POLITECNICO DI TORINO

Master of Science program in

Mechanical Engineering

Master of Science Thesis

Underfloor Heating System

Supervisor:

Marco Masoero

Candidate:

David Espuny

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NOMENCLATURE

TEMPERATURES

PARAMETER	VALUE	UNITS	NOMENCLATURE	REGULATION
Minimum Dry-Bulb Temperature	2,4	ºC	ϑ_{o}	External_Climatic_Conditions.pdf | page 109 | IDEA [4]
Comfortable Inner Temperature	20	ºC	$\vartheta_{i,i}$	UNE_1264_2.pdf | page 7| 4 | UNE
Comfortable Bath Temperature	24	ºC	$\vartheta_{i,B}$	BOE_1027_2007.pdf | page 15 | BOE [2]
Comfortable Peripheral Temperature	20	ºC	$\vartheta_{i,P}$	BOE_1027_2007.pdf | page 15 | BOE
Maximum Inner Floor Temperature	29	ºC	$\vartheta_{F,max,i}$	UNE_1264_3.pdf | page 7| 4.1.1.3 | UNE [3]
Maximum Bath Floor Temperature	33	ºC	$\vartheta_{F,max,B}$	UNE_1264_3.pdf | page 7| 4.1.1.3 | UNE
Maximum Peripheral Floor Temperature	35	ºC	$\vartheta_{F,max,P}$	UNE_1264_3.pdf | page 7| 4.1.1.3 | UNE
Non Heating Room Temperature	12	ºC	ϑ_{U}	
Inner Temperature Rise	17,6	ºC	$\Delta\vartheta_{i}$	
Bath Temperature Rise	21,6	ºC	$\Delta\vartheta_{B}$	
Peripheral Temperature Rise	21,6	ºC	$\Delta\vartheta_{P}$	
Inner-Peripheral Temperature Rise	8,8	ºC	$\Delta\vartheta_{I-P}$	
Bath-Peripheral Temperature Rise	10,8	ºC	$\Delta\vartheta_{B-P}$	
Inner-Bath Temperature Rise	0	ºC	$\Delta\vartheta_{I-B}$	Is supposed that the temperature jump between two air-conditioned rooms are null

THERMAL TRANSMITTANCE

PARAMETER	VALUE (S)	VALUE(A)	UNITS	NOMENCLATURE	REGULATION
wall	0,86	0,25	$W/m^2 \cdot K$	U_w	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR [1]
inner walls	1,2	0,5	$W/m^2 \cdot K$	U_{iw}	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR
floor	0,64	0,2	$W/m^2 \cdot K$	U_f	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR
crystal	3,5	1,2	$W/m^2 \cdot K$	U_c	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR
door	1,5	1,5	$W/m^2 \cdot K$	U_d	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR
ceiling	0,49	0,15	$W/m^2 \cdot K$	U	Manual_Tecnico_Suelo_Radiante.pdf | page 77 | UPONOR

CHARACTERISTIC CURVES

PARAMETER	VALUE	UNITS	NOMENCLATURE	REGULATION
Tube Distance		m	T	UNE 1264 2 | page 8 | 6.1 | UNE
Above Layer Tube Thickness	0,045	m	S_u	UNE 1264 2 | page 9 | 6.2 | UNE
Floor thermal resistance	0,05	$m^2 \cdot K/W$	$R_{\lambda,B}$	UNE 1264 2 | page 8 | 6.1 | UNE
Resistencia superior	1,25	$m^2 \cdot K/W$		
Tube Diameter	0,016	m	D	UNE 1264 2 | page 8 | 6.1 | UNE
Tub Thickness	0,002	m	S_R	UNE 1264 2 | page 9 | 6.2 | UNE
Average Air-Water Temperature		ºC	$\Delta\vartheta_H$	UNE 1264 2 | page 8 | 6.1 | UNE
System Characteristic Rate	6,7	$W/m^2 \cdot K$	B	UNE 1264 2 | page 9 | 6.2 | UNE
Thermal Transmission Equivalent Rate		$W/m^2 \cdot K$	K_H	UNE 1264 2 | page 9 | 6.2 | UNE
Tube Thermal Conductivity	0,35	$W/m \cdot K$	λ_R	UNE 1264 2 | page 9 | 6.2 | UNE
Above Layer Tube Conductivity	1,2	$W/m \cdot K$	λ_E	UNE 1264 2 | page 9 | 6.2 | UNE
Additional Thermal Transmission	10,8	$W/m^2 \cdot K$	α	UNE 1264 5 | page 11 | Table A.1 | UNE
Fixed Conductivity	1	$W/m \cdot K$	$\lambda_{u,0}$	UNE 1264 2 | page 9 | 6.2 | UNE

WATER FLOW RATE

PARAMETER	VALUE	UNITS	NOMENCLATURE	REGULATION
Water Heat Capacity	4184	$J/kg \cdot K$	C_W	UNE 1264 3| page 10 | 4.1.3.3 | UNE
Underfloor Heating Panel Thermal Resistance	0,17	$m^2 \cdot K/W$	$R_{\alpha,roof}$	UNE 1264 3| page 11 | 4.1.3.3 | UNE
Insulator Thermal Resistance	1,25	$m^2 \cdot K/W$	$R_{\lambda,ins}$	UNE 1264 3| page 8 | 4.1.2.2 | UNE
Plaster Thermal Resistance	0,06	$m^2 \cdot K/W$	$R_{\lambda,plaster}$	Webgraphy 1 | Javier Ponce Formación | Example 2
Roof Thermal Resistance	0,24	$m^2 \cdot K/W$	$R_{\lambda,roof}$	Webgraphy 1 | Javier Ponce Formación | Example 2

PRESSURE DROP

PARAMETER	VALUE	UNITS	NOMENCLATURE	REGULATION
Water Density	1000	kg/m^3	ρ_W	
Dynamic Viscosity	0,000891	kg/m·s	μ	
Relative Roughness	0,00044		ε/D	Moody's Diagram
Gravity Acceleration	9,81	m^2/s	g	

NATURAL GAS

PARAMETER	VALUE	UNITS	NOMENCLATURE	REGULATION
Lower Power Capacity	12,53	kW·h/kg	PCI_{NG}	
Natural Gas Cost	0,55	$€/m^3$	C_{NG}	
Natural Gas Density	0,75	kg/m^3	ρ_{NG}	

ABSTRACT

My first idea about my thesis was about doing a system that can save energy to an old house. In order to do this, I would choose and old house, situated in my hometown and I would compute all calculations and needs to achieve a good Underfloor Heating and Cooling system. How?

On the one hand, I would explain how these systems work. I would try to clarify all the components and its behavior in an underfloor system (type of system, underfloor layers, covers, etc. This would be the theoretical part of my thesis.

On the other hand, in order to compute all the system, I would provide a building description. I would realize all the layouts to understand and explain better how it is. (Building dimensions, rooms and enclosures, glass positions, glass dimensions…). Then I would develop and study of winter and summer thermal loads which would make me capable to go further to dimensioning. Once I get this, I would size all the system. I would compute coils –with its characteristic curves-, design of the fluid temperature, water discharge, circuit length and manifolds sizing, pressure drop and finally, boiler and its performance.

To do this, the thesis will be regulated by Spanish regulations because the house is in Spain. To this end, the requirements of current regulations have been met: the Building Technical Code (CTE), the Regulation for Thermal Installations (RITE) and the radiant floor design regulations (UNE-EN-1264).

1. UNDERFLOOR HEATING SYSTEM

1.1. INTRODUCTION

The aim of the air conditioning of a closed enclosure is to achieve a certain temperature by heating or cooling said enclosure. The heating of enclosed zones has been taking place for hundreds of years, while the use of cooling machines is much more modern. A heating or cooling system can be divided into three parts:

- Generation system: It is the energy source of the system. For heating, the heat sources can be solar panels, boilers, heat pumps, while cooling machines use mechanical compression or absorption chillers.
- Distribution system: Through a network of pipes, ducts, pumps, etc. A heat transfer fluid is circulated to the enclosure to be heated.
- Emission system: The emitters absorb or transfer energy to the room to be heated, achieving the comfort temperature. Some examples of emitters are underfloor heating floor, radiators, fan coils, etc.

1.2. UNDERFLOOR HEATING

Since the Romans began to heat closed spaces, circulating under the pavement fumes produced by the burning of coal or wood in a fireplace or oven so that the heat radiated to the pavement. This type of heating is called a hypocaust.

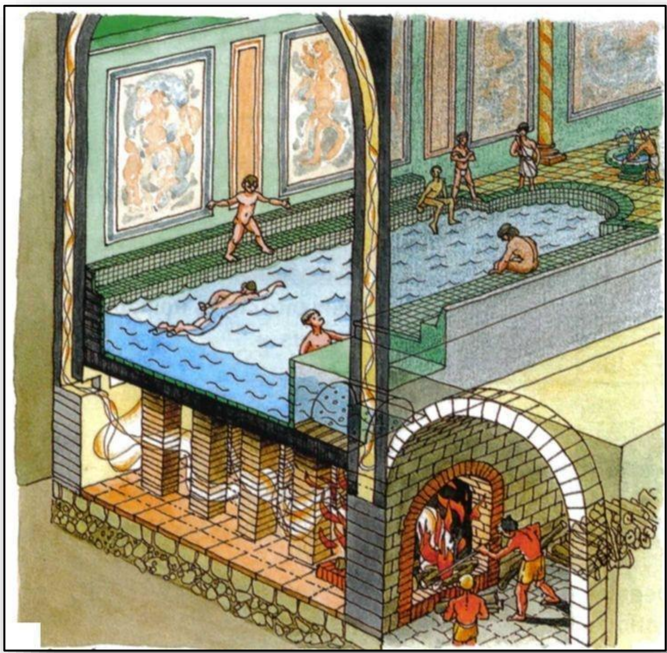


Figure 1. Hypocaust heating system

This system has been evolving over time to become the current systems of Invisible Air Conditioning by radiant floor. In the 30s these systems had steel pipes, while in the 60s and 70s these pipes were made of copper, there was a significant risk of corrosion and water leakage over time, preventing these systems from becoming in a standard.

In order to guarantee the durability and correct operation over time of the radiant floor systems, pipes were developed from plastics, this being the perfect solution unlike its predecessors. This type of pipes have been designed to overcome all the inconveniences that arise when in this type of installations pipes of different materials are used:

- Deterioration.
- Reduction of the flow through depositions.
- Noise through the passage of water.

The basic principle of the traditional system consists of the water impulsion at medium temperature (around 40 ºC in winter and 14 ºC in summer) through circuits of plastic pipes (cross-linked polyethylene, PB, etc.) with barrier against oxygen diffusion. These circuits are embedded in a layer of cement mortar, on which a final pavement of ceramic type, stone, parquet, etc. is placed.

In winter, the mortar absorbs the heat dissipated by the pipes and gives it to the upper pavement, which, in turn, emits this energy towards the walls and ceiling of the room by means of radiation, and to a lesser degree by natural convection.

On the other hand, in summer, the pavement absorbs heat by radiation and partly by convection, from the walls and the roof, transmitting to the mortar layer and the radiant floor pipe, transporting heat through the water to the outside of the floor living place.

1.2.1. System components

In the present project, many components will be studied. In order to familiarizing with them in Figure 1.2 is showed a schematic of the components of the radiant floor installation.

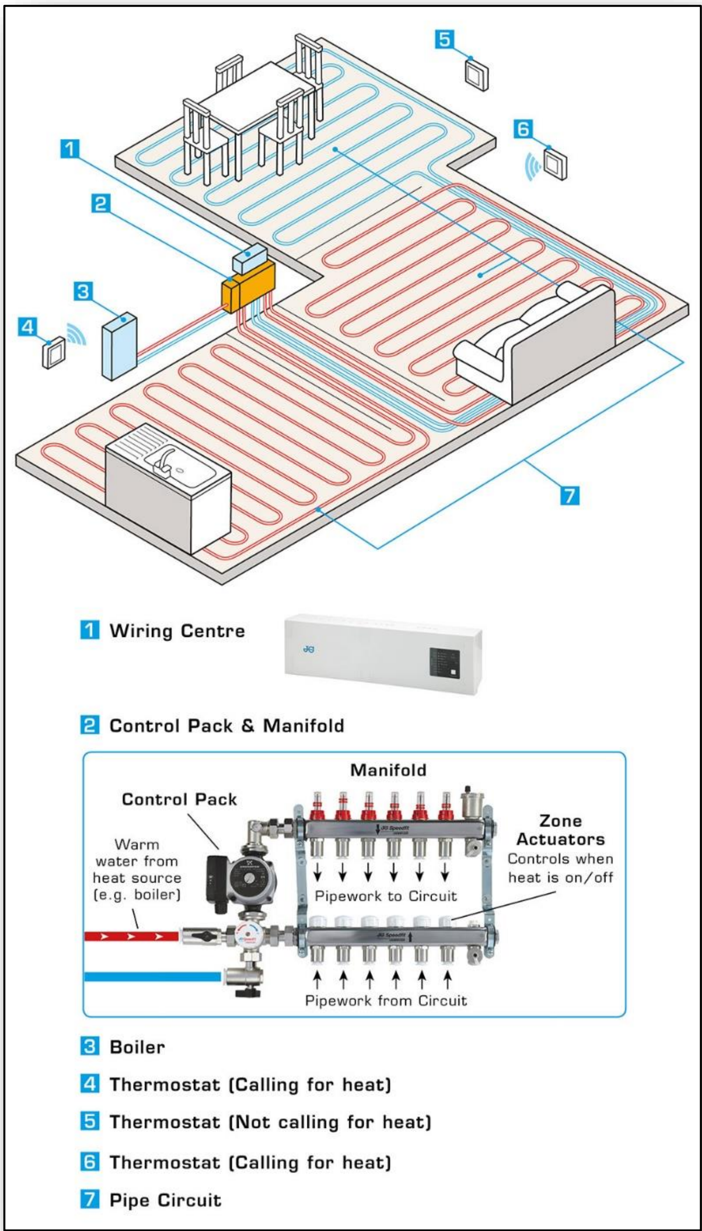


Figure 2. System Components

Following the previous schema, these are the main components:

- Boiler: although for underfloor heating applications, air-water heat pumps are used and are the most energy efficient systems on the market, as well as being able to heat heat transfer fluid according to the needs of the system, in this project, is considered a natural gas boiler. The only reason is because there is a natural gas installation in the original house and it is not within the scope of the project to reengineer the generator system. So, due to the radiant floor system works at water flow temperatures in winter between 35 and 45ºC, it makes these machines suitable for these air conditioning systems.

- Distribution: a hydraulic pump through a network of pipes drives the heat transfer fluid where it is distributed to each room by manifolds (Figure 1.3.) The manifold assembly incorporates a series of elements for its correct operation:
    - Drains to extract the air contained in the pipe network that hinders water circulation and reduces heat transmission.
    - Filling and emptying valves.
    - Flow valves that allow opening or closing the water flow to the circuits depending on the temperature reached in the room by means of an ambient thermostat in the room.
    - Direct reading flow regulators that allow to easily adjusting the adequate flow in each circuit.
    - Thermometers, both on the way out and on the return for visual checking of the system temperatures.

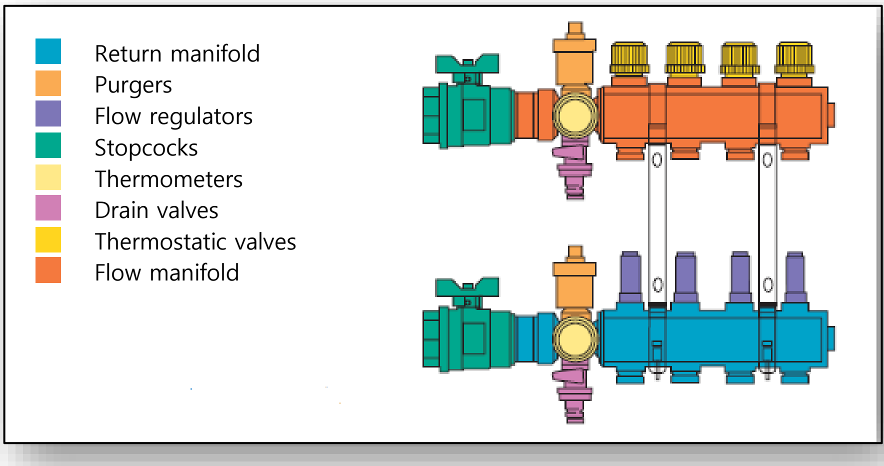


Figure 3. Components of the manifold assembly

- **Transmitting element**: in the same way, the components of the tramitting element below the ground will now be defined:

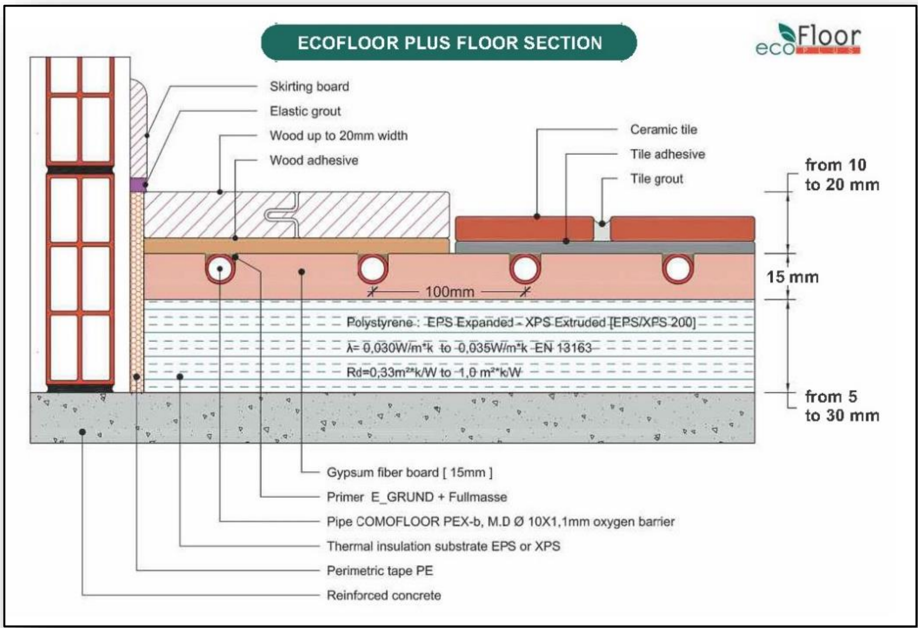


Figure 4. Floor Section

- *Tube*: It is the main element of the emitter assembly. Generally, in the underfloor heating floor system it is common the use of plastic pipes with anti-oxygen barrier that favors the sealing of the pipe, while decreasing the oxidation of the same. Some of the most common piping models are described below:

  1. Cross-linked polyethylene pipe (PE-X): High density cross-linked polyethylene pipe, whose mechanical properties are:

     - Great thermal resistance in conditions of high pressure.
     - Great ease of handling.
     - Very flexible, allows to operate with it in any condition.
     - High resistance to erosion. It allows high speeds of circulation.
     - Resistant to impacts due to the elasticity of the material.

  2. Polybutylene pipe (PB): It is made of polybutylene (crystallized thermoplastic polymer). It has the following mechanical properties:
     - Great ease of handling.
     - Very flexible pipe, can work in severe conditions.
     - Resistant to impacts due to the elasticity of the

material.

- It has an index of rigidity and low fragility; it is unlikely that blowouts occur.
- Recyclable. The PB is a completely recyclable material, therefore with a low environmental impact.

The benefits presented by the tubes described above are very similar and both perfectly valid for the underfloor heating system. A point to take into account when mounting the tube in the room to be heated is the way of mounting the pipe on the surface of the room, as it greatly affects the distribution of temperature on the floor. There are three types of distributions:

- o *In tubing*: The distribution of the tube begins at one end of the room and ends at the opposite end advancing in parallel lines equidistant from each other. This distribution is the simplest but presents a great inconvenience, the temperature distribution is not uniform throughout the stay, see Figure 2. 5. This is because the water is cooling along the circuit. It can be used in small rooms, although it is the least used distribution.
- o *In double tubing*: Like the serpentine distribution, it goes from one end of the room to the other, advancing in parallel lines equidistant from each other but leaving gaps where the return lines are placed until arriving again at the starting point . With this distribution, the temperature drop throughout the stay is reduced. It is used in rooms with large surfaces. It has the disadvantage that it has small radii of curvature.

- *<u>In spiral</u>*: it is done in spiral form of square or rectangular form starting from one end and advancing from outside the room towards the center of it leaving gaps to return to the starting point. This type of tube distribution is the one that keeps the temperature uniform in the whole room, since an inlet tube is alternated with a return tube. It is the most used and will be used in the present project.

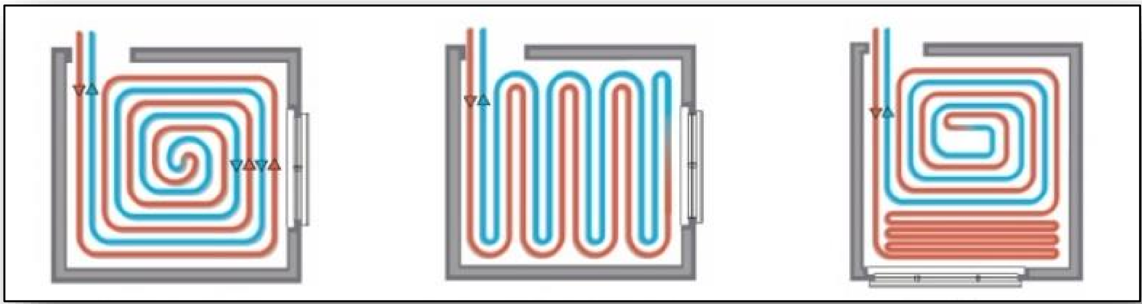


Figure 5. Tubing types

- **Insulating materials**: The thermal resistance of this insulating material allows maximizing that heat transmission to the room located above it. The placement of these insulating plates should be done so that the joints between the 15 panels are not aligned with each other. The perimeter-skirting band is placed along the walls allowing the movement of the plate and avoiding heat losses in the perimeter of the room. The plastic bib adhered to it is placed on the insulating plates, to prevent any leakage of mortar between the base and the plates. The top of the perimeter skirting should not be cut until the floor covering has been completed. The application of this type of installation requires minimum levels of insulation in the home for proper operation.

- *Mortar slab*: The mortar plate surrounds the pipes, stores and transmits the heat given by the water that circulates through them. The minimum thickness of this layer above the pipes, according to the UNE-1264 standard and for reasons of execution, must be at least 30mm. It is recommended the use of additives that fluidize mortar, which allows a perfect coating of the tubes and avoids possible air pockets that negatively affect the transmission of heat.
- *Floor covering*: The underfloor heating floor systems allow the use of any type of flooring, however, and as is logical, their behavior before heat transmission will differ in relation to the different coefficients of thermal conductivity.

- Regulation of the installation: the regulation elements constitute a very important part of the installation. The operating parameters must be adjusted to optimize the behaviour of the installation both from the point of view of comfort and energy savings. The advantages of regulation are multiple. It allows adapting the operation of the installation to the variations of the external conditions taking into account the inertia of the installation. Adjust the parameters to the comfort level defined by the user. It controls the formation of condensations on the surface of the floor, an important requirement in the cooling period. For an installation of air conditioning by underfloor heating, allow to act on two parameters, the temperature of the water flow to the installation and the temperature of the floor. For the regulation of the flow temperature and condensation control, the components of the installation are:
  - Cold heat regulator.

- Motorized 4-way valve.
- Temperature probe.
- Outdoor temperature probe.
- Surface temperature probe.
- Temperature / relative humidity control.

Regarding the control of the ambient temperature, the elements are:

- Control unit via radio.
- Thermostats via radio.
- Antenna.
- Thermoelectric valves.

1.2.2. System types

There are different radiant floor structures. Next, the most important ones will be described [1]:

- TYPE A AND C: the pipes are embedded inside a layer of mortar (plaque) between the floor and the floor covering in these systems. This is the oldest and most economical radiant floor system.

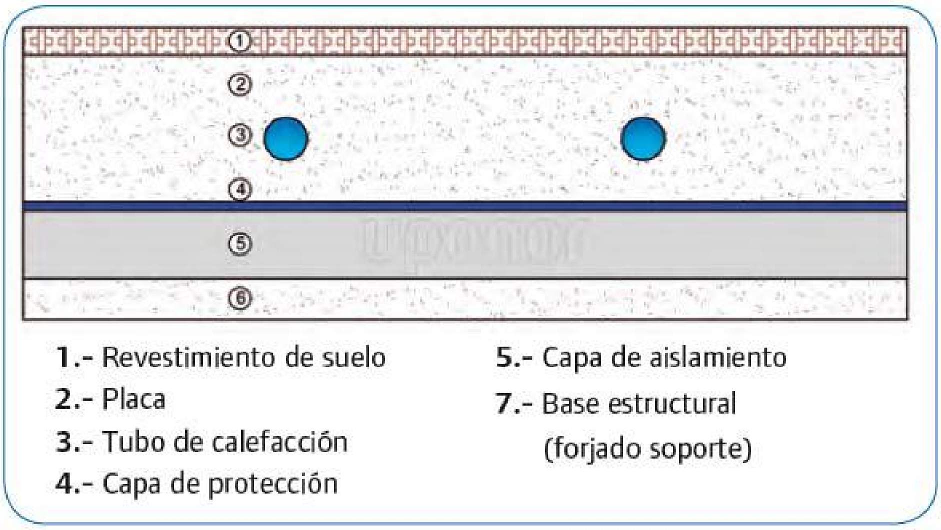


Figure 6. Radiant floor structure type A and C

- TYPE B: these are systems where the tubes are located in the insulation layer, leaving the layer of mortar on top, where diffusers are interposed to optimize the heat transfer to the layer mortar. Then you can see the layout in the following figure:

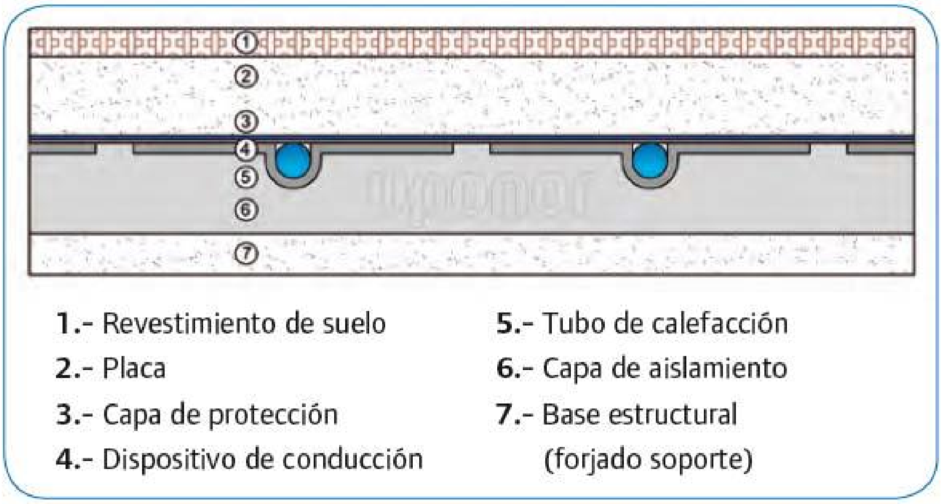


Figure 7. Radiant floor structure type B

- TYPE D: this type of structure is also known as flat sections; they are systems with parallel circulation and / or transversal flow over the entire surface.

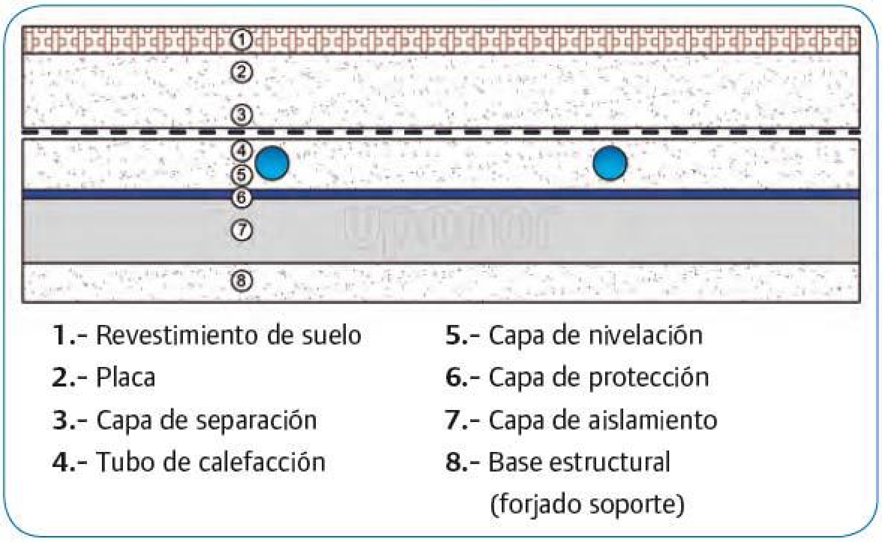


Figure 8. Radiant floor structure type D

1.2.3. Advantage and disadvantages of underfloor heating systems

In this section, many concepts are categorized with its positive advantages or not compared with other energy systems:

<u>Advantages</u>:

- **Comfort of the occupants**: according to the RITE [2] the comfort temperature is set between 21 and 23 ºC in winter and 23 and 25 ºC in summer, while Standard UNE-1264 [3] establishes a temperature of 20ºC for winter and 26 ºC for summer. These temperature differences in the design will decrease the energy demand. In addition, because the transfer or absorption of heat is done on the entire surface of the floor, it makes the temperature in the room more uniform than with other systems. The following figure shows a comparison of the ideal heating curve with different air conditioning systems: underfloor heating, radiators and hot air (fan coil).

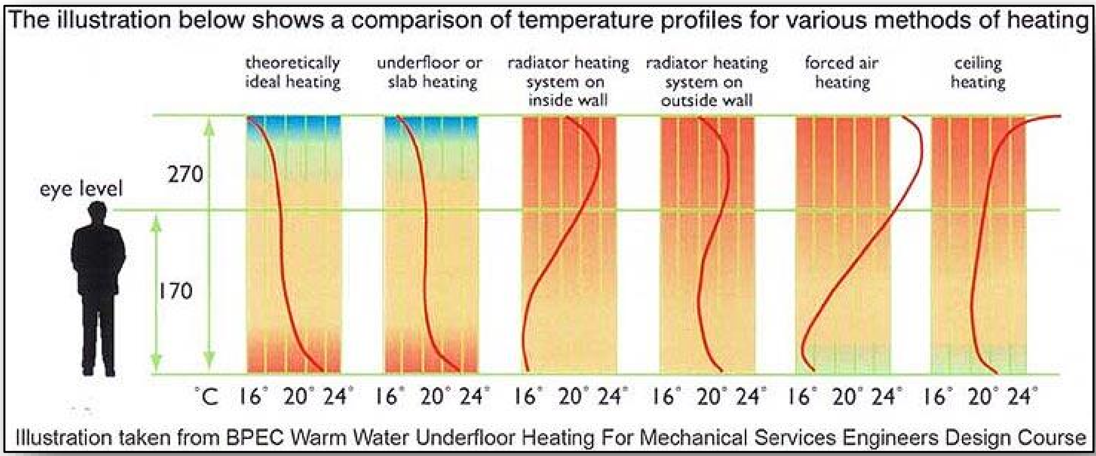


Figure 9. Comparision of temperatures profiles

- Energy efficiency: When lower heat transfer fluid temperatures are required in winter (between 35 and 45 ºC) and higher in summer (between 12 and 16 ºC) compared to conventional systems (for radiators in winter approximately 70 ºC and in summer of 7ºC by systems with fancoil) causes that the energetic efficiency increases.
- Healthy: Due to the absence of forced convection (fan coil), the movement of dust in the room is reduced. In addition, the low degree of humidity achieved prevents the appearance of mites and the development of allergies.
- Ecological: Due to the low operating temperature, it can be combined with renewable energy sources at low temperatures, such as solar or geothermal.
- Compatibility: It is compatible with any type of soil and energy source.

Disadvantages

- **Construction**: Requires an integral reform in the floor of the building (if it is not newly built).
- **Breakdowns**: If there is any breakage of the emitter tubes, some of the ground should be lifted for repair.
- **Conditioning time**: With respect to other air conditioning systems, such as the fan coil, the radiant floor requires more time to reach the comfort temperature, because it depends in large part on the thermal inertia.

2. EXISTING HOUSE: DATA COLLECTION AND PRELIMINARY DESIGN

2.1. BUILDING DESCRIPTION

The building we chose for the development of this project is a house located in Carrer Berenguer IV 57, Tortosa, Spain. This building counts with a total interior space of 136m^2, which are divided into the spaces shown on Table 1.

Space	Floor Area[m^2]	Volume [m^3]
Entrance hall	6,69	*16,73*
Corridor	23,99	*59,98*
Dining room	27,81	*69,53*
Kitchen	13,47	*33,68*
Laundry room	7,00	*17,50*
Office	10,83	*27,08*
Bedroom 1	12,17	*30,43*
Dressing room 1	3,52	*8,80*
Bedroom 2	13,55	*33,88*
Dressing room 2	7,04	*17,60*
Bathroom 1	5,55	*13,88*
Bathroom 2	4,45	*11,13*
TOTAL	136,07	*340,18*

Table 1. Space distribution, floor area and volume

The layout of these spaces inside the building will be shown on the blueprint on Annex 1. The estimation of the glass floor area in the building is shown on Table 2.

ID	Element	Space	Floor Area [m^2]
1	Glass	Dining room	9,68
2	Glass + Door	Dining room	9,70
3	Window	Kitchen	3,61
4	Window	Office	2,88

5	Door	Bedroom 1	2,01
6	Window	Bedroom 1	4,15
7	Door	Corridor	1,53
8	Window	Bedroom 2	2,68
9	Window	Bathroom 1	0,72
10	Window	Bathroom 2	0,64

Table 2: Glass floor area estimation

The position of this glass on the building spaces will also be shown on Annex 1.

2.2. WINTER THERMAL LOAD ESTIMATION

Once the building's layout has been described and its exterior windows and exterior doors have been located and measured, we can proceed to estimate the winter thermal load. To do so, we will estimate the thermal load for every conditioned room in the building, allowing us to detect which is the most demanding one and therefore the maximum heat power requested to the facility.

For the **calculation of the thermal power** of each room, the following formula has been used:

$$Q_{TP} = (Q_{CT} + Q_{AV} - Q_{IG}) \cdot (1 + \mathrm{F})$$

Equation 1. Thermal Power

Where the formula elements stand for:

Nomenclature	Definition	Development
Q_{TP}	Thermal Power	
Q_{CT}	Closure Transmissions	$Q_{CT} = U \cdot S \cdot (\vartheta_i - \vartheta_o)$
Q_{AV}	Air Ventilation	$Q_{AV} = 0{,}33 \cdot V \cdot (\vartheta i - \vartheta o)$

Q_{IG}	Internal Gains	$Q_{IG} = 0$
F	Factors	$F = Z_o + Z_{is} + Z_{ew}$

Table 3. Thermal power parameters' nomenclature and development

As given by the assignment wording, the **internal gains** will be neglected so we assume they are equal to zero. On the other hand, let's see how the other parameters' calculations have been done.

Starting with the **closure transmissions power (Q_{CT})**, for every conditioned room, we need first to detect the closures that can have thermal transmissions. These are walls, windows or doors that are in contact with the outside of the building. Furthemore, any inner walls or doors that are in contact with non-conditioned rooms and at last, the floor and the ceiling.

Then for each of these closures the following parameters have to be defined: transmittance (U), floor area (S), design indoor temperature (ϑ_i), the temperature difference between the design indoor temperature and the minimum temperature at the other side of the closure (ϑ_o), and finally, the orientation of the closures in contact with the exterior.

On the next table, the values of all these parameters and the final value of the closure transmissions' power for each conditioned room are shown.

Room	Closure	Orientation	U [W/m²·K]	S [m²]	ϑ_i [°C]	ϑ_o [°C]	Q_{CT} [W]	
Dining Room	Window	SE	3,5	9,70	20	2,4	597,37	
	Window	SW	3,5	9,68	20	2,4	595,98	
	Wall	SW	0,86	5,13	20	2,4	77,57	
	Door		1,5	3,65	20	11,2	48,18	
	Ceiling		0,49	27,75	20	11,2	119,66	
	Floor		0,64	27,75	20	11,2	156,29	1595,04
Kitchen	Wall	SW	0,86	8,21	20	2,4	124,28	
	Window	SW	3,5	3,61	20	2,4	222,62	

	Inner Wall		1,2	8,21	20	11,2	86,71	
	Door		1,5	3,13	20	11,2	41,25	
	Roof		0,49	13,58	20	11,2	58,54	
	Floor		0,64	13,58	20	11,2	76,45	609,85
Office	Wall	SW	0,86	6,42	20	2,4	97,24	
	Window	SW	3,5	2,88	20	2,4	177,13	
	Inner Wall		1,2	6,18	20	11,2	65,21	
	Door		1,5	3,13	20	11,2	41,25	
	Roof		0,49	10,83	20	11,2	46,68	
	Floor		0,64	10,83	20	11,2	60,97	488,48
Bedroom 1	Crystal Door	SW	3	2,01	20	2,4	105,97	
	Window	SW	3,5	4,14	20	2,4	255,30	
	Wall	SW	0,86	4,27	20	2,4	64,68	
	Inner Wall		1,2	7,30	20	11,2	77,09	
	Door		1,5	6,25	20	11,2	82,50	
	Roof		0,49	12,13	20	11,2	52,32	
	Floor		0,64	12,13	20	11,2	68,34	706,21
Bedroom 2	Wall	NW	0,86	7,80	20	2,4	118,00	
	Window	NW	3,5	2,68	20	2,4	165,04	
	Wall	NE	0,86	8,73	20	2,4	132,06	
	Inner Wall		1,2	4,13	20	11,2	43,56	
	Door		1,5	3,13	20	11,2	41,25	
	Roof		0,49	13,30	20	11,2	57,35	
	Floor		0,64	13,30	20	11,2	74,91	632,17
Dressing Room 2	Wall	NE	0,86	8,08	20	2,4	122,22	
	Inner Wall		1,2	2,20	20	11,2	23,23	
	Door		1,5	3,13	20	11,2	41,25	
	Roof		0,49	6,88	20	11,2	29,67	
	Floor		0,64	6,88	20	11,2	38,75	255,12
Bathroom 1	Window		3,5	0,64	24	2,4	48,40	
	Inner Wall		1,2	9,53	24	13,2	123,44	
	Door		1,5	3,13	24	13,2	50,63	
	Roof		0,49	6,14	24	13,2	32,50	
	Floor		0,64	6,14	24	13,2	42,45	297,41
Bathroom 2	Roof		0,49	4,72	24	13,2	24,96	
	Floor		0,64	4,72	24	13,2	32,60	
	Inner Wall		1,2	3,73	24	13,2	48,28	
	Window		3,5	0,32	24	2,4	24,19	130,03

Table 4. Closure transmissions' power calculation

The values of the transmittance (U) have been taken from an *Uponor's technical manual* [1], while the values of the design indoor temperature (ϑ_i) have been taken from the Spanish technical standard *UNE-EN 1264-2* [3].

On second place, we have the **air ventilation power (Q_{AV})**, for which we need to define for each room the minimum volume flow rate of air for outdoor air in order to ensure the hygiene conditions of living (V), the design indoor temperature (ϑ_i) and the outside air temperature from which the outdoor air is made (ϑ_o).

On the next table, the values of all these parameters and the final value of the air ventilation power for each conditioned room are shown.

Room	V [m^3/h]	ϑ_i [ºC]	ϑ_o [ºC]	Q_{AV} [W]
Dining Room	36,00	20	2,4	209,09
Kitchen	39,60	20	2,4	230,00
Office	14,40	20	2,4	83,64
Bedroom 1	14,40	20	2,4	83,64
Bedroom 2	28,80	20	2,4	167,27
Bathroom 1	39,60	24	13,2	141,13
Bathroom 2	39,60	24	13,2	141,13
Dressing room 2	14,4	20	2,4	83,64

Table 5. Air ventilation power

The values of the outside air temperature from which the outdoor air is made (ϑ_o) are taken from an external climatic conditions technical guide.

At last, we have the **multiplying factors (F)**, which are composed by three different factors and they solve possible increases of thermal power due to certain parameters. The first one is the **orientation factor (Z_O)** and it depends on where is each room oriented; its value is 0,2 if the orientation of one closure is North, 0,1 if it's East or West and 0 if it's South. Then we have the **interruptions**

factor (Z_{is}) and its value is fixed for conditioned rooms, which has a value of 0,1 for every conditioned room as it takes into account the time the conditioning system is not working. And finally **the external walls factor (Z_{ew})** that depends if each room has more than one wall to the outside because this is a case where the conditioned air has certain changes; it's value is 0,1 for a room with two or more walls in contact with the outside of the building and 0 otherwise.

For each room all of these factors sum up giving F as shown on the following table.

Room	Z_0	Z_{is}	Z_{ew}	F
Dining Room	0,1	0,1	0,1	0,3
Kitchen	0,1	0,1	0	0,2
Office	0,1	0,1	0	0,2
Bedroom 1	0,1	0,1	0	0,2
Bedroom 2	0,2	0,1	0,1	0,4
Bathroom 1	0	0,1	0	0,1
Bathroom 2	0	0,1	0	0,1
Dressing room 2	0,1	0,1	0	0,2

Table 6. Factors calculation

Once all of these calculations have been made, we can construct the following table with the values of the **thermal power (Q_{TP})** for every conditioned room.

Room	Q_{CT}	Q_{AV}	F	Q_{TP} [W]
Dining Room	1595,04	209,09	0,30	2345,37
Kitchen	609,85	230,00	0,20	1007,82
Office	488,48	83,64	0,20	686,53
Bedroom 1	706,21	83,64	0,20	947,81
Bedroom 2	632,17	167,27	0,40	1119,22
Bathroom 1	297,41	141,13	0,10	482,40
Bathroom 2	130,03	141,13	0,10	298,28
Dressing room 2	255,12	83,64	0,20	406,50
			TOTAL POWER:	7293,94

Table 7. Thermal power calculation

Once we have the thermal power for every conditioned room, we can find the **thermal load (q_H)** as the division of the thermal power of each room by its floor floor area (S).

Room	S [m2]	q_H [W/m2]
Dining Room	27,75	84,52
Kitchen	13,58	74,24
Office	10,83	63,42
Bedroom 1	12,13	78,11
Bedroom 2	13,30	84,14
Bathroom 1	6,14	78,56
Bathroom 2	4,72	63,24
Dressing room 2	6,88	59,09
	TOTAL	585,32

Table 8. Thermal load calculation

Finally, we can determine that the most demanding thermal load for any conditioned room is 84,52 W/m^2 for the dining room.

2.3. TUBING DESIGN

To design the tubing, the parameters we have to determine are the tubes' diameter and the pitch between them in order to get a water flow able to give us enough thermal energy to heat the conditioned rooms to the desired temperatures.

For the tubes' diameter, the chosen one is 16mm, which is a pretty standard measure for this means.

Once the diameter has been determined, we have to see for which pitch we can accomplish our goal, and to do so we will plot the characteristic curves for the A type system for different tube pitches. Then we will compare these curves with the limit curves, which express the maximum thermal load (q_G) that can be held by a certain pitch in a heating system.

2.3.1. Characteristic curves

Therefore, starting with the characteristic curves, they are defined by the following equation:

$$q_H = K_H \cdot \Delta\vartheta_H$$

Equation 2. Characteristic curves

Later on, we will plot the thermal flow density (q_H) versus the average air-water temperature ($\Delta\vartheta_H$), and the remaining term K_H is the thermal transmission equivalent coefficient, which represents the slope of the function.

This term K_H is determined by the following equation:

$$K_H = B \cdot a_B \cdot a_T^{m_T} \cdot a_U^{m_U} \cdot a_D^{m_D}$$

Equation 3. Thermal transmission equivalent coefficient

All the terms of this equation are defined by the Spanish technical standard UNE-EN 1264-2 [3]. Let's define first those that are constant:

- $B = B_0 = 6{,}7\,\frac{W}{m^2 \cdot K}$
- $a_B = \frac{\frac{1}{\alpha}+\frac{s_{u,0}}{\lambda_{u,0}}}{\frac{1}{\alpha}+\frac{s_{u,0}}{\lambda_E}+R_{\lambda,B}} = \frac{\frac{1}{10{,}8}+\frac{0{,}045}{1}}{\frac{1}{10{,}8}+\frac{0{,}045}{1{,}2}+0{,}05} = 0{,}764$
- $a_T = 1{,}188$
- $m_U = 100 \cdot (0{,}045 - s_u) = 100 \cdot (0{,}045 - 0{,}045) = 0$
- $m_D = 250 \cdot (D - 0{,}02) = 250 \cdot (0{,}016 - 0{,}02) = -1$

The values of α $(10{,}8[W/(m^2 \cdot K)])$, $s_{u,0}$ $(0{,}045[m])$, $\lambda_{u,0}$ $(1[W/(m \cdot K)])$and λ_E $(1{,}2\ [W/(m \cdot K)])$ are also defined by the same technical standard, while for the value of $R_{\lambda,B}$ $(0{,}05[m^2 \cdot K/W])$ we have chosen the value according to the type of floor in the conditioned building.

The remaining factors all depend on the pitch (T). The term m_T is given by the formula:

$$m_T = 1 - \frac{T}{0{,}075}$$

Equation 4. m_T Coefficient

While the values of a_U and a_D depend on the pitch (T) and on the thermal resistance of the flooring $(R_{\lambda,B})$. These values are expressed on tables of the same mentioned technical standard. The following table shows the values of a_U, a_D, m_T and K_H for different pitches.

T [m]	m_T	a_U	a_D	K_H
0,05	0,33	1,056	1,013	5,352
0,1	-0,33	1,050	1,025	4,715
0,15	-1,00	1,046	1,034	4,167
0,2	-1,67	1,041	1,040	3,694
0,225	-2,00	1,038	1,043	3,477
0,3	-3,00	1,031	1,049	2,910
0,375	-4,00	1,022	1,051	2,445

Table 9. Thermal transmission equivalent coefficient

2.3.2. Limit curves

Once we have calculated the characteristic curves, let's calculate the limit curves. The formula to do so is very similar to that one of the characteristic curves but with some changes, let's see it.

$$q_G = \mathrm{K} \cdot \Delta\vartheta_H$$

Equation 5. Limit curves calculation

The difference relies on the slope (K), so let’s develop that term to obtain a clearer representation of it.

$$q_G = \varphi \cdot B_G \cdot (\frac{\Delta\vartheta_H}{\varphi})^{n_G}$$

As with the characteristic curves, all the terms of this equation are defined by the Spanish technical standard UNE-EN 1264-2. So, let's analyse it term by term.

- For the calculation of B_G we have to first check the coefficient s_u/λ_E. In our case it has a value of $0{,}045/1{,}2 = 0{,}0375$ so that the value of B_G will be determined by the table A.4a of the aforementioned technical standard and will have a different value for every pitch.

- For n_G the same coefficient $s_u/\lambda_E = 0{,}0375$ is taken into account and in this case, we will find its values in the table A.5a of the same technical standard mentioned above. Also, in this case the value will be different for every pitch.
- At last we have the φ which is given by the following formula $\varphi = \frac{(\vartheta_{F,max} - \vartheta_i)}{\Delta\vartheta_o}$ where $\Delta\vartheta_o = 9K$ as given by the technical standard. On the other hand, we get the values of $\vartheta_{F,max}$ and ϑ_i from a technical manual. These last two values are different depending on the type of the conditioned room, so for peripheral zones (p) the values are $\vartheta_{F,max} = 35^oC$ and $\vartheta_i = 20^oC$, for inside normal spaces (i) $\vartheta_{F,max} = 29^oC$ and $\vartheta_i = 20^oC$, and for bathroom or similar spaces (b) $\vartheta_{F,max} = 33^oC$ and $\vartheta_i = 24^oC$.

Now a table is constructed with all these values for the same pitches we used before and find the slopes for the limit curves.

T [m]	B_G	n_G	φ_i	φ_b	φ_p	K_i	K_b	K_p
0,05	100	0	1	1	1,67	100,00	100,00	166,67
0,1	89,3	0,033	1	1	1,67	89,30	89,30	146,35
0,15	76,3	0,076	1	1	1,67	76,30	76,30	122,32
0,2	63,1	0,123	1	1	1,67	63,10	63,10	98,76
0,225	56,5	0,146	1	1	1,67	56,50	56,50	87,40
0,3	36,4	0,245	1	1	1,67	36,40	36,40	53,53
0,375	18,2	0,405	1	1	1,67	18,20	18,20	24,66

Table 10. Limit curves' slope calculation

All of the plotted curves will be shown on Annex 2, but for having an idea of how it looks now we'll show the characteristic curve and the limit curves for a pitch of 0,2m.

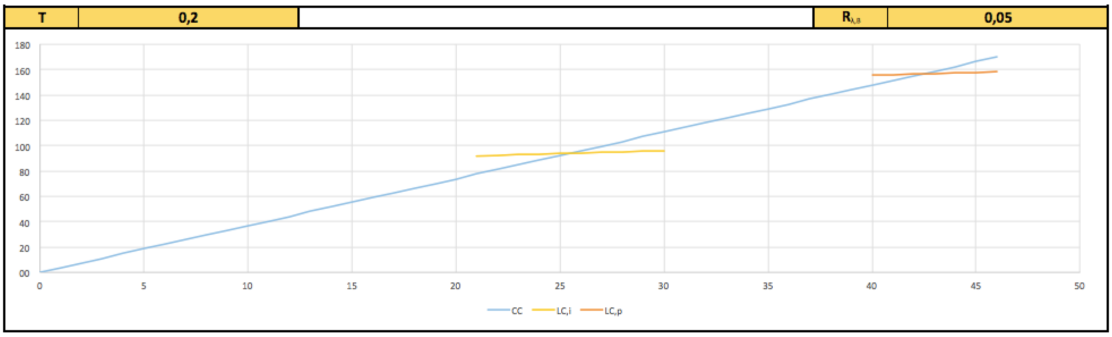


Figure 10. Limit and characteristic curve for a T = 0,2m pitch

The blue line is the characteristic curve, the yellow line is the inside normal spaces and bathroom limit curve, while the orange line is the peripheral zones limit curve. The X-axis shows the average air-water temperature ($\Delta\vartheta_H$), while thermal flow density (q_H) is represented in the Y-axis.

2.3.3. Design fluid temperature

For the system to be able to give the required thermal load, we have to check on the curves plot that the thermal flow density (q_H) is lower than the maximum thermal load (q_G). On the other hand, for type A floor heating and cooling systems, the pitches used are between 0,1 and 0,2 m. We see that for those pitches the plot of the curves, attached on Annex 2, and confirms the $q_H < q_G$ condition. So we choose an initial pitch of T=0,15m.

Now, considering the maximum thermal load of one conditioned room (bathrooms excluded) we calculate the average air-water temperature:

$$\Delta\vartheta_H = \frac{q_{Hmax}}{K_H} = \frac{84{,}52}{4{,}167} = 20{,}28\ ^oC$$

Equation 6. Air-water temperature

We will now dimension the whole system from this maximum thermal flow density, as it is the limiting one.

For the more unfavourable room, the thermal jump (σ) must be equal or lower than 5, so we can calculate $\Delta\vartheta_V$:

$$\Delta\vartheta_V = \Delta\vartheta_H + \frac{\sigma}{2} = 20{,}28 + \frac{5}{2} = 22{,}78\ ^oC$$

Equation 7. Thermal jump

Now we can calculate the heat carrier fluid temperature ϑ_V:

$$\vartheta_V = \Delta\vartheta_V + \vartheta_i = 22{,}78 + 20 = 42{,}78\ \ ^oC$$

Equation 8. Heat carrier fluid temperature

And therefore, the return fluid temperature ϑ_R:

$$\vartheta_R = \vartheta_V - \ \sigma = 42{,}78 - 5 = 37{,}78\ ^oC$$

Equation 9. Return fluid temperature

Once the heat carrier fluid temperature, for the most unfavourable room, has been calculated, we know it will remain constant, as we will only have one fluid temperature when this is pumped. So, we can calculate the thermal jump for the rest of the rooms by recalculating the average air-water temperature of each room with its own thermal flow density.

$$\Delta\vartheta_H = \frac{q_H}{K_H}\ ;\ \sigma = 2 \cdot (\vartheta_V - \Delta\vartheta_H)$$

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ [ºC]	σ [ºC]	T [m]
Dining Room	22,78	20,28	5,00	0,15
Kitchen	22,78	17,82	9,93	0,15
Office	22,78	15,22	15,13	0,15
Bedroom 1	22,78	18,74	8,08	0,15
Bedroom 2	22,78	20,19	5,18	0,15
Bathroom 1	22,78	18,85	7,86	0,15
Bathroom 2	22,78	15,18	15,21	0,15
Dressing room 2	22,78	14,18	17,21	0,15

Table 11. Thermal jump calculation

Following the technical standards, the thermal jump should be between 5ºC and 15ºC, and we can see that for three of the rooms it exceeds the 15ºC, so we should change the pitch for those rooms and recalculate the thermal jump.

Assigning a pitch of 0,2m instead of 0,15m to those three rooms, we obtain the following results:

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ ' [ºC]	σ ' [ºC]	T ' [m]
Dining Room	22,78	20,28	5,00	0,15
Kitchen	22,78	17,82	9,93	0,15
Office	22,78	17,17	11,22	0,2
Bedroom 1	22,78	18,74	8,08	0,15
Bedroom 2	22,78	20,19	5,18	0,15
Bathroom 1	22,78	18,85	7,86	0,15
Bathroom 2	22,78	17,12	11,32	0,2
Dressing room 2	22,78	16,00	13,57	0,2

Table 12. Thermal jump recalculation

Now all the thermal jumps are in the correct range, so the coil's pitch for every room has been defined.

Considering these pitches and the conditioned rooms' floor areas, a layout of the Tubing' circuits has been designed and will be shown on Annex 3.

2.4. WATER FLOW RATE

The aim of this section is to calculate the water volume of the heating system circuit, and we will do it by summing up the water volume of every room that will be calculated according to the following formula:

$$m_H = \frac{S \cdot q_H}{\sigma \cdot C_w} \cdot (1 + \frac{R_o}{R_u} + \frac{\vartheta_i - \vartheta_u}{q_H \cdot R_u})$$

Equation 10. Water flow rate calculation

The values of the panel heating floor area (S), the thermal flow density (q_H), the thermal jump (σ) and the inner room temperature (ϑ_i) have already been determined. On the other hand, let's see what values are missing:

- C_w is the water heat capacity, and has a value of $4190\, J/(kg \cdot K)$
- R_o is the rising thermal resistance and has a value of 0,19.
- R_u is the descending thermal resistance and has a value of 1,72.
- ϑ_u is the non-heating room temperature and has a value of $12\ ^oC$

Therefore, we obtain the following results:

Room	q_H [W/m^2]	σ [°C]	S [m^2]	m_H [l/h]
Dining Room	84,52	5,00	27,75	469,85
Kitchen	74,24	9,93	13,58	102,29
Office	63,42	11,22	10,83	62,23
Bedroom 1	78,11	8,08	12,13	118,00
Bedroom 2	84,14	5,18	13,30	216,48
Bathroom 1	78,56	7,86	6,14	61,68
Bathroom 2	63,24	11,32	4,72	26,81
Dressing room 2	59,09	13,57	6,88	30,61

Table 13. Water flow rate

So, summing up the water flow rate of every single room we obtain a total water flow rate of:

$$m_{H,TOT} = 1087{,}97\ l/h = 0{,}30\ l/s$$

We can also calculate the fluid velocity in every room by dividing the water flow rate by the tubes section area:

$$v_w[m/s] = \frac{m_H[m^3/s]}{S[m^2]}\ ;\ S = \pi \cdot r^2 = \pi \cdot 0{,}008^2 = 2{,}01 \cdot 10^{-4} m^2$$

Room	v_w [m/s]
Dining Room	0,65
Kitchen	0,14
Office	0,09
Bedroom 1	0,16
Bedroom 2	0,30
Bathroom 1	0,09
Bathroom 2	0,04
Dressing room 2	0,04

Table 14. Fluid velocity

2.5. CIRCUIT LENGTH AND MANIFOLD SIZING

We will now determine the number of manifolds we need in our system by calculating the circuit length for each room. The maximum length of one single circuit is of 120m, so if the required length for one room exceeds this number we will have two circuits for that room. In addition, for the manifolds there is a limitation of 9 circuits for each one, so if we have more than 9 circuits we will need 2 manifolds.

For each room, the circuit's length will be calculated as two times the distance from the manifold to the room (l), plus the room's floor floor area (S) divided by the room's circuit pitch (T).

$$L = 2 \cdot l + \frac{S}{T}$$

Equation 11. Circuit Length

Room	S [m²]	T [m]	l [m]	L [m]
Dining Room	27,75	0,15	4,28	193,56
Kitchen	13,58	0,15	3,845	98,19
Office	10,83	0,2	7,44	69,01
Bedroom 1	12,13	0,15	11,225	103,35
Bedroom 2	13,30	0,15	11,631	111,94
Bathroom 1	6,14	0,15	6,44	53,82
Bathroom 2	4,72	0,2	13,416	50,41
Dressing room 2	6,88	0,2	8,225	50,85
				731,12

Table 15. Circuit length

We can see that the only room that exceeds the 120m circuit length is the dining room, so we will have two circuits of equal length for that room. Recalculating that room, we obtain the following data:

Room	L [m]	$N_{CIRCUITS}$
Dining Room	96,78	2
Kitchen	98,19	1
Office	69,01	1
Bedroom 1	103,35	1
Bedroom 2	111,94	1
Bathroom 1	53,82	1
Bathroom 2	50,41	1
Dressing room 2	50,85	1
		9

Table 16. Number of circuits

Our final number of circuits is of 9; so, one manifold will be enough. Let's also mention that dividing the dining room's circuit in two different ones, the water flow rate and the fluid velocity of those circuits is also divided by two. Therefore, the new values will be:

$$m'_H = \frac{m_H}{2} = \frac{469{,}85}{2} = 234{,}93\ l/h = 6{,}5 \cdot 10^{-5} m^3/s$$

$$v'_w = \frac{v_w}{2} = \frac{0{,}65}{2} = 0{,}33\ m/s$$

2.6. PRESSURE DROP

In this part, we will calculate the circuit pressure drop by the following formula:

$$\Delta P = \Delta P_{MAX} + \Delta P_{ACCESSORIES} + \Delta P_{MANIFOLD}$$

Equation 12. Circuit Pressure

Let's see how each term of the previous equation is calculated:

- ΔP_{MAX} is the maximum of the pressure drops of the 9 circuits, which will be calculated like $\Delta P_{CIRCUIT} = f \cdot \frac{8 \cdot L \cdot Q^2}{g \cdot \pi^2 \cdot D^5} = f \cdot \frac{8 \cdot L \cdot Q^2}{9{,}81 \cdot \pi^2 \cdot 0{,}016^5}$; where f is the friction factor, L the length of the circuit, Q the water flow rate, g the gravity and $D = 0{,}016m$ the tubes' diameter.
- $\Delta P_{ACCESSORIES}$ is the accessories pressure drop, which is a 30% of ΔP_{MAX}.
- $\Delta P_{MANIFOLD}$ is the manifold's pressure drop, which is calculated from a graph shown in Annex 4, and it has a value of $1{,}5kPa$.

For the calculation of the friction factor (f) we need to calculate the Reynolds number of each circuit like $e = \frac{\rho \cdot v \cdot D}{\mu} = \frac{1000 \cdot v \cdot 0{,}016}{0{,}000891}$; where ρ is the fluid density, v is the fluid velocity, D is the tube's diameter and μ is the fluid dynamic viscosity.
If the Reynolds number is lower than 2300 we will have laminar flow, and the friction factor will be calculated as $= \frac{64}{Re}$; if it has a value in between 2300 and 4000 we have critical or transition flow and for more than 4000 we have turbulent flow. For these two last cases, the friction factor will be calculated following Moody's diagram that will be shown in Annex 5.
So, we obtain the following values of $\Delta P_{CIRCUIT}$.

Room	Q [m^3/s]	L [m]	v_W [m/s]	Re	f	$\Delta P_{CIRCUIT}$ [kPa]
Dining Room	0,000065	96,78	0,32	5828	0,036	18,06
Kitchen	0,000028	98,19	0,14	2538	0,046	10,11
Office	0,000017	69,01	0,09	1544	0,041	3,90
Bedroom 1	0,000033	103,35	0,16	2927	0,044	11,81
Bedroom 2	0,000060	111,94	0,30	5371	0,037	19,70
Bathroom 1	0,000017	53,82	0,09	1530	0,042	3,04
Bathroom 2	0,000007	50,41	0,04	665	0,096	2,85
Dressing room 2	0,000009	50,85	0,04	760	0,084	2,87

Table 17. Circuit pressure drop

Now the circuit pressure drop can be finally calculated.

$$\Delta P_{MAX} = MAX(\Delta P_{CIRCUIT}) = 19{,}70kPa$$

$$\Delta P_{ACCESSORIES} = 0{,}3 \cdot \Delta P_{MAX} = 0{,}3 \cdot 19{,}70 = 5{,}91kPa$$

$$\Delta P = \Delta P_{MAX} + \Delta P_{ACCESSORIES} + \Delta P_{MANIFOLD} = 19{,}70 + 5{,}91 + 1{,}5 = 27{,}11kPa$$

As the maximum circuit pressure drop was found for bedroom 2, we divide the final pressure drop by the bedroom 2 circuit length and we find the following value:

$$\frac{\Delta P_{MAX}}{L} = \frac{27110}{111{,}94} = 242{,}18\ Pa/m < 400\ Pa/m$$

As the resulting value is lower than $400\ Pa/m$ and this value it is acceptable, and we will have no pressure drop problems.

2.7. BOILER AND THERMAL EFFICIENCY

2.7.1. Boiler required thermal power

The next task is to size the boiler, and for it, we have to know the thermal power it should have. The net thermal power (P_U) needed for each conditioned room is the one calculated on the previous section EXISTING HOUSE 2. However, the power the boiler should give is not that one, but the thermal power (P_H) that we will extract from dividing the net thermal power by the thermal efficiency (η):

$$P_H = P_U/\eta$$

Equation 13. Thermal Power

The value of the thermal efficiency is different for each room and it is calculated like:

$$\eta = 1/(1 + \frac{R_o}{R_u} + \frac{\vartheta_i - \vartheta_u}{q_H \cdot R_u})$$

Equation 14. Thermal Efficiency

All the terms of the equation have already been defined on previous points, o let's see what values we obtain:

Room	P_U [W]	P_H [W]	η
Dining Room	2345,37	2730,37	0,86
Kitchen	1007,82	1180,94	0,85
Office	686,53	811,81	0,85
Bedroom 1	947,81	1107,69	0,86
Bedroom 2	1119,22	1303,23	0,86
Bathroom 1	482,40	563,60	0,86
Bathroom 2	298,28	352,77	0,85
Dressing room 2	406,50	482,87	0,84
	7293,94	8533,27	

Table 18. Boiler thermal power

So, the final total thermal power our boiler should give us is of 8,53 kW. Now we have to choose a boiler.

We chose a natural gas conventional boiler RE-VIS RE-ONE TOUCH 26, that has a power range that goes from 7,6 kW to 25,9 kW. That means, for the power we need it will be working approximately at a 30% of its total power giving us the maximum efficiency of 93,9%.

2.7.2. COMBUSTION ANALYSIS

The maximum efficiency though is given when the combustion takes place with a little bit of excess air, around 5 to 10%, as shown on figure 2.

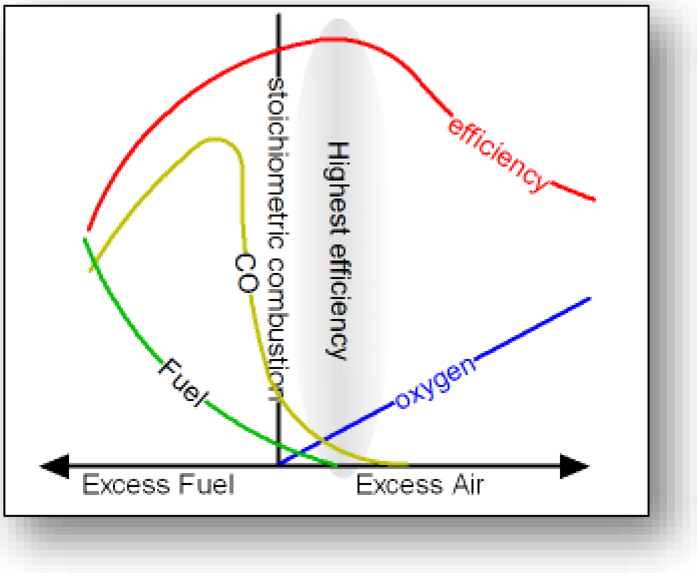


Figure 11. Combustion efficiency diagram

The combustion formula is $CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$, so we can deduct from there how many kg of air we would need for every kg of fuel (CH_4).

$$AAF = 1kgCH_4 \cdot \frac{1molCH_4}{16kgCH_4} \cdot \frac{2molO_2}{1molCH_4} \cdot \frac{32kgO_2}{1molO_2} \cdot \frac{1kgAir}{0{,}2035kgO_2} \cdot 1{,}1 = 21{,}62 \frac{kgAir}{kgCH_4}$$

The 1,1-multiplying factor in the end was taking into account a 10% excess air, and the final value means that for every kg of fuel (CH_4) we will need 21,62kg of air for the combustion to take place.
To have an approximation of the fuel annual cost, let's calculate the fuel mass flow per hour giving the needed power.

$$\dot{m}_f = \frac{P_H}{LHV \cdot \eta} = \frac{8{,}533}{45{,}904 \cdot 10^3 \cdot 0{,}939} = 1{,}98 \cdot 10^{-4} kg/s$$

Equation 15. Fuel Mass Flow

Assuming a specific cost of $\mathbf{0{,}733€/}\boldsymbol{kg_{CH4}}$ and that the boiler is working for our heating system an average of 20 hours a day for 4 months, the annual cost is the following:

$$Annual\ cost = (4 \cdot 30 \cdot 20 \cdot 3600)\frac{s}{year} \cdot \frac{1{,}98 \cdot 10^{-4}\ kg}{1\ s} \cdot \frac{0{,}733\ €}{1\ kg} = 1253{,}7€/year$$

Equation 16. Annual cost

Now we can also calculate the air mass flow like:

$$\dot{m}_a = AAF \cdot \dot{m}_f = 21{,}62 \cdot 1{,}98 \cdot 10^{-4} = 4{,}28 \cdot 10^{-3} kg/s$$

Equation 17. Air Mass Flow

Looking at the combustion we can also determine the hot products mass flow ($\dot{m}_p$) or exhaust gases flow ($\dot{m}_g$).

$$\dot{m}_p = \dot{m}_g = \dot{m}_f + \dot{m}_a = 1{,}98 \cdot 10^{-4} + 4{,}28 \cdot 10^{-3} = 4{,}478 \cdot 10^{-3} kg/s$$

The combustion temperature (T_c) or hot products temperature (T_p) can be calculated by using the following formula based on an analysis inside the combustion chamber:

$$T_c = T_p = T_{ca} + \frac{LHV}{Cp \cdot (1 + AAF)} = 20 + \frac{45{,}904 \cdot 10^3}{2{,}34 \cdot (1 + 21{,}62)} = 887{,}24^o C$$

Equation 18. Combustion Temperature

Where the T_{ca} is the temperature of the air at the entrance of the combustion chamber.

3. RENEWED HOUSE: OFF-DESIGN CALCULATIONS

In this second part of the project, we consider some improvements in the building external insulation. Therefore, we should recalculate the winter thermal flow density to see how much it has diminished and to be able to recalculate and resize the installation.

Let's then remind the formula we used to calculate the thermal power of each room:

$$Q_{TP} = (Q_{CT} + Q_{AV} - Q_{IG}) \cdot (1 + \mathrm{F})$$

Equation 19. Thermal Power

Like in the previous part, the internal gains are neglected, so that $Q_{IG} = \mathbf{0}$. The air ventilation power (Q_{AV}) and the multiplying factors ($\mathbf{F}$) remain the same than in the part S, as they are not affected by the new and more efficient enclosures. So, we can see that the only term of the equation that changes is the closure transmissions power (Q_{CT}), which had the following formulation:

$$Q_{CT} = U \cdot S \cdot (\vartheta_i - \vartheta_o)$$

Equation 20. Closure Transmission Power

The floor area area (S), the design indoor temperature (ϑ_i) and the temperature difference between the design indoor temperature and the minimum temperature at the other side of the closure (ϑ_o) remain the same with respect to the part S. We should only recalculate the values of the transmittance (U).

Watching the Uponor's technical manual from where we had taken the transmittance values for part S we see that those are the values for EXISTING HOUSEs not well insulated. Now we change those transmittance values for new ones according to the renewed insulation we are supposing, so let's build a table with the new transmittance values confronted to the old ones:

Type of enclosure	Old U [W/m²K]	New U [W/m²K]
Ceiling	0,49	0,15
Floor	0,64	0,2
Wall	0,86	0,25
Inner wall	1,2	0,5
Window	3,5	1,2
Door	1,5	1,5

Table 19. Transmittance value change

Once we have these new values, the procedure is the same one than in the part EXISTING HOUSE. So, the results of the calculation of the thermal power of each room and the thermal flow density are expressed on the following tables:

Room	Q_{CT}	Q_{AV}	F	Q_{TP} [W]
Dinning Room	565,35	209,09	0,30	1006,76
Kitchen	231,65	230,00	0,20	553,97
Office	190,76	83,64	0,20	329,27
Bedroom 1	275,99	83,64	0,20	431,55
Bedroom 2	229,64	167,27	0,40	555,68
Bathroom 1	141,87	141,13	0,10	311,30
Bathroom 2	46,24	141,13	0,10	206,11
Dressroom 2	107,65	83,64	0,20	229,54
			TOTAL POWER:	3624,19

Table 20. Thermal power

Room	S [m^2]	q_H [W/m^2]
Dining Room	27,75	36,28
Kitchen	13,58	40,81
Office	10,83	30,42
Bedroom 1	12,13	35,56
Bedroom 2	13,30	41,78
Bathroom 1	6,14	50,69
Bathroom 2	4,72	43,70
Dressing room 2	6,88	33,36
	TOTAL	312,60

Table 21. Thermal flow density

Finally, we can determine that the most demanding thermal flow density for any conditioned room is 41,78 W/m^2 for the bedroom 1, because the bathrooms are not considered.

We can see that compared to part EXISTING HOUSE, the maximum thermal flow density for any room has decreased more than a 50% as before it was 84,52 W/m^2 while now it's 41,78 W/m^2. The characteristic and limit curves remain unchanged, as they depend on the system and not on the thermal flow density, so we already have those from part EXISTING HOUSE.

We maintain the chosen tube's diameter of 16mm, and now we should proceed to define the pitch of the circuit for every room, so we will do that on two different ways. For part A.1 we will start from a 0,15m pitch just like in part EXISTING HOUSE, and for part A.2 we will start from a 0,2 pitch, that is the ideal one for only heating systems.
After determining the pitch, we will carry on with the rest of the calculations as we did in part EXISTING HOUSE to have a costs and other parameters comparison.

3.1. 0,15 INITIAL PITCH

3.1.1. Design fluid temperature

The first step is to recalculate the average air-water temperature:

$$\Delta\vartheta_H = \frac{q_{Hmax}}{K_H} = \frac{41{,}78}{4{,}167} = 10{,}03\ ^oC$$

Equation 21. Average air-water temperature

We dimension the whole system for the maximum thermal flow density and a thermal jump of 5 degree for the most unfavourable room:

$$\Delta\vartheta_V = \Delta\vartheta_H + \frac{\sigma}{2} = 10{,}03 + \frac{5}{2} = 12{,}53\ ^oC$$
$$\vartheta_V = \Delta\vartheta_V + \vartheta_i = 12{,}53 + 20 = 32{,}53\ ^oC$$
$$\vartheta_R = \vartheta_V - \sigma = 32{,}53 - 5 = 27{,}53\ ^oC$$

Having the values for the mot unfavourable room we can now proceed to calculate the rest of the rooms.

$$\Delta\vartheta_H = \frac{q_H}{K_H}\ ;\ \sigma = 2 \cdot (\vartheta_V - \Delta\vartheta_H)$$

Equation 22. Thermal Jump

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ [ºC]	σ [ºC]	T [m]
Dining Room	12,53	8,71	7,64	0,15
Kitchen	12,53	9,79	5,46	0,15
Office	12,53	7,30	10,45	0,15
Bedroom 1	12,53	8,53	7,98	0,15
Bedroom 2	12,53	10,03	5,00	0,15
Bathroom 1	12,53	12,17	0,72	0,15

Bathroom 2	12,53	10,49	4,08	0,15
Dressing room 2	12,53	8,01	9,04	0,15

Table 22. Thermal jump calculation

For the both bathrooms the thermal jump is lower than 5, so we should change the pitch in order to have a pitch between 5 and 15 degrees. Assigning a pitch of 0,05m for the bathroom 1 and of 0,1m for the bathroom 2 we obtain the following results where we can see that all the thermal jumps are in the correct range, so the coil's pitch for every room has been defined.

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ ' [ºC]	σ ' [ºC]	T ' [m]
Dining Room	12,53	8,71	7,64	0,15
Kitchen	12,53	9,79	5,46	0,15
Office	12,53	7,30	10,45	0,15
Bedroom 1	12,53	8,53	7,98	0,15
Bedroom 2	12,53	10,03	5,00	0,15
Bathroom 1	12,53	9,47	6,11	0,05
Bathroom 2	12,53	9,27	6,52	0,1
Dressing room 2	12,53	8,01	9,04	0,15

Table 23. Thermal jump recalculation

3.1.2. Water flow rate

The procedure to calculate the water flow rate is the same than in part EXISTING HOUSE, so using the new values of thermal flow density and thermal jumps we obtained from the design fluid temperature calculations we can calculate the water flow rate for every conditioned room:

Room	q_H [W/m^2]	σ [ºC]	S [m^2]	m_H [l/h]
Dining Room	36,28	7,64	27,75	140,33
Kitchen	40,81	5,46	13,58	106,68
Office	30,42	10,45	10,83	34,21
Bedroom 1	35,56	7,98	12,13	57,68
Bedroom 2	41,78	5,00	13,30	116,71
Bathroom 1	50,69	6,11	6,14	52,68
Bathroom 2	43,70	6,52	4,72	33,09
Dressing room 2	33,36	9,04	6,88	27,29

Table 24. Water flow rate calculation

So, summing up the water flow rate of every single room we obtain a total water flow rate of:

$$m_{H,TOT} = 568{,}66\ l/h = 0{,}16\ l/s$$

We can also calculate the fluid velocity in every room by dividing the water flow rate by the tubes section area:

$$v_w[m/s] = \frac{m_H[m^3/s]}{S[m^2]}\ ;\ S = \pi \cdot r^2 = \pi \cdot 0{,}008^2 = 2{,}01 \cdot 10^{-4} m^2$$

Room	v_w [m/s]
Dining Room	0,19
Kitchen	0,15
Office	0,05
Bedroom 1	0,08
Bedroom 2	0,16
Bathroom 1	0,07
Bathroom 2	0,05
Dressing room 2	0,04

Table 25. Fluid velocity calculation

3.1.3. Circuit length and manifolds sizing

As for part EXISTING HOUSE, the circuit length for each room will be calculated as two times the distance from the manifold to the room (l), plus the room floor area (S) divided by the room circuit pitch (T).

$$L = 2 \cdot l + \frac{S}{T}$$

Equation 23. Circuit length

Room	S [m²]	T [m]	l [m]	L [m]
Dining Room	27,75	0,15	4,28	193,56
Kitchen	13,58	0,15	3,845	98,19
Office	10,83	0,15	7,44	87,05
Bedroom 1	12,13	0,15	11,225	103,35
Bedroom 2	13,30	0,15	11,631	111,94
Bathroom 1	6,14	0,05	6,44	135,70
Bathroom 2	4,72	0,1	13,416	74,00
Dressing room 2	6,88	0,15	8,225	62,32
				866,09

Table 26. Circuit length calculation

We can see that the only rooms that exceeds the 120m circuit length are the dining room and the bathroom 1, so we will have two circuits of equal length for those rooms. Recalculating those rooms, we obtain the following data:

Room	L [m]	$N_{CIRCUITS}$
Dining Room	96,78	2
Kitchen	98,19	1
Office	87,05	1
Bedroom 1	103,35	1
Bedroom 2	111,94	1
Bathroom 1	67,85	2

Bathroom 2	74,00	1
Dressing room 2	62,32	1
	TOTAL:	10

Table 27. Number of circuits

Our final number of circuits is of 10; so, we will need two manifolds because the maximum number of circuits for one manifold is of nine.

Let's also mention that dividing a room circuit in two different ones, the water flow rate and the fluid velocity of those circuits is also divided by two. So, the new values for the dining room will be:

$$m'_H = \frac{m_H}{2} = \frac{140,33}{2} = 70,165\ l/h = 1,95 \cdot 10^{-4} m^3/s$$

$$v'_w = \frac{v_w}{2} = \frac{0,19}{2} = 0,095\ m/s$$

In addition, the new values for bathroom 1 will be:

$$m'_H = \frac{m_H}{2} = \frac{52,68}{2} = 26,34\ l/h = 7,32 \cdot 10^{-6} m^3/s$$

$$v'_w = \frac{v_w}{2} = \frac{0,07}{2} = 0,035\ m/s$$

3.1.4. Pressure drop

For the pressure drop calculation, we follow the same procedure than in part EXISTING HOUSE and we find that the maximum pressure drop for any of the 10 circuits is given for bedroom 2, so with this value we can calculate the circuit pressure drop:

$$\Delta P_{MAX} = MAX(\Delta P_{CIRCUIT}) = 12{,}65kPa$$

$$\Delta P_{ACCESSORIES} = 0{,}3 \cdot \Delta P_{MAX} = 0{,}3 \cdot 12{,}65 = 3{,}80kPa$$

$$\Delta P = \Delta P_{MAX} + \Delta P_{ACCESSORIES} + \Delta P_{MANIFOLD} = 12{,}65 + 3{,}80 + 1{,}5 = 17{,}95kPa$$

As the maximum circuit's pressure drop was found for bedroom 2, we divide the final pressure drop by the bedroom's 2 circuit's length and we find the following value:

$$\frac{\Delta P_{MAX}}{L} = \frac{17950}{111{,}94} = 160{,}35\ Pa/m < 400\ Pa/m$$

As the resulting value is, lower than $400\ Pa/m$ it is acceptable, and we will have no pressure drop problems.

3.1.5. Boiler and thermal efficiency

3.1.5.1. Boiler required thermal power

Following the same procedure than in part EXISTING HOUSE.7.1, we find that for this case the final total thermal power of the boiler has to be of $P_H = 4{,}46kW$. As the power that the boiler should give is significantly lower than before, we would choose a different natural gas boiler, that in this case would be RE-VIS RE-ONE TOUCH 10, that has a power range that goes from 4,2 kW to 15 kW. That means, for the power we need it will be working approximately at a 30% of its total power giving us the maximum efficiency of 93,8%.

3.1.5.2. Combustion analysis

The AAF and the excess air percentage remain the same, but as the required power is lower now, the fuel mass flow will decrease, so let's recalculate it:

$$\dot{m_f} = \frac{P_H}{LHV \cdot \eta} = \frac{4{,}46}{45{,}904 \cdot 10^3 \cdot 0{,}938} = 1{,}04 \cdot 10^{-4} kg/s$$

Equation 24. Fuel Mass Flow

Assuming a specific cost of $0{,}733€/kg_{CH4}$ and that the boiler is working for our heating system an average of 20 hours a day for 4 months, the annual cost is the following:

$$Annual\ cost = (4 \cdot 30 \cdot 20 \cdot 3600) \frac{s}{year} \cdot \frac{1{,}04 \cdot 10^{-4}\ kg}{1\ s} \cdot \frac{0{,}733\ €}{1\ kg} = 658{,}6€/year$$

Now we can also calculate the air mass flow like:

$$\dot{m}_a = AAF \cdot \dot{m_f} = 21{,}62 \cdot 1{,}04 \cdot 10^{-4} = 2{,}25 \cdot 10^{-3} kg/s$$

Equation 25. Air Mass Flow.

Looking at the combustion we can also determine the hot products mass flow ($\dot{m}_p$) or exhaust gases flow ($\dot{m}_g$).

$$\dot{m}_p = \dot{m}_g = \dot{m}_f + \dot{m}_a = 1{,}04 \cdot 10^{-4} + 2{,}25 \cdot 10^{-3} = 2{,}354 \cdot 10^{-3} kg/s$$

The combustion temperature (T_c) or hot products temperature (T_p) will remain the same than the one of the part EXISTING HOUSE.

3.1.5.3. Heat exchanger

Looking now how to transfer the heat gained in combustion of natural gas, a device that allows passing this heat to another fluid (water in this case) without any contact between them is needed. This functions is covered by heat exchanger. Meanwhile one fluid has a loss of heat, another gain it by transfer. So, we have that:

$$P_H = \dot{m}_w \cdot Cp_w \cdot (T_{wo} - T_{wi}) = \dot{m}_p \cdot Cp_{NG} \cdot \left(T_p - T_g\right)$$

Equation 26. Heat exchanger

Where:

- $\dot{m}_w = m_H \cdot \rho_{H2O} = 0{,}16 l/s \cdot 1 kg/l = 0{,}16 kg/s$ is the water mass flow in the heat exchanger.
- $Cp_w = 4{,}18 kJ/(kg \cdot ºC)$is the water specific heat.
- $T_{wo} = 32{,}53^{o}C$ is the water temperature at the exit of the heat exchanger.
- $T_{wi} = 25{,}07^{o}C$ is the water temperature at the entrance of the heat exchanger. It has been calculated taking into account the return temperatures of every coil circuit and weighted with the fluid quantity of every circuit.
- $\dot{m}_p = 2{,}354 \cdot 10^{-3} kg/s$ is the hot products mass flow.
- $Cp_{NG} = 2{,}34 kJ/(kg \cdot^{o} C)$ is the natural gas specific heat.
- $T_p = 887{,}24^{o}C$ is the hot products temperature.
- T_g is the exhaust gases temperature.

Substituting all of the values in the previous equation we obtain that the exhaust gases temperature is $T_g = 77{,}56C$.

The total heat transfer coefficient in the heat exchanger between the water and the gases remains the same $U = 138W/m^2 \cdot K$. However, the number of tubes $N_{tubes} = 4$ and the fluid velocity $v = 0{,}090m/s$ change.

The next step is to calculate the dimensions of the heat exchanger's shell. We will calculate its length ad its diameter. Let's start for the length.

As the values of the boiler thermal power $\dot{Q} = 4{,}46\text{kW}$, the logarithmic temperature difference $\Delta T_{lm} = 276{,}8^oC$ and the number of tubes $N_{tubes} = 4$ changed, the length of the exchanger's shell will also change, and the new value is of $L = 0{,}348m = 34{,}8cm$.

The final step is defining the heat exchanger's shell diameter D_s.

The number of tubes changed from 8 to 4 in comparison to part EXISTING HOUSE, so the new D_s is:

$$D_s = D_o \cdot (N_t/K_1)^{\frac{1}{n}} = 0{,}0267 \cdot (4/0{,}319)^{\frac{1}{2{,}142}} = 0{,}087m = 8{,}7cm$$

Equation 27. Diameter of tubes

3.1.5.4. Flue gas exhaust duct design

Calculating the new value of the exhaust gases flux in cubic meters per second:

$$Q = \frac{10{,}5\, m^3\, gas}{1\, m^3\, fuel} \cdot \frac{1\, m^3\, fuel}{0{,}7\, kg\, fuel} \cdot \frac{1{,}04 \cdot 10^{-4} kg}{1s} = 1{,}56 \cdot 10^{-3} m^3/s \;\; gas$$

We can obtain the new exhaust tube's diameter:

$$d = \sqrt{\frac{4 \cdot Q}{\pi \cdot v}} = \sqrt{\frac{4 \cdot 1{,}56 \cdot 10^{-3}}{\pi \cdot 1{,}05}} = 0{,}043m = 4{,}3cm$$

Equation 28. Exhaust tube diameter

3.2. 0,20 INITIAL PITCH

3.2.1. Design fluid temperature

The first step is to recalculate the average air-water temperature:

$$\Delta\vartheta_H = \frac{q_{Hmax}}{K_H} = \frac{41{,}78}{3{,}694} = 11{,}31\ ^oC$$

Equation 29. Average air-water temperature

We dimension the whole system for the maximum thermal flow density and a thermal jump of 5 degree for the most unfavourable room:

$$\Delta\vartheta_V = \Delta\vartheta_H + \frac{\sigma}{2} = 11{,}31 + \frac{5}{2} = 13{,}81\ ^oC$$
$$\vartheta_V = \Delta\vartheta_V + \vartheta_i = 13{,}81 + 20 = 33{,}81\ ^oC$$
$$\vartheta_R = \vartheta_V - \sigma = 33{,}81 - 5 = 28{,}81\ ^oC$$

Having the values for the mot unfavourable room, we can now proceed to calculate the rest of the rooms.

$$\Delta\vartheta_H = \frac{q_H}{K_H}\ ;\ \sigma = 2 \cdot (\vartheta_V - \Delta\vartheta_H)$$

Equation 30. Thermal jump

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ [ºC]	σ [ºC]	T [m]
Dining Room	13,81	9,82	7,98	0,2
Kitchen	13,81	11,05	5,52	0,2
Office	13,81	8,24	11,15	0,2
Bedroom 1	13,81	9,63	8,36	0,2
Bedroom 2	13,81	11,31	5,00	0,2
Bathroom 1	13,81	13,72	0,17	0,2

Bathroom 2	13,81	11,83	3,96	0,2
Dressing room 2	13,81	9,03	9,56	0,2

Table 28. Thermal jump calculation

For the both bathrooms the thermal jump is lower than 5, so we should change the pitch in order to have a pitch between 5 and 15 degrees. Assigning a pitch of 0,1m for the bathroom 1 and of 0,15m for the bathroom 2 we obtain the following results where we can see that all the thermal jumps are in the correct range, so the coil's pitch for every room has been defined.

Room	$\Delta\vartheta_V$ [ºC]	$\Delta\vartheta_H$ ' [ºC]	σ ' [ºC]	T ' [m]
Dining Room	13,81	9,82	7,98	0,2
Kitchen	13,81	11,05	5,52	0,2
Office	13,81	8,24	11,15	0,2
Bedroom 1	13,81	9,63	8,36	0,2
Bedroom 2	13,81	11,31	5,00	0,2
Bathroom 1	13,81	10,75	6,12	0,1
Bathroom 2	13,81	10,49	6,65	0,15
Dressing room 2	13,81	9,03	9,56	0,2

Table 29. Thermal jump recalculation

3.2.2. Water flow rate

The procedure to calculate the water flow rate is the same than in part EXISTING HOUSE, so using the new values of thermal flow densitys and thermal jumps we obtained from the design fluid temperature calculations we can calculate the water flow rate for every conditioned room:

Room	q_H [W/m^2]	σ [°C]	S [m^2]	m_H [l/h]
Dining Room	36,28	7,98	27,75	134,38
Kitchen	40,81	5,52	13,58	105,53
Office	30,42	11,15	10,83	32,07
Bedroom 1	35,56	8,36	12,13	55,04
Bedroom 2	41,78	5,00	13,30	116,71
Bathroom 1	50,69	6,12	6,14	52,56
Bathroom 2	43,70	6,65	4,72	32,43
Dressing room 2	33,36	9,56	6,88	25,81

Table 30. Water flow rate calculation

So, summing up the water flow rate of every single room we obtain a total water flow rate of:

$$m_{H,TOT} = 554{,}53\ l/h = 0{,}15\ l/s$$

We can also calculate the fluid velocity in every room by dividing the water flow rate by the tubes section area:

$$v_w[m/s] = \frac{m_H[m^3/s]}{S[m^2]}\ ;\ S = \pi \cdot r^2 = \pi \cdot 0{,}008^2 = 2{,}01 \cdot 10^{-4} m^2$$

Room	v_W [m/s]
Dining Room	0,19
Kitchen	0,15
Office	0,04
Bedroom 1	0,08
Bedroom 2	0,16
Bathroom 1	0,07
Bathroom 2	0,04
Dressing room 2	0,04

Table 31. Fluid velocity calculation

3.2.3. Circuit length and manifold sizing

As for part EXISTING HOUSE, the circuit length for each room will be calculated as two times the distance from the manifold to the room (l), plus the room's floor floor area (S) divided by the room's circuit pitch (T).

$$L = 2 \cdot l + \frac{S}{T}$$

Equation 31. Circuit length

Room	S [m²]	T [m]	l [m]	L [m]
Dining Room	27,75	0,2	4,28	147,31
Kitchen	13,58	0,2	3,845	75,57
Office	10,83	0,2	7,44	69,01
Bedroom 1	12,13	0,2	11,225	83,12
Bedroom 2	13,30	0,2	11,631	89,77
Bathroom 1	6,14	0,1	6,44	74,29
Bathroom 2	4,72	0,15	13,416	58,27
Dressing room 2	6,88	0,2	8,225	50,85
				648,18

Table 32. Circuit length calculation

We can see that the only room that exceeds the 120m circuit length is the dining room, so we will have two circuits of equal length for this room. Recalculating this room, we obtain the following data:

Room	L [m]	$N_{CIRCUITS}$
Dining Room	73,65	2
Kitchen	75,57	1
Office	69,01	1
Bedroom 1	83,12	1
Bedroom 2	89,77	1
Bathroom 1	74,29	1
Bathroom 2	58,27	1
Dressing room 2	50,85	1
		9

Table 33. Number of circuits

Our final number of circuits is of 9; so, one manifold will be enough.

Let's also mention that dividing a room's circuit in two different ones, the water flow rate and the fluid velocity of those circuits is also divided by two. So, the new values for the dining room will be:

$$m'_H = \frac{m_H}{2} = \frac{134{,}38}{2} = 67{,}19\ l/h = 1{,}86 \cdot 10^{-5} m^3/s$$

$$v'_w = \frac{v_w}{2} = \frac{0{,}19}{2} = 0{,}095\ m/s$$

3.2.4. Pressure drop

For the pressure drop calculation, we follow the same procedure than in part EXISTING HOUSE and we find that the maximum pressure drop for any of the 9 circuits is given for bedroom 2, so with this value we can calculate the circuits' pressure drop:

$$\Delta P_{MAX} = MAX(\Delta P_{CIRCUIT}) = 10{,}94kPa$$

$$\Delta P_{ACCESSORIES} = 0{,}3 \cdot \Delta P_{MAX} = 0{,}3 \cdot 10{,}94 = 3{,}28kPa$$

$$\Delta P = \Delta P_{MAX} + \Delta P_{ACCESSORIES} + \Delta P_{MANIFOLD} = 10{,}94 + 3{,}28 + 1{,}5 = 15{,}72kPa$$

As the maximum circuit's pressure drop was found for bedroom 2, we divide the final pressure drop by the bedroom 2 circuit length and we find the following value:

$$\frac{\Delta P_{MAX}}{L} = \frac{15720}{89{,}77} = 175{,}11\ Pa/m < 400\ Pa/m$$

As the resulting value is, lower than $400\ Pa/m$ it is acceptable, and we will have no pressure drop problems.

3.2.5. Boiler and thermal efficiency

3.2.5.1. Boiler required thermal power

This part is completely equal to the part A.1.5.1, therefor the total thermal power of the boiler is $P_H = 4{,}46kW$.

3.2.5.2. Combustion analysis

All the values calculated on this same point for part A.1 are also the same, so:

$$\dot{m}_f = 1{,}04 \cdot 10^{-4} kg/s$$
$$Annual\ cost = 658{,}6€/year$$
$$\dot{m}_a = 2{,}25 \cdot 10^{-3} kg/s$$
$$\dot{m}_p = \dot{m}_g = 2{,}354 \cdot 10^{-3} kg/s$$

3.2.5.3 Heat exchanger

Looking now how to transfer the heat gained in combustion of natural gas, a device that allows passing this heat to another fluid (water in this case) without any contact between them is needed. These functions are covered by heat exchanger. Meanwhile one fluid has a loss of heat, another gain it by transfer. So, we have that:

$$P_H = \dot{m}_w \cdot Cp_w \cdot (T_{wo} - T_{wi}) = \dot{m}_p \cdot Cp_{NG} \cdot \left(T_p - T_g\right)$$

Equation 32. Heat exchanger

Where:

- $\dot{m}_w = m_H \cdot \rho_{H2O} = 0{,}15 l/s \cdot 1 kg/l = 0{,}15 kg/s$ is the water mass flow in the heat exchanger.
- $Cp_w = 4{,}18 kJ/(kg \cdot^o C)$is the water specific heat.
- $T_{wo} = 33.81^o C$ is the water temperature at the exit of the heat exchanger.

- $T_{wi} = 26{,}89^oC$ is the water temperature at the entrance of the heat exchanger. It has been calculated considering the return temperatures of every coil circuit and weighted with the fluid quantity of every circuit.
- $\dot{m}_p = 2{,}354 \cdot 10^{-3} kg/s$ is the hot products mass flow.
- $Cp_{NG} = 2{,}34 kJ/(kg \cdot^o C)$ is the natural gas specific heat.
- $T_p = 887{,}24^oC$ is the hot products temperature.
- T_g is the exhaust gases temperature.

Substituting all of the values in the previous equation we obtain that the exhaust gases temperature is $T_g = 99,5^oC$.

The total heat transfer coefficient in the heat exchanger between the water and the gases remains the same $U = 138W/m^2 \cdot K$. And so, do the number of tubes $N_{tubes} = 4$ and the fluid velocity $v = 0{,}090 m/s$ change.

The next step is to calculate the dimensions of the heat exchanger's shell. We will calculate its length ad its diameter. Let's start for the length.

As the value of the logarithmic temperature variation $\Delta T_{lm} = 308,9^oC$ changed, the length of the exchanger's shell will also change, and the new value is of $L = 0{,}312m = 31{,}2cm$.

The final step is defining the heat exchanger's shell diameterD_s that will be the same one then for part A.1:

$$D_s = 0{,}087m = 8{,}7cm$$

3.2.5.4. Fuel gas exhaust duct design

The exhaust tube's diameter is also the same one than for part 1. :

$$d = 0{,}043m = 4{,}3cm$$

4. COMPARISON BETWEEN OLD AND RENEWED HOUSE

To conclude this, we will compare the results of the three different solutions: EXISTING HOUSE part, RENEWED HOUSE part 1 and RENEWED HOUSE part 2.

PART	$\vartheta_{V,des}$ [ºC]	$\vartheta_{R,des}$ [ºC]	T [m]	m_H [l/s]	$N_{CIRCUITS}$	L (m)	ΔP [kPa]	$N_{MANIFOLDS}$	P_{BOILER} [KW]	$\dot{m}$ [kg/h]	ANNUAL COST
EXI. *(2.)*	42,78	37,78	0,15	0,30	9	731,12	27,11	1	8,53	0,71	1253,7
NEW 1 *(3.1.)*	32,53	27,53	0,15	0,16	10	866,09	17,95	2	4,46	0,37	658,6
NEW 2 *(3.2.)*	33,81	28,81	0,2	0,15	9	648,18	15,72	1	4,46	0,37	658,6

Table 34. Three solutions comparison

When looking at the values of the part EXISTING HOUSE *(point 2.)* and comparing them to RENEWED HOUSE 1 *(point 3.1.)* and RENEWED HOUSE 2 *(point 3.2.)* we can clearly see that the latest are much better for obvious reasons. The enclosures for parts RENEWED HOUSE 1 and RENEWED HOUSE 2 had been modified so that the thermal flow density was lower. So, the comparison that really makes sense is between RENEWED HOUSE 1 and RENEWED HOUSE 2.

For RENEWED HOUSE 1 we just recalculated all the installation for the new thermal flow density, while for the part RENEWED HOUSE 2 we imposed a pitch of 0,2m, which is the ideal for only heating systems, which is what we have. Therefor we can see that the values for RENEWED HOUSE.2 are better than those for RENEWED HOUSE.1, not in terms of fuel consume (and fuel annual cost), but in terms of number of circuits, manifolds and circuits' total length.

5. REFERENCES

[1]: Uponor technical approvals, Manual climatización invisible, 2014.

[2]: Reglamento de la Instalaciones Térmicas en Edificios (RITE), 2011

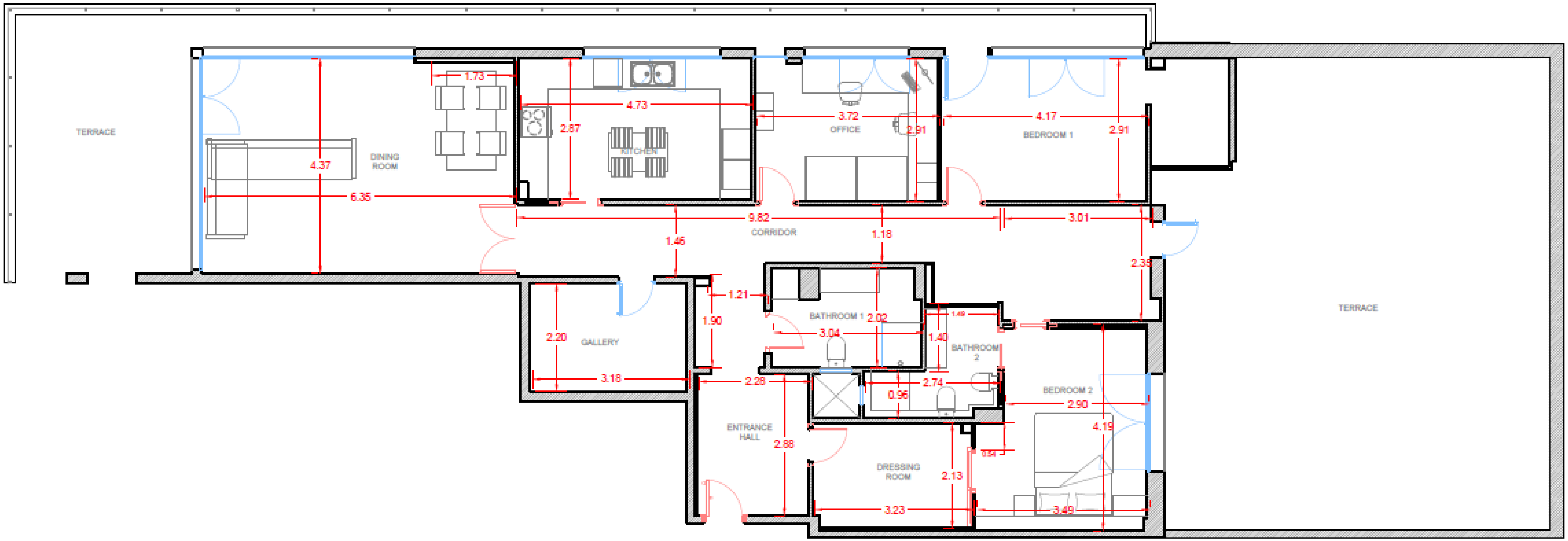
[3]: Sistemas de calefacción y refrigeración de circulación de agua en superficies (Standard UNE-1264), 2013

[4]: Guía Técnica Condiciones Climáticas Exteriores, Ministerio de España, 2010

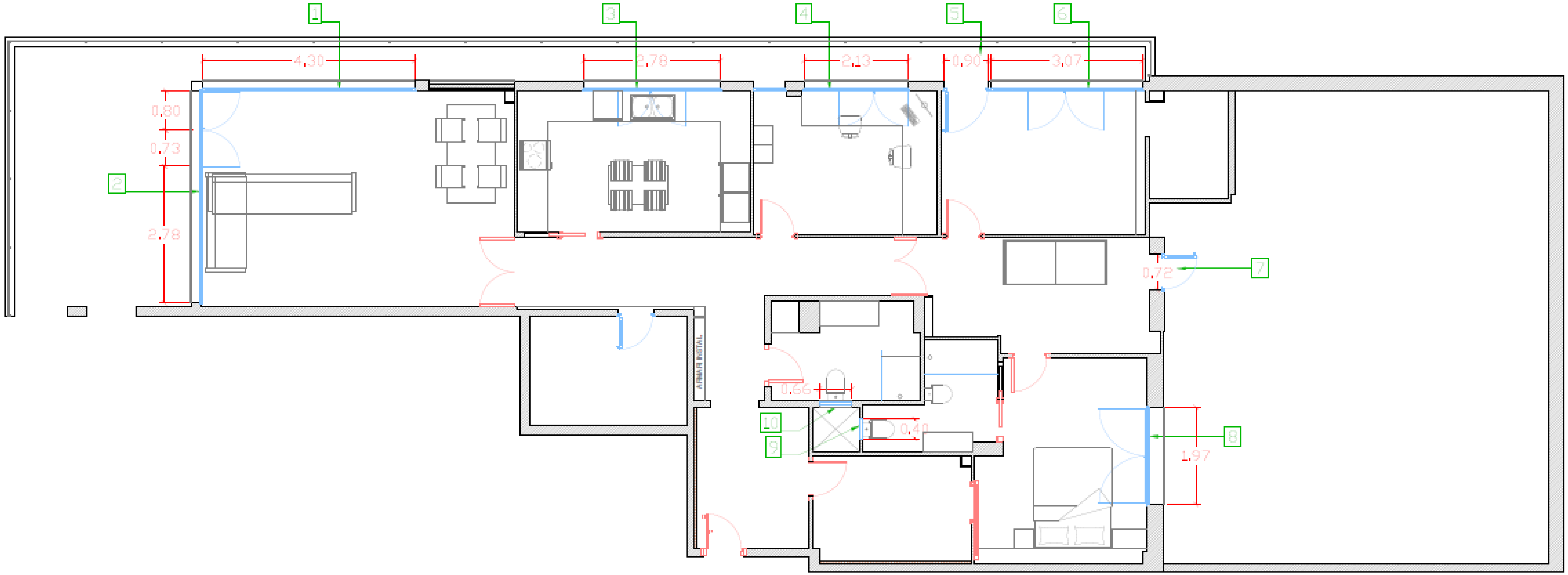
ANNEX 1

EXISTING HOUSE 1. BUILDING DESCRIPTION (Without Dimensions)

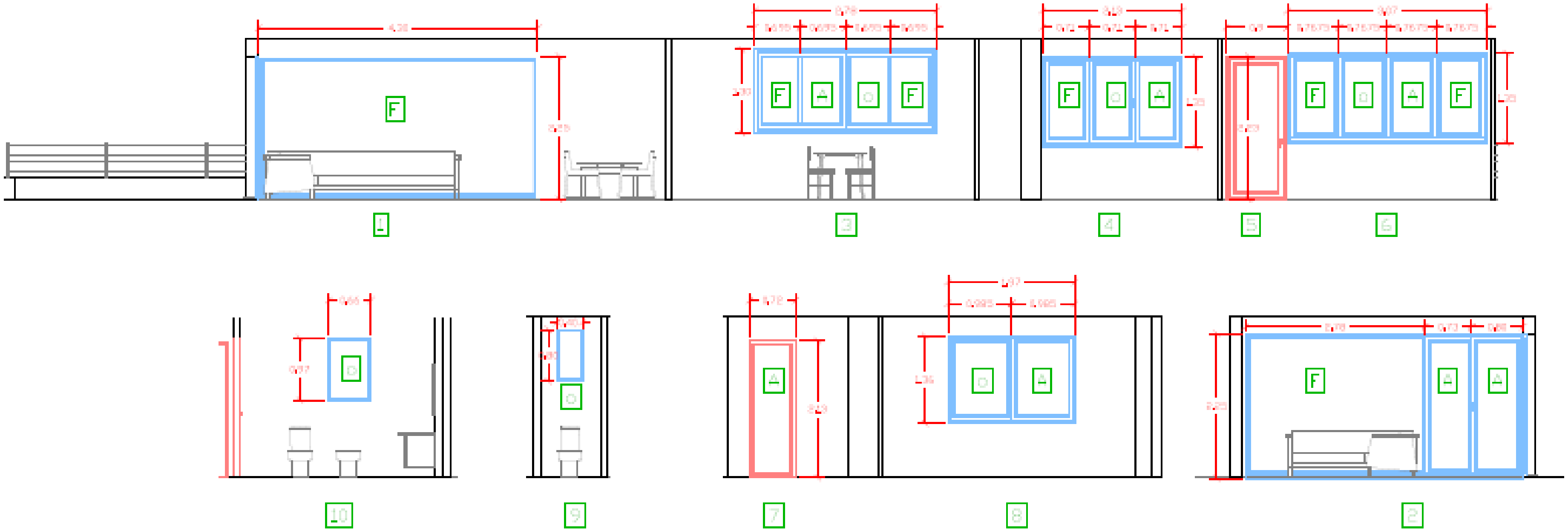
EXISTING HOUSE 1. BUILDING DESCRIPTION (With Dimensions)



EXISTING HOUSE 1. BUILDING DESCRIPTION (Glass Positions)

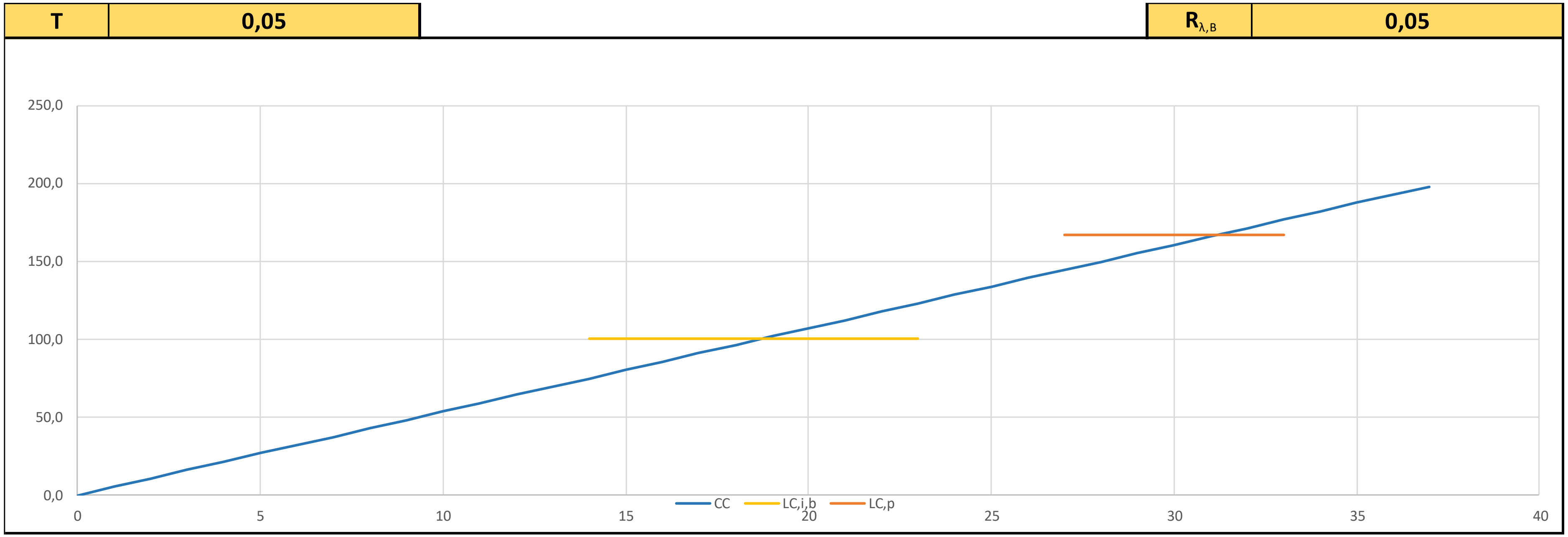


EXISTING HOUSE 1. BUILDING DESCRIPTION (Glass Dimensions)

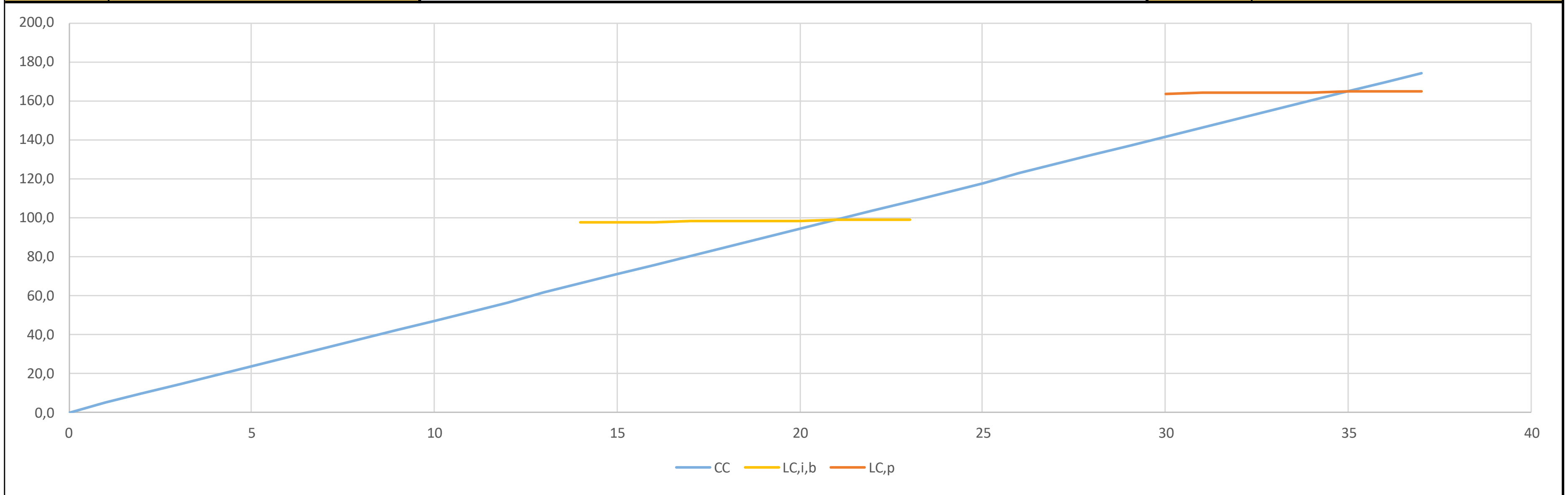


ANNEX 2

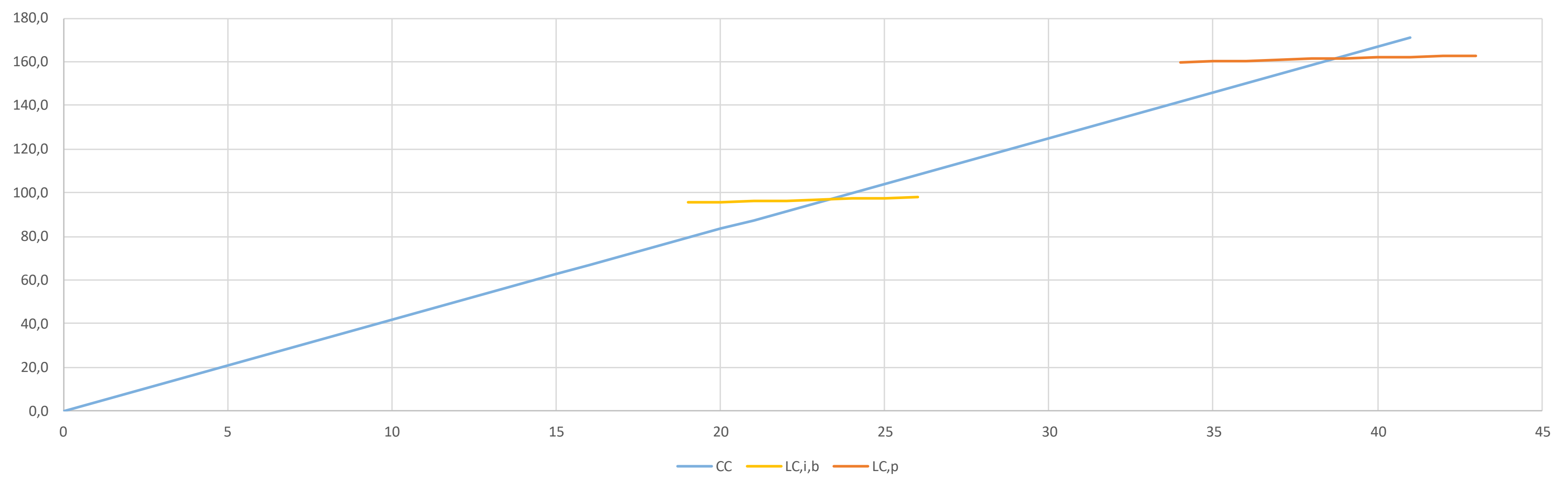
EXISTING HOUSE 3.1. CHARACTERISTIC AND LIMIT CURVES



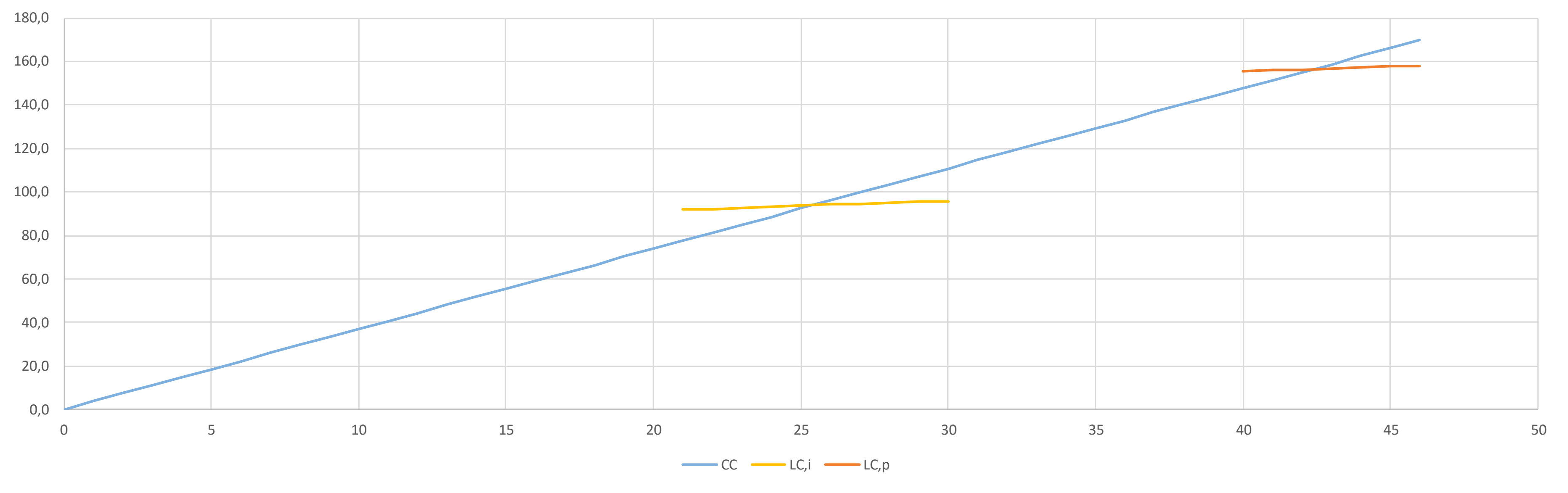
T	0,1		$R_{\lambda,B}$	0,05



T	0,15		$R_{\lambda,B}$	0,05

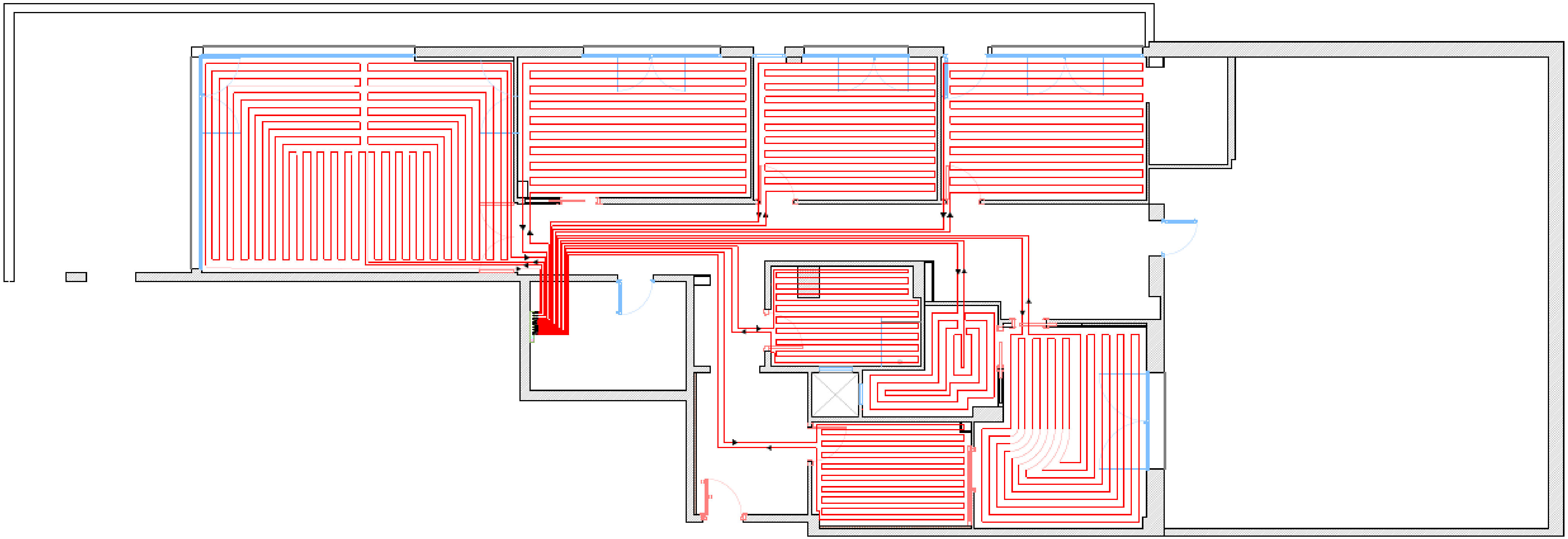


T	**0,2**		**$R_{\lambda,B}$**	**0,05**

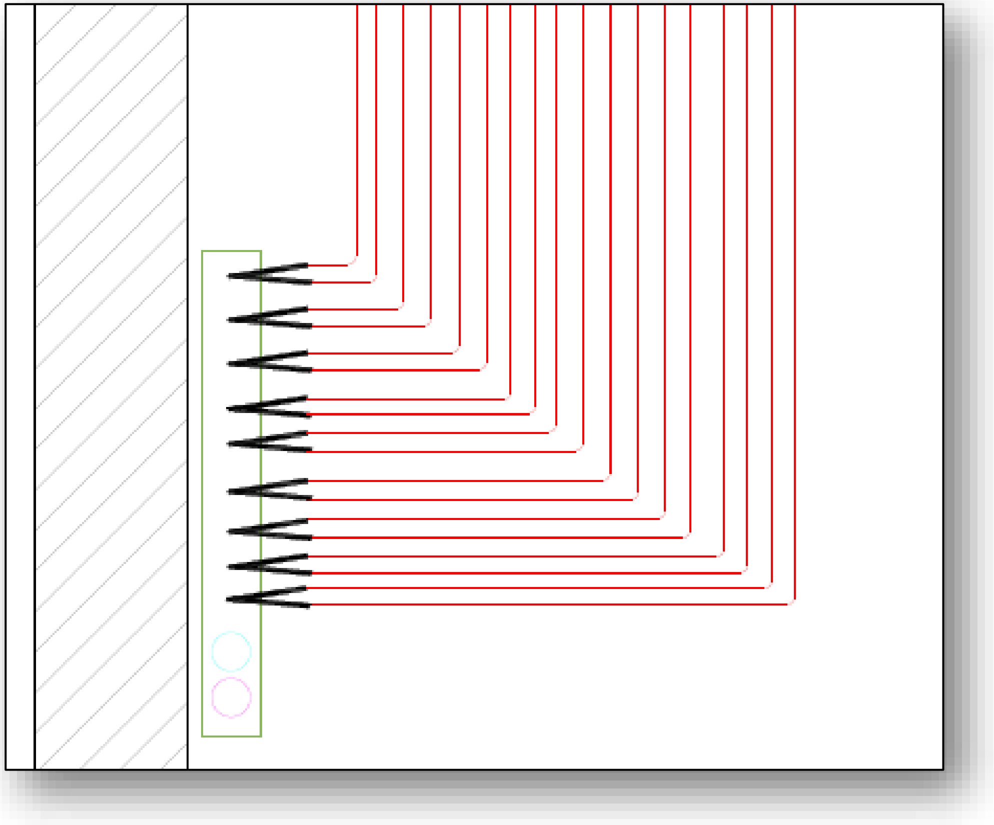


ANNEX 3

EXISTING HOUSE 3.3. Design Fluid Temperature (Tubing)

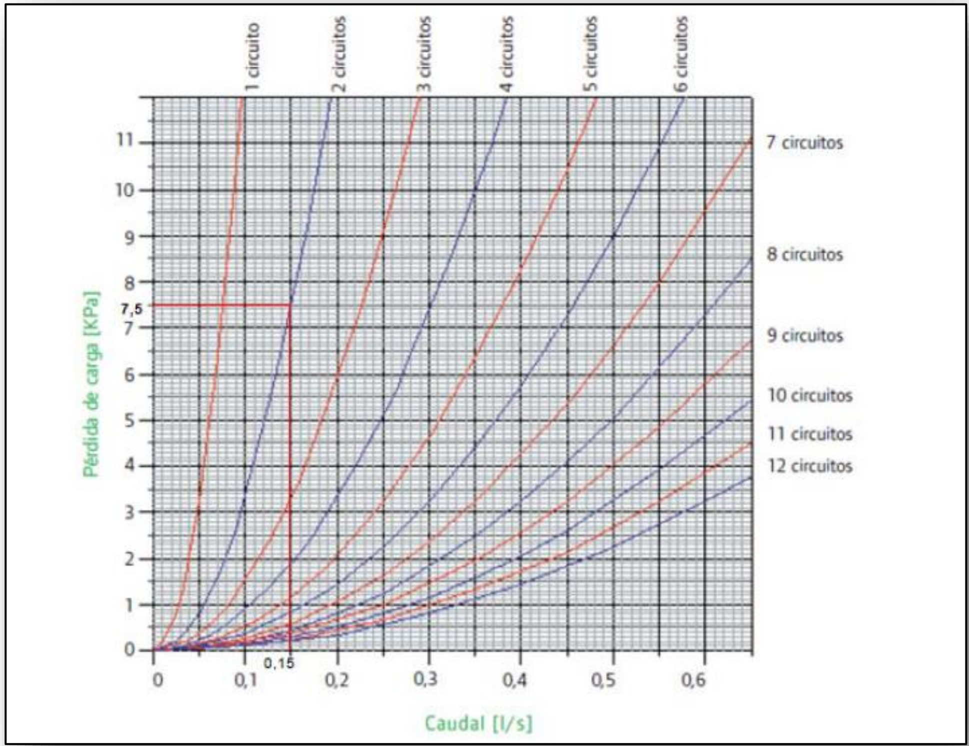


EXISTING HOUSE 3.3. Design Fluid Temperature (Manifold)



ANNEX 4

EXISTING HOUSE 6 .PRESSURE DROP



ANNEX 5

EXISTING HOUSE 6. MOODY'S DIAGRAM