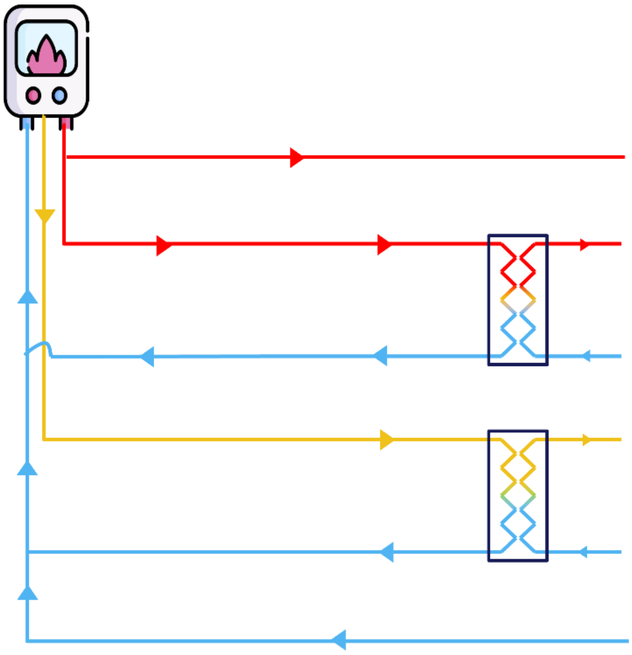


Figure 74 Three pipes configuration

For DHC networks capable of supplying space heating, DHW and space cooling a configuration with four pipes in parallel is used (see Figure 75). Here, two pipes are used for heating demand (one pipe for hot supply and one for the warm return) and the two other pipes are used to supply cooling demand, one pipe for cold supply and one for warm return). All the demand can be reached simultaneously.

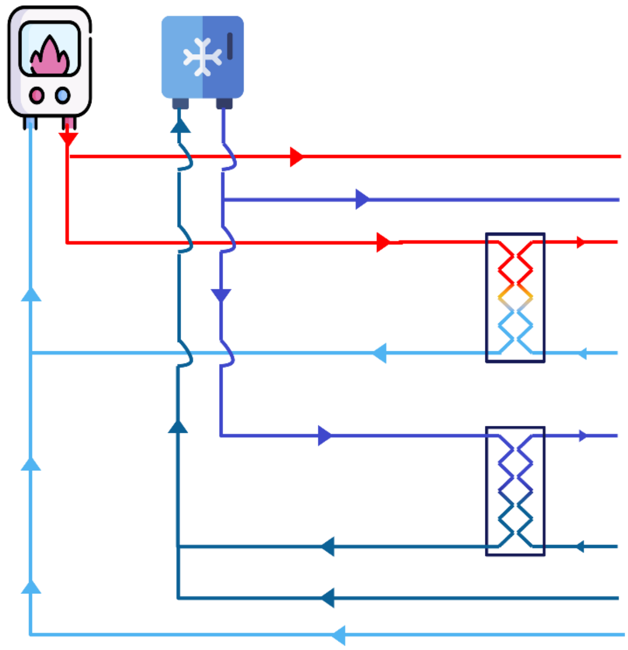


Figure 75 Four pipes configuration

Once the number of pipes is set in the DHC network, its structure must be defined. The are three possible structures: the radial structure, the looped structure and the meshed structure. The radial structure is the simplest and the more common one. It consists of linking a unique central heat generation plant and the substations (see Figure 76). Generally, the network layout follows the road layout. This structure enables to reduce the length and the investment cost for the primary network. Nevertheless, it is not suitable for long networks due to the important heat losses.

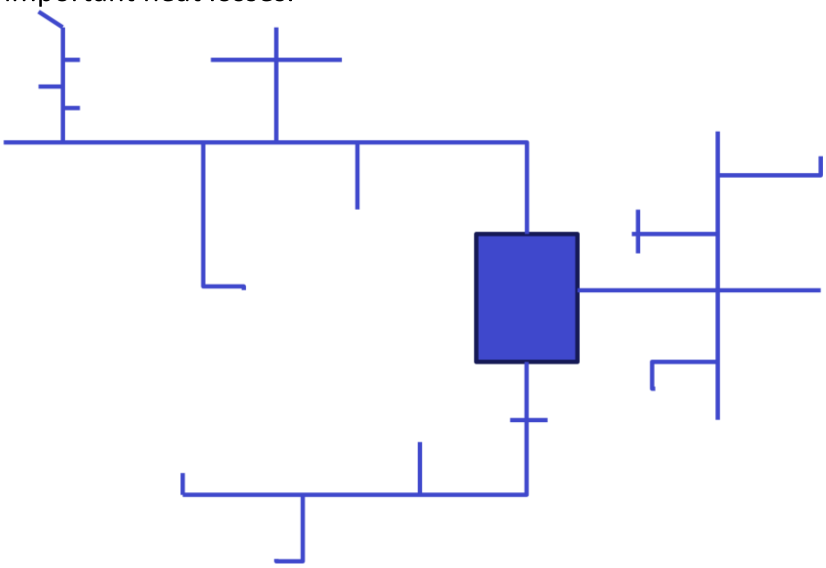


Figure 76 Radial network structure

For long networks, the looped structure is preferred. This structure consists of linking several heat generation plants to the substation by forming a loop (see Figure 77). It is more expensive to install than the radial structure, but it allows to ensure the security of the heat supply.

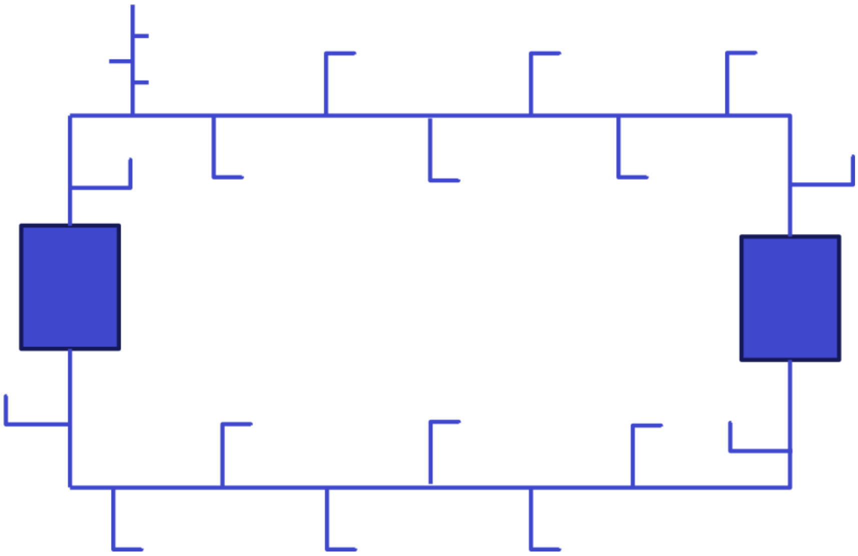


Figure 77 Looped network structure

For the long network a meshed structure can also be used. It works in the same manner as the looped structure, but several paths can be used to link a substation to a heat generation plant (see Figure 78). This structure has been used in old networks in very dense cities like Paris, Stockholm or Berlin. Nowadays it is no longer used due to its difficulties in operating.

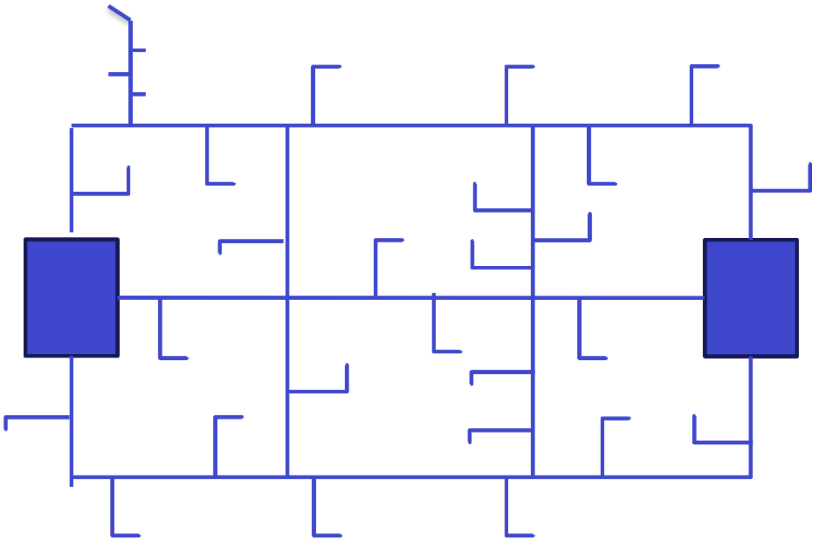


Figure 78 Meshed network structure

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