Performance evaluation of building thermal mass coupled with photovoltaic/thermal panels

Master thesis report

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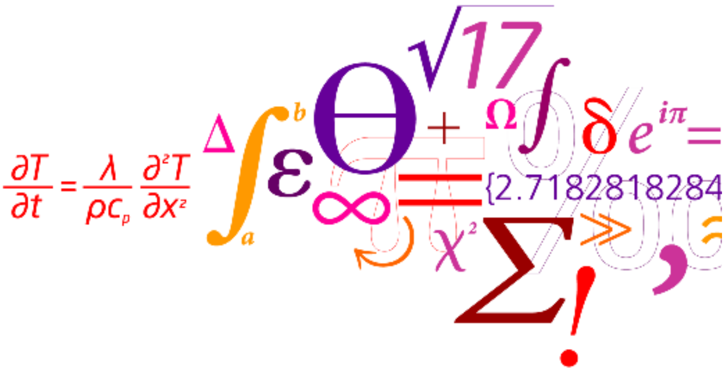

Abstract

Buildings´ energy consumption is accounted for a share of 40% of the overall energy consumed globally. Many efforts have been directed towards the improvement of the existing technologies and the implementation of innovative techniques in order to reduce the energy needs.

This report, *Performance evaluation of building thermal mass coupled with photovoltaic/thermal panels*, aims at highlighting the benefits and the disadvantages of the combination of a thermal storage consisting of the thermal mass of the dwelling and photovoltaic/thermal collectors (PV/Ts). The report is related to the Solar Decathlon Europe 2014 competition and team DTU´s dwelling, *Embrace*, represents the building this thesis is based on. Special focus is given to the implementation of the night radiative cooling strategy, which exploits the nocturnal long-wave radiation emission towards the sky to cool the water flowing inside the solar collectors. Various combinations involving different structures and different systems are analyzed. Since Solar Decathlon competition will take place in Paris, evaluations will be carried out for both Copenhagen and Paris climates.

The hydronic system is designed to include PV/T collectors, heat pump, domestic hot water tank, water storage tank (buffer tank), embedded system and auxiliary devices. The heating and cooling needs of the house will be addressed by the radiant system.

Preliminary calculations have been performed to size the embedded system, comparing the results of two different standard: EN1264-2 (2008) and EN 15377-1 (2008). Further comparisons involve a commercially-available software, MIRAGE. Analysis shows that the results obtained by EN1264-2 calculation method are in good accordance with the outcomes of the software.

TRNSYS simulations represent the core of the report. For the purposes of the final evaluation, indoor comfort conditions, devices´ operational time and energy consumption are taken into account.

Increasing building´s thermal mass significantly affects the performance of the system, resulting in better indoor conditions for the occupants and generally less frequent activation of the heat pump and the auxiliary devices (pumps, etc.), especially if the system is equipped with the water storage tank. Obviously, this is beneficial from the energy consumption point of view.

Thermal mass also affects the exploitation of the night radiative cooling: long-wave radiative losses increase in buildings with higher thermal capacity.

Results indicate that the presence of the water storage tank can more than double the energy consumption (with respect to the cases in which the tank is absent), since the heat pump is activated more frequently.

The tank also affects the exploitation of the nocturnal radiative cooling to the sky, resulting in lower values of radiative losses from the PV/Ts.

Based on the considerations mentioned above, conclusions are drawn. From the energetic point of view, the most beneficial combination appears to be the one involving high building´s thermal capacity but excluding the buffer tank. This results in reduced energy consumptions and satisfying indoor comfort conditions for the occupants (even though they are better in case the system is equipped with the tank). Night radiative cooling is also better-exploited. Economic considerations (pay-back time due to the savings, etc.) are not taken into account.

Further investigations could involve different control strategies and phase change materials (PCMs) to increase the thermal mass in light structures.

Keywords: Solar Decathlon Europe, Embrace, thermal mass, photovoltaic/thermal panels, radiant heating and cooling, night radiative cooling, indoor comfort conditions, energy consumption.

Abstract

Gli edifici incidono significativamente sul consumo energetico globale (ad essi è attribuibile una quota del 40% del totale). Pertanto, la maggioranza degli sforzi è diretta verso lo sviluppo di nuove tecnologie e il miglioramento di quello esistenti, con lo scopo di ridurre i fabbisogni energetici totali.

La presente tesi, *Performance evaluation building thermal mass coupled with photovoltaic/thermal panels* (*Valutazione delle prestazioni dell'accoppiamento di massa termica dell'edificio e pannelli fotovoltaici/termici*), mira ad evidenziare i benefici e gli svantaggi nel combinare pannelli fotovoltaici/termici (PV/Ts) con la massa termica della struttura. Questo elaborato è associato al progetto portato avanti dal DTU per la competizione Solar Decathlon Europe 2014; l'edificio che prenderà parte alla competizione è alla base di tutte le considerazioni contenute in questa tesi. Particolare attenzione è dedicata all'attuazione della tecnologia di *free-cooling* denominata *Night Radiative Cooling*, che consiste nella perdita termica verso il cielo per radiazione ad onda lunga dalla superficie dei pannelli PV/T quando il refrigerante è fatto circolare di notte. Le combinazioni incluse in questo elaborato coinvolgono diverse strutture e sistemi differenti. La competizione Solar Decathlon Europe 2014 si concluderà a Parigi, quindi le simulazioni implementano due condizioni climatiche, Copenaghen e Parigi.

I sistemi idronici sono disegnati per includere i collettori PV/T, una pompa di calore, un serbatoio per l'acqua calda sanitaria, un ulteriore serbatoio di accumulo, i sistemi radianti a pavimento e tutti gli ausiliari di sistema. Al sistema idronico complessivo è richiesto di soddisfare i fabbisogni dell'edificio, durante la stagione di riscaldamento e quella di raffrescamento.
Il primo passo nel completamento dell'elaborato è il dimensionamento del sistema radiante, i cui calcoli si basano su due diverse norme internazionali: EN1264-2 (2008) e EN15377-1 (2008). I valori ottenuti sono poi comparati con i risultati di MIRAGE, programma disponibile in commercio. L'analisi evidenzia che i risultati del software sono in buon accordo con quelli della norma EN1264, mentre sono più distanti da quelli della norma EN15377.

Il cuore dell'elaborato sono le simulazioni in ambiente TRNSYS. I parametri presi in considerazione per la valutazioni delle prestazioni complessive sono le condizioni di comfort termico dell'ambiente interno, i tempi di attività dei componenti del sistema e il relativo consumo energetico degli stessi.
L'aumento della massa termica dell'edificio influenza significativamente le prestazioni del sistema, con l'effetto di ottenere migliori condizioni di comfort dell'ambiente interno percepite dagli occupanti e, generalmente, l'attivazione meno frequente della pompa di calore e di molti dispositivi ausiliari (pompe, ecc.), specialmente se il sistema è equipaggiato con il serbatoio di accumulo a valle di

quello per l'acqua calda sanitaria. Pertanto, l'aumento della capacità termica delle strutture ha benefici anche dal punto di vista del consumo energetico.
La massa termica della struttura influenza anche lo sfruttamento del *free-cooling* notturno: le perdite radiative per emissione verso il cielo aumentano in caso di edifici con maggiore capacità termica.
I risultati indicano che la presenza del serbatoio di accumulo può più che raddoppiare i consumi energetici globali (con riferimento ai casi in cui il sistema non è equipaggiato con il serbatoio), principalmente a causa dell'attività più frequente della pompa di calore.
Inoltre, il serbatoio di accumulo influenza anche lo sfruttamento del *Night Radiative Cooling*, risultando in minori emissioni notturne verso il cielo in caso in cui esso sia presente.

Le conclusioni sono tratte con riferimento alle considerazioni precedentemente esposte. Dal punto di vista dei consumi energetici, la combinazione più favorevole è quella che coinvolge elevata massa termica dell'edificio, escludendo però il serbatoio di accumulo. Questo corrisponde a minori consumi energetici, garantendo condizioni di comfort dell'ambiente interno soddisfacenti (tuttavia, il comfort termico è migliore nei casi in cui il sistema è equipaggiato con il serbatoio). Anche il *free-cooling* notturno è sfruttato più favorevolmente. Dall'elaborato sono state escluse considerazioni economiche (tempo di ritorno degli investimenti, ecc.).

Ulteriori analisi potrebbero coinvolgere strategie di controllo diverse o *Phase Change Materials (PCM)* per aumentare la capacità termica dell'edificio nel caso di strutture leggere.

Parole chiave: Solar Decathlon Europe, Embrace, massa termica, pannelli fotovoltaici/termici, sistemi per il riscaldamento/raffrescamento radiante, night radiative cooling, condizioni di comfort interno, consumi energetici.

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS

Photovoltaic/thermal panels: PV/Ts

Domestich Hot Water: DHW

Water storage tank: WST

Phase change materials: PCMs

Heat pump: HP

Radiant floor: RF

Ground floor: GF

First floor: FF

1. INTRODUCTION

1.1 General presentation

Building sector accounts for approximately 40% of the world´s overall energy consumption (International Energy Agency, 2012) and, due to the lacks either in design or in operation of devices, appears to be the one with the greatest potential of being more efficient.

First of all, it is important to highlight the difference between “Energy saving” and “Energy Efficiency”. Saving energy could be very simple, starting from the very basic actions everyone has several times a day: turning off a light when not necessary, lowering or increasing the indoor temperature set-point respectively in winter and summer conditions, etc. On one hand, these adjustments could surely lead to lower energy consumptions, but one the other hand they present several disadvantages. Thus in fact, the possibility for the user to choose his/her own comfort conditions is reduced. The real challenge is the “Energy Efficiency” which means consuming less energy while guaranteeing at least the same comfortable environment to the occupants, allowing them to live, study, work in a place as stimulating as possible to achieve their goals in every aspect of life.

The incoming shortage of “traditional” energy resources is pushing professionals to consider every project in a different way compared to the past. The new growing idea is to benefit from every exploitable resource. Using the same device, traditionally considered usable only to take advantage of the day-time solar radiation, during night-time (when the Sun is set and there is not any radiation to benefit from) is a part of this idea. More detailed information will be explained in the following chapters.

Strictly related to the concept of using every exploitable source and in the way it´s available (replacing high temperature sources with lower temperature sources due to better-performing technology for instance), the idea of storing energy, allowing it to be used in a moment that doesn’t have to be the one of the actual production, is spreading among new projects. This idea could satisfy many needs and could be implemented in systems such as those described in this report which activate the structure of the building in order to smoothen the peak in power demand. The consequent dynamic behavior of the dwelling could be considered as an “Energy Efficiency” strategy that results in savings in energy consumptions.

A separate discussion is set aside for economic considerations. All of the considered technologies are more expensive (at the moment) compared to other alternatives, but research is focusing on their potentialities and the price is expected to decrease in the next years.

1.2 Motivation and objectives

Starting from these basic concepts, this study intends to evaluate the possibility to couple an innovative energy source as the sky during the night-time and an innovative storage system to store energy. This technology was born a few years ago, therefore it needs to be evaluated, in terms of effectiveness, efficiency, energy savings, feasibility and from the economical point of view.

The aim of this report is to highlight the benefits and the disadvantages of the system considered, focusing mostly on the energy aspect, by providing information from both qualitative and quantitative points of view.
Firstly, calculation will be carried out and then their results will be implemented in tools as simulations software (Mirage, TRNSYS), in order to consider the interaction of night-time radiative cooling on a modern low-energy-consuming building. Then, simulation programs will be the main instruments through which the final results will be obtained. Simulations will involve two different climate conditions (Copenhagen and Paris) and will be carried out for several combinations of the system and the thermal mass, each of which will be analyzed, evaluated and compared to the others.

The final performance evaluation will be based on the fulfillment of the indoor comfort requirements, the periods of activity of the system´s components, the energy consumptions of single components and of the overall system.

Every step of this report will involve *Embrace*, team DTU´s project for the Solar Decathlon Europe 2014 (SDE) competition, despite this report and the competition pursue mostly different goals. In fact, SDE 14 objectives extend beyond the energy performance and its evaluation. The dwelling has been considered as starting point, but due to later changes in the design and to initial lack of information, this report and the development of Embrace will follow parallel directions, since for the purposes of this report and intermediate design has been taken into account instead of the ultimate one.

2. TECHNOLOGY OVERVIEW

Solar technologies regarding the production of electricity and hot water are well known and wide spread in the midst of the energy market, although they are usually installed separately. Studies in the latest years resulted in a new technology, which combines both of the productions mentioned above, the Photovoltaic Thermal panels (PV/T panels or PV/Ts). This kind of a product seems to be very promising (Eicker & Dalibard, 2011), despite some issues that inevitably occurs when a new technology appears on the market.

PV/Ts basically consist of solar photovoltaic cells and pipes placed underneath them in as good as possible thermal contact. PV/T panels are therefore able to convert incoming solar radiation into electricity and heat to be used for hot water purposes, but the balance, in terms of energy produced, is strongly affected by the decisions of the designer. Depending on the choices and on the applications, the main application could be either the electrical production or the thermal production. In the first case, since photovoltaic cells suffer from a drop in efficiency with the rise in temperature due to increased resistance, the fluid flowing inside the pipes has to cool down the photo-voltaic (from now on PV) cells as much as possible, in order to keep the efficiency as high as possible. In the second case the fluid is expected to exit from the panels loop at a certain temperature, which depends on the application (domestic hot water, space heating, etc.) fed by the fluid. The design of the solar collectors will be based on the main production chosen, but the production depends on the position they are placed, whether and the back side is insulated or not, etc.

The typical structure of PV/Ts presents some differences in comparison with any other photovoltaic panel, whose purpose is to collect the solar radiation also for thermal production. It basically consists of a glazing layer above the semiconducting cells (in order to protect them from weather conditions), the cells themselves, a glazing layer below the cells, a heat-absorber plate (glued to the glazing layer) in strict thermal connection with the underlying pipes where the cooling medium flows through. In addition, the panels can include a glass layer and an air layer (the so called “covered” solution) which aims to reduce the thermal losses towards the surroundings. Overall structure is shown in Figure 1. Incoming solar radiation is firstly collected by semiconducting photovoltaic cells and converted into electricity. The non-exploited fraction of solar radiation is able to reach the heat-absorber plate and be transferred to the cooling medium. It is clear that in order to achieve good heat transmission performances towards the fluid flowing in the pipes, a good thermal connection between the plate and the pipe has to be guaranteed.

Depending on the main production, either electricity or hot water, there are other techniques that could be implemented to counter-act the decrease of efficiency due to increasing temperature in the first case and to limit the heat loss on the backside of the collectors in second case. If the electrical production has the priority, it’s necessary to cool down the cells and the simplest solution is to let the air flow underneath them by leaving an air gap. If the priority is given to the thermal production, the most common solution consists of an insulation layer which prevents heat from flowing backwards to the surroundings.

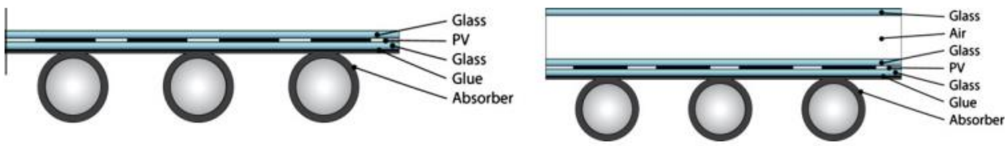


Figure 1: Uncovered and covered PV/T collectors - Eicker & Dalibard, *Photovoltaic-thermal collectors for night radiative cooling of buildings*, 2011

A new and interesting field of research involving PV/T panels concerns the use of the night radiation towards the sky to cool down the fluid flowing into the pipes. Cold water produced offers several possibilities that will be discussed later in this chapter.

Radiative cooling is based on the heat loss by long-wave radiation emission towards the sky, whose temperature (T_{sky}) can fall below 0°C (even -10°C) in clear sky conditions (Eicker & Dalibard, 2011). Thus, the heat exchange could rely on temperature differences between collectors and sky of about 20÷30 K. The long-wave heat loss is strongly affected by the water content of the atmosphere. In particular, 90% of the sky radiation is originating from the first kilometer above the ground, and 40% from only the 10 meters layer above the ground (Bliss, 1961). Therefore the radiation strongly varies from site to site.

Since the PV/T panels are mostly designed for day-time radiation and to prevent them from high thermal losses, to ensure a good night-time radiative heat exchange towards the sky usually the outer glazing layer is removed (Eicker & Dalibard, 2011), despite the penalization in day-time performances.

The main features of PV/T technology could be summarized as follows:

- Due to the heat removed by the circulating water, the photo-voltaic cells can be cooled down, keeping their electrical efficiency as close as possible to the nominal value.
- They allow the water to be heated to different temperatures, thus satisfying different demands of different applications.
- Although this is an effective method, it causes the thermal component to under-perform compared to a solar thermal collector due to increased heat loss.
- Since it´s a combination of two different technologies in the same product, PV/Ts allow to decrease the amount of required surface for electrical and thermal needs.
- The production of cooling during night-time is allowed.

The cold water produced could satisfy many needs, depending on its amount and its temperature. Cooling energy carried by water is usually stored in a tank and if the temperature of the water in the tank is low enough it can be circulated into the pipes of the radiant systems of the building to cool down the structures during night, allowing them to absorb the heat loads during the next day. This idea belongs in general to the concept of "Low temperature heating and high temperature cooling"; in fact, in the applications in which the mentioned strategy is involved, it is possible to integrate different energy resources, even those which are supposed to have a low-grade of exploitability. For this reason it is possible to state that the use of water cooled by the night radiation towards the sky is a particular case of the concept presented above.

Since the system will likely be equipped with a water storage tank (in addition to the necessary domestic hot water tank), the possibility of avoiding the tank and let the water coming from PV/Ts circulate directly in the radiant system will be evaluated.

This process is strongly linked to the thermal mass of the building, more the thermal mass is, more the structures can store cool to face the heat loads of the next day (until a certain value of thermal capacity, after which the storage capability reaches an asymptotic value).

In case of light structures or insufficient thermal mass, the thermal mass can be increased by using particular materials, such as the phase change materials (PCMs). During day-time they are able to absorb the heat gains inside the house especially due to their melting process (melting temperature for the most commonly used PCMs in building industry usually is in the range 22÷24°C (Climsel C24 Datasheet) in order to match the range indoor comfort lies) and during the night they are discharged by cold water produced thanks to night-time radiative cooling effect from the PV/Ts. In ideal operating conditions, the next morning the PCMs are completely discharged and the process of absorption/release of heat can start again. PCMs can be integrated in several ways in the hydronic system (even though they could be discharged by ventilation air, however this requires high air flow rates which feasibility needs to be analyzed). PCMs implementation though is not included in this report but it could be an interesting extension of the studies presented in this report.

3. SOLAR DECATHLON EUROPE 2014

3.1 The competition

The Solar Decathlon is an international competition created by the U.S. Department of Energy which aims to stimulate the participant universities from all over the world to meet, design and operate an energetically self-sufficient house, grid connected, using solar energy as the only energy source and equipped with all the technologies that permit high energy efficiency (Solar Decathlon 2014 - Rules, Introduction chapter, 2012).

The first Solar Decathlon competition was held in 2002, organized among U.S. universities, but after October 2007, thanks to an agreement signed between the Ministry of Housing of the Government of Spain and the United States Government, the contest spread out towards Europe and resulted in the Solar Decathlon Europe (SDE) competition. The first edition of the SDE contest took place in 2010 in Madrid, as well as the second one in 2012. The third will be held at "Cité du Soleil" in Versailles, between June and July 2014.

The SDE 2014 Organization´s goal is to contribute to the knowledge and diffusion of solar and sustainable housing and therefore its main objectives are (Solar Decathlon 2014 - Rules, Introduction chapter, 2012):

- To stimulate students participating in the competition to think creatively and challenge them to develop original solutions in order to reach the goals fixed by the Organization.
- To increase awareness on the benefits and opportunities offered by the use of renewable energies and sustainable construction
- To promote materials and systems those reduce the environmental impact over its whole lifetime, optimizing its economic feasibility and guaranteeing high comfort and savings to occupants.
- To educate the general public about responsible energy use, energy efficiency, showing a high efficient use of renewable energy and the technologies available to help them to reduce their energy consumption.
- To underline the correct order of intervention in order to decrease energy demand: first reducing the building energy consumption and increasing its energy efficiency and afterwards integrating solar active systems and other renewables technologies. Moreover the building systems must be selected and dimensioned using environmental and cost-effective criteria.
- To encourage the use of solar technologies
- To support the integration of new solar technologies in the construction materials for the building envelope
- To show that high-performance and new-concept solar houses can be environmentally sustainable and, at the same time, comfortable, attractive, affordable.

The 2014 Solar Decathlon Europe Organization in France, is intending to provide habitats that meet the triple challenge (energy, environment and society) we are all facing; thus, it decided to focus on six items as follows (Solar Decathlon Europe 2014 - Rules, 2012):

- Density: one of the purposes of the contest is to support projects of collective housing rather than individual houses.
- Mobility: it refers primarily to the location of housing relative to “resources”, i.e. shopping, work, leisure. Moreover, the question of energy coupling between positive-energy buildings and electrical transportation system (or, in general, any other kind of vehicle which combines the functions of intermittent energy storage device and transportation facility) is highlighted.
- Sobriety: it refers mainly to the limitation of energy demand and thus energy consumption. Regarding the 2014 competition, a limit for the photovoltaic power installed is established, as well as an important evaluation of energy efficiency and a strong incentive to produce and consume wisely.
- Innovation: since it’s an academic competition, innovation is the heart of every component of the project (architecture, construction, energy systems, furnishings, house appliances)
- Affordability: in the midst of a major economic crisis, the financial factor will be assessed as a determining factor of each proposal as well as a bold criterion of the final project presented in 2014 in Versailles.
- The project in its environment and the project in the competition: each participant’s project should fit the cultural, climatic and social contexts of the team’s home-region as well as guarantee high performances during the short period of time during which the competition actually takes place.

In addition to the concepts previously stated, the competition aims to demonstrate that a well-designed house can satisfy its needs of electricity for lighting, cooking, washing clothes and dishes, powering home and home-office electronics, maintaining comfortable levels of indoor temperature and air quality. SDE integrates in the European Union goals for 2020, especially for what regards energy consumption of dwellings but also for what concerns the reduction in the use of the polluting energy sources (the so called “20-20-20” - (Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the Energy Performance of Buildings, 2010) - saving 20% of primary energy consumption, reducing 20% of the greenhouse gas emissions and producing 20% of the energy from renewable energy sources).

3.2 Rules and scoring

The Solar Decathlon Europe Rules are meant to meet the Organization objectives and to promote a fair and interesting competition among teams. The rules for SDE 2014 are based on those of the previous competition but some changes occurred, emphasizing urban sustainability, energy efficiency, mobility, innovation and cultural diffusion.

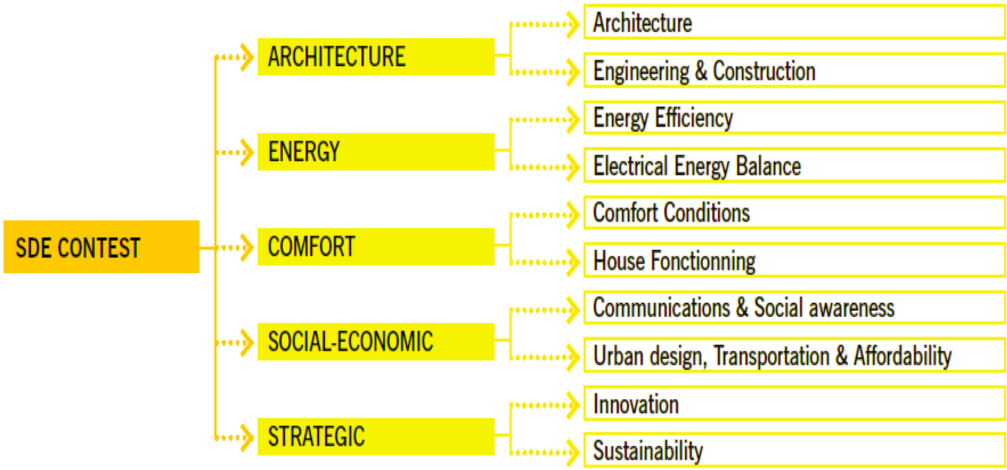


Figure 2: Contest categories for the Competition. ***Solar Decathlon Europe 2014 – Rules, Rule 13: General contest information***

Every house will be provided with a solar envelope, whose maximum dimensions are reported in Figure 3.

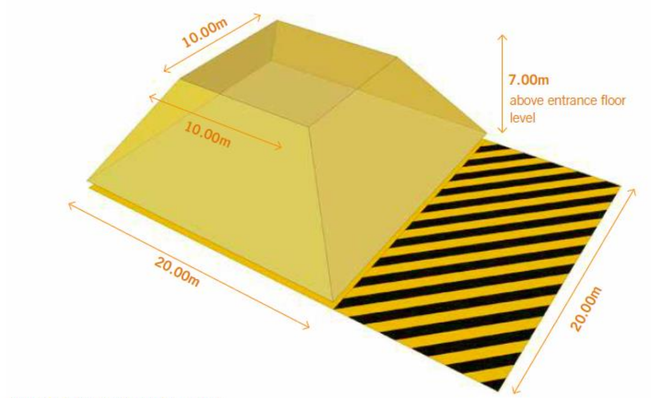


Figure 3: Maximum dimensions allowed for the solar envelope. ***Solar Decathlon Europe 2014 – Rules, Rule 5: The solar envelope***

In the Solar Decathlon Europe 2014 (SDE 2014) there will be three ways to score points:

- Jury evaluation: composed by internationally renowned experts in the different sectors involved in the project, a multidisciplinary jury will use their experience and knowledge to evaluate the project.

- Task completion: the teams will obtain points for successfully completing the requested tasks.
- Monitored Performance Scoring: the house will be continuously monitored during the competition period and measurements of the considered values will be carried out constantly. The scoring is based on the approach to the goal predetermined in the contest.

The distribution of the possible achievable points among all the contest categories could be summarised as follow:

No.	Contest/Sub-contest Name	Contests Points	Sub Contests Points	Assigned by
1	Architecture	120		Jury
2	Engineering & Construction	80		Jury
3	Energy Efficiency	80		Jury
4	Electrical Energy Balance	120		
	4.1 Load consumption per surface area		40	Monitored performance
	4.2 Positive electrical balance		15	Monitored performance
	4.3 Temporary Generation-Consumption Correlation		25	Monitored performance
	4.4 House adjustment to network load state		25	Monitored performance
	4.5 Power peaks		15	Monitored performance
5	Comfort Conditions	120		
	5.1 Temperature		50	Monitored performance
	5.2 Humidity		10	Monitored performance
	5.3 Indoor Air Quality – CO2		15	Monitored performance
	5.4 Indoor Air Quality - VOC		10	Test
	5.5 Natural Lighting		20	Test
	5.6 Acoustic		15	Test
6	House Functioning	120		
	6.1 Refrigeration		5	Monitored performance
	6.2 Freezing		5	Monitored performance
	6.3 Clothes Washer		20	Task + Monitored
	6.4 Clothes Drying		10	Task Completion
	6.5 Dishwashing		10	Task + Monitored
	6.6 Home Electronics		5	Task + Monitored
	6.7 Oven		5	Task + Monitored
	6.8 Cooking		5	Task Completion
	6.9 Hot Water Draws		20	Task Completion
	6.10 Dinner		15	Guests evaluation
	6.11 Water Balance		20	Counting
7	Communication and Social Awareness	80		Jury
8	Urban design, Transportation and Affordability	120		Jury
9	Innovation	80		Jury
10	Sustainability	80		Jury

Figure 4: Total score for different contest categories. ***Solar Decathlon Europe 2014 – Rules– Rule 14: General competition criteria***

3.2.1 Comfort conditions rules

Since the studies presented at a later stage in this report mainly deal with indoor climate and heating and cooling systems, the rules regarding the section "Comfort Conditions" are described, which belong to the sub-section "Rule 19 – contest 5".

The Organization and the jury appointed to evaluate the projects will focus on the capacity of the system to provide interior comfort through the control of

temperature, relative humidity, acoustic, lighting and indoor air quality, by measurements taken from the house during the competition period. In particular, indoor temperature, natural lighting, air quality (CO_2 concentration and the amount of Volatile Organic Compounds - VOC), humidity and acoustic performance will be taken into account.
The maximum score for this contest is 120 points (out of a total of 1000 points).

3.2.1.1 Temperature

Indoor temperature will be constantly measured by two (or in some cases three) sensors inside the building. During the competition, the SDE 2014 Organization will announce the temperature range for the next day, according to EN 15251 standard, in order to adapt comfort conditions to the weather. The temperature will be obtained according to the following equation:

$$T^{\circ}_{ea} = (T^{\circ}_{ed-1} + 0.8 \cdot T^{\circ}_{ed-2} + 0.6 \cdot T^{\circ}_{ed-3} + 0.5 \cdot T^{\circ}_{ed-4} + 0.4 \cdot T^{\circ}_{ed-5} + 0.3 \cdot T^{\circ}_{ed-6} + 0.2 \cdot T^{\circ}_{ed-7}) / 3.8$$

Equation 1

where $T°_{ea}$ is the average exterior temperature of the day and $T°_{ed-x}$ are the daily average temperatures of precedent days. Then, the available points range is calculated every day by the following expression:

$$T^{\circ}_{i,\min} < T^{\circ}_{i} < T^{\circ}_{i,\max}$$

where $T°_{i}$ is the indoor operative temperature and:

$$T^{\circ}_{i,\min} = 0.33 \cdot T^{\circ}_{ea} + 18.8 - 1$$
$$T^{\circ}_{i,\max} = 0.33 \cdot T^{\circ}_{ea} + 18.8 + 1$$

Equation 2

All available points are earned at the conclusion of each scored period by keeping the time-averaged interior operative temperature between a range of 2°C (±1°C) depending on the weather. Reduced points, scaled linearly, are earned if the indoor temperature is maintained in a range of 2°C below or above the full points range. Figure 5 shows the limits for the achievable points in the indoor temperature contest.

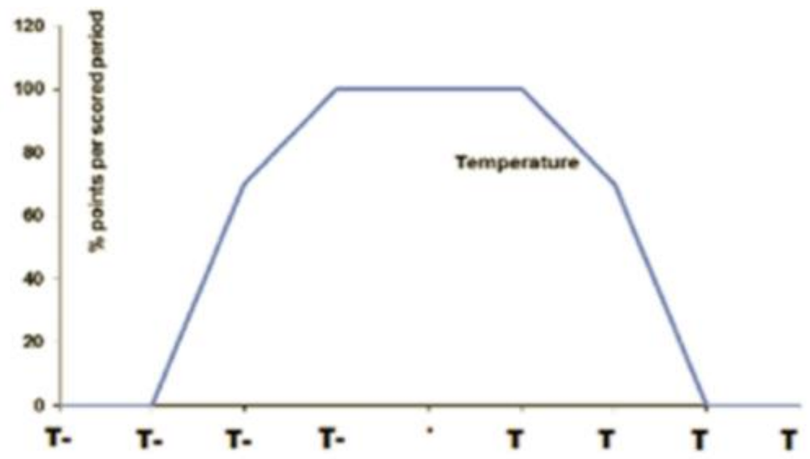


Figure 5: Earned points for different indoor temperature ranges. ***Solar Decathlon Europe 2014 – Rules – Rule 19, contest 5: Comfort conditions***

3.2.1.2 Relative humidity

Values of the relative humidity will be constantly measured. All available points are earned at the conclusion of each scored period by keeping the time-averaged interior relative humidity between 40% and 55%. Reduced points are earned if the time averaged relative humidity is kept between 25÷40% or 55÷60%.

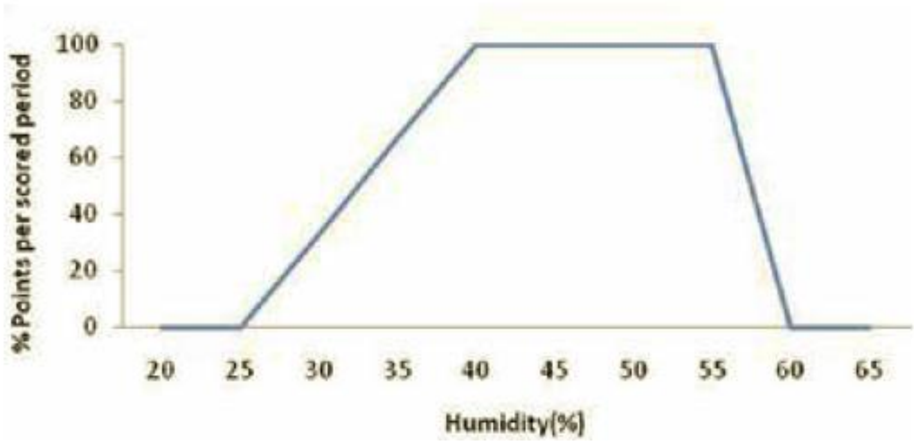


Figure 6: Earned points for different humidity ranges. ***Solar Decathlon Europe 2014 – Rules – Rule 19, contest 5: Comfort conditions***

3.2.1.3 Air quality – CO_2 and VOC concentration

The CO_2 concentration will be constantly measured, while the formaldehyde concentration will be measured punctually. All the available points are scored if the CO_2 concentration is kept below 800 ppm and the formaldehyde concentration below 30 µg/m^3 for the evaluation period. Reduced points are earned if the measured values don't fit within the intervals previously stated.

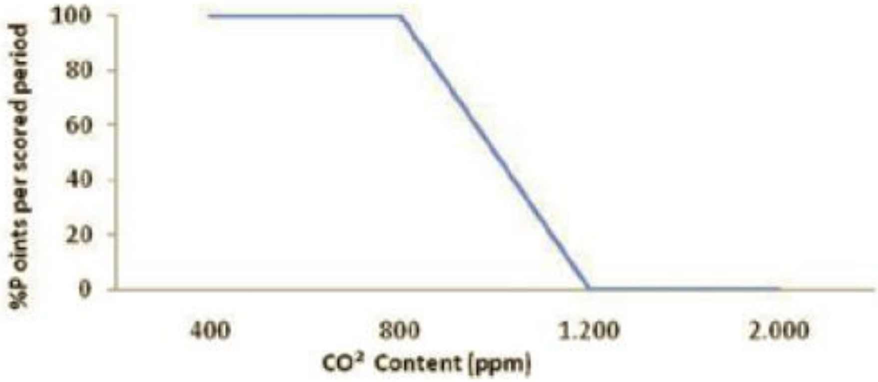


Figure 7: Earned points for different CO_2 indoor concentrations. ***Solar Decathlon Europe 2014 – Rules – Rule 19, contest 5: Comfort conditions***

3.2.1.4 Natural lighting

All available points are earned by keeping the Daylight Factor (lighting level/exterior lighting) above 4% during measurements period (cloudy sky).

3.2.1.5 Sonic environment and acoustic performance

The measured parameters to evaluate the acoustic performance in Versailles are the sound insulation from the inside, the reverberation time in the living room, the sound level of HVAC systems and all other active systems, inside and outside. Regarding the HVAC systems, all the available points are earned if the acoustic level is kept below 25 dB(A). No more details are given in this report, since it doesn't deal with the acoustic performances of the dwelling. Further information can be found in Rules of SDE 14 (Solar Decathlon Europe 2014 - Rules, 2012).

3.2.1.6 Passive evaluation period

During two consecutive days indicated in the Competition Calendar the participants will be allowed to use only "passive systems" for the heating and cooling systems of the houses. For the purposes of the competition this means that it's forbidden to use any kind of system which implements a thermodynamic cycle or internal heat or cool production devices. Pumps and fans will be accepted, but it won't be possible to use electrical heaters, chillers, heat pumps and other equipment that include a thermodynamic cycle, as well as batteries. Teams will have to plan their own strategies to maintain good comfort levels inside the dwellings during the passive evaluation period.

4. EMBRACE

The Solar Decathlon competition gives the opportunity to make engineers and architects work together in an iterative design process, from the beginning of the design phase to its ultimate steps. The result of this close collaboration is *Embrace*, which will be a living unit, a house that is brought to life by combining passive, architectural and active technical solutions in one building. Being the main purpose of SDE competition and the main goal of any participant team, electricity needs of the house will mostly be satisfied by its own electric production (thanks to the PV/T panels), even though *Embrace* will be connected to the electric grid for safety reasons.

4.1 House description

All over the world, there is a strong tendency of people moving from the countryside and smaller cities to bigger cities and it´s more than reasonable considering that this trend will be maintained in the future too. “Megacities” is a phenomenon, getting more and more common as a result of this increased urbanization and therefore the demand for dwellings is rising constantly. Thereby alternative solutions have to be found and Team DTU has focused on the concept of densification of the cities. Thus, *Embrace* is meant to be built not only on the ground, but also on the rooftop of existing buildings. It´s adjustable in regards to accessibility from the existing building and fitting the existing building. The name explains the building as it embraces the environment it is placed in, the users of the existing building and new inhabitants in the rooftop apartments. *Embrace* creates not only space for more people, it also includes people from the existing building by creating a more urban cityscape, with recreational areas that also give opportunity for gatherings with other people in the building.
The design of *Embrace* has been developed around three main concepts, *Smart*, *Save*, *Share*. “*Smart*” means that the design of the building itself and its orientation have been optimized to make it achieve the best possible performance regarding energy consumption, sustainability and indoor environment quality. Moreover, the dwelling has been provided the technical devices necessary for saving energy and using the solar energy in the best possible way. Finally, when living in this dense community, sharing space and technical devices is a big advantage. When sharing all of them, space in every unit will be saved by creating a compact living style and that´s the meaning of “*Share*” (Deliverable 3 - Team DTU for SDE 14, 2013).

Figure 8: ***Embrace*** **visualization.** ***Delivery 3, Project Manual – Team DTU for SDE 14 competition***

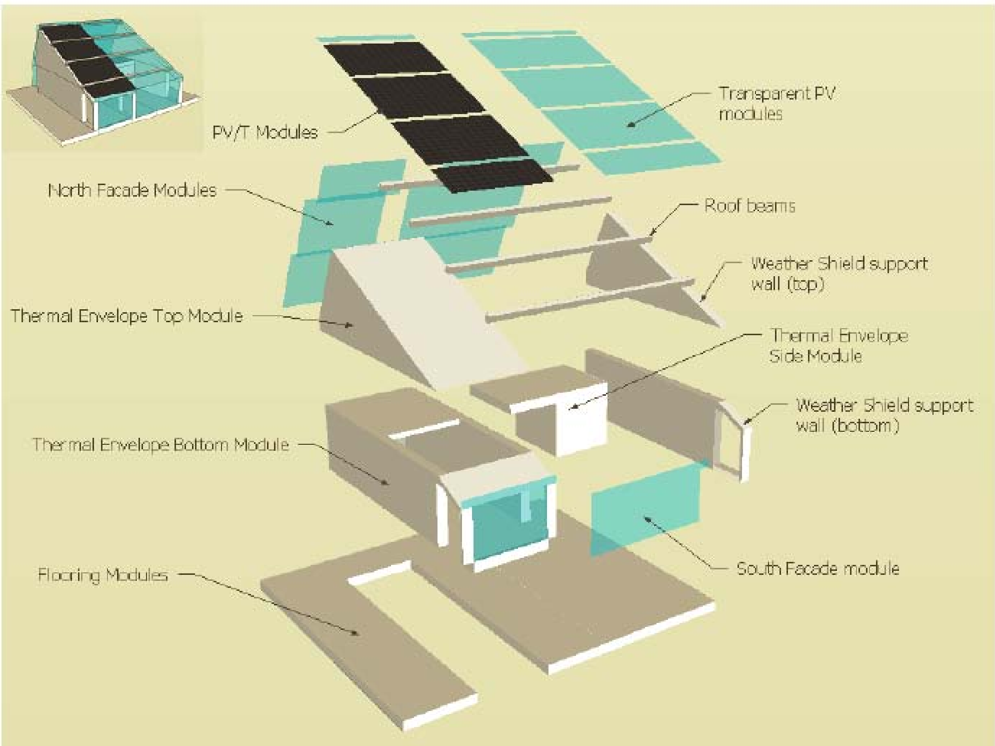


Figure 9: Representation of ***Embrace´s*** **blocks and main components.** ***Delivery 3, Project Manual – Team DTU for SDE 14 competition***

4.1.1 Separating envelope

The concept of *Embrace* is also about separation. In fact, the design consists of two different envelopes, a weather shield and the actual house (also called *thermal* *envelope*). The weather shield (the outer envelope) is meant to protect the inner structure (the actual living-dwelling) from the rain, the sun and the wind. Separation of the envelope gives the opportunity to use the outdoor area for a longer period of the year, because it´s free of rain and it provides a higher temperature than the actual outdoor temperature because of the solar gains in winter. Another advantage is the possibility to attach an urban garden under the weather shield and thereby bring the typical detached house from the countryside to the city center of the big cities.

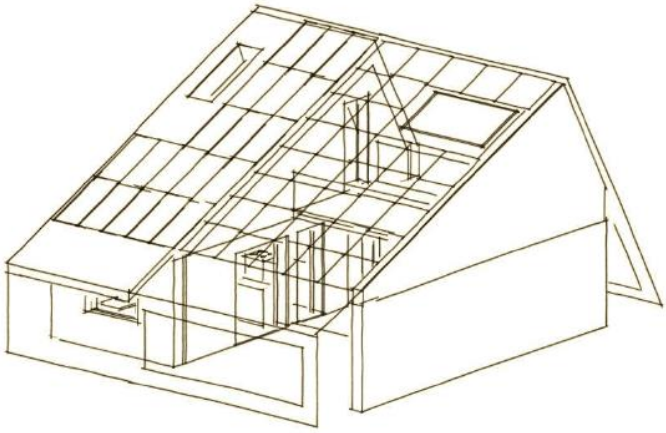


Figure 10*: Embrace* in a sketch view. *Delivery 3, Project Manual – Team DTU for SDE 14 competition*

4.1.2 Sharing functions

In order to build in a denser way, the house spacing and functions need to be reconsidered. Regarding this, rooms, functions and devices have to be evaluated for what people "need-to-have" and not for what is "nice-to-have". All extra spaces and devices can be extracted from the single dwelling and moved to a common place, where several dwellings or the entire community can share them.

4.1.3 Users

Modern-day customers desire more functions, space and resources than past-day customers. Through sharing, *Embrace* optimizes the use of functions and area, giving an idea of how to share and manage resources as much as possible without major investments, expenses or deprivation. The dwelling is suited for 2-2.5 persons, where the half person can be a child or an adult only living there for short time.

4.2 Load calculations

Since the competition is meant to stimulate participants to develop a strongly-innovative house, particular attention should be directed towards the energy efficiency of the dwelling. Moreover, the building has been thought to satisfy the strict regulation of the SDE14 committee both in Copenhagen's and Paris' climatic conditions. Since the two climates are different, as a result the power and energy demands to satisfy the needs of the building will be different. Both of the conditions have been evaluated in the design conditions (for winter and summer cases) and in "average" conditions (for winter and summer cases). The hypothesis and the input values used in the calculations are described in the next subchapters.

4.2.1 Weather shield

The main purpose of the weather shield is to protect the inner dwelling from all the weather distresses, especially rain, snow, wind, providing a warmer environment during winter, allowing its inhabitants to use this space even when outdoor conditions are adverse to outdoor activities. This feature is obtained through the solar gains and the weather shield will act similar to a "greenhouse". Despite the interesting and positive aspects of the weather shield (also called solar envelope) in winter-time, it could cause problems in summer-time due to overheating and the result could be higher temperatures than the outside, creating a less comfortable environment for people to live in. In order to prevent this, the solar envelope has to be provided with several openings, mainly consisting of two open facades (the East and West sides). In addition, it presents other openings on the South and North facades. Small lamella openings could be mounted on these sides to increase the ventilation (and hence the air change process), but since this solution hasn't reached an ultimate design it hasn't been taken into account in simulations as well as calculations.
Another still open-question is the material which the weather shield will be made of. Taking inspiration from several projects around the world (Dome of Visions - Copenhagen), the main material could be either glass-based or polycarbonate-based. Both of them have advantages and constraints and at this step of the design process a clear choice hasn't been taken.

4.2.1.1 Simulation of the weather shield in TRNSYS

During the simulations of the weather shield in TRNSYS, a simple glass has been considered (for the sake of ease), taken from the predefined materials library of the software. Particular attention was given to the g-value of the material: the original value of 0.86 (assumed as valid for every glazed surface) was lowered to 0.35 for the glazed 26° tilted façade facing South because of the presence of semi-transparent photovoltaic cells (see Figure 8 for reference). The Daylight group of Team DTU had suggested an average value of 0.3 but it was increased in order to

take into account, even if in a very approximate way, the effect of direct sun penetrating the weather shield through its openings.
Regarding the ventilation, being natural, it fully relies on wind features. The ventilation process has been modeled taking into account only the open facades on the East and West sides, regardless of the openings of the North and South facades. Wind has been considered as affecting the natural ventilation either when its direction is between South-West and North-West (azimuth angle between 30 and 150°, west opening considered as cross section) or between North-East and South-East (azimuth angle from 210 to 330°, east opening considered as cross section), using the wind direction and wind speed data from the Energy+ weather file (Internation Weather for Energy Calculations - IWEC) to implement into the model equations to calculate the rate of air change per hour (ACH). Figure 12 and Figure 13 show the wind distribution for each direction over the entire year.

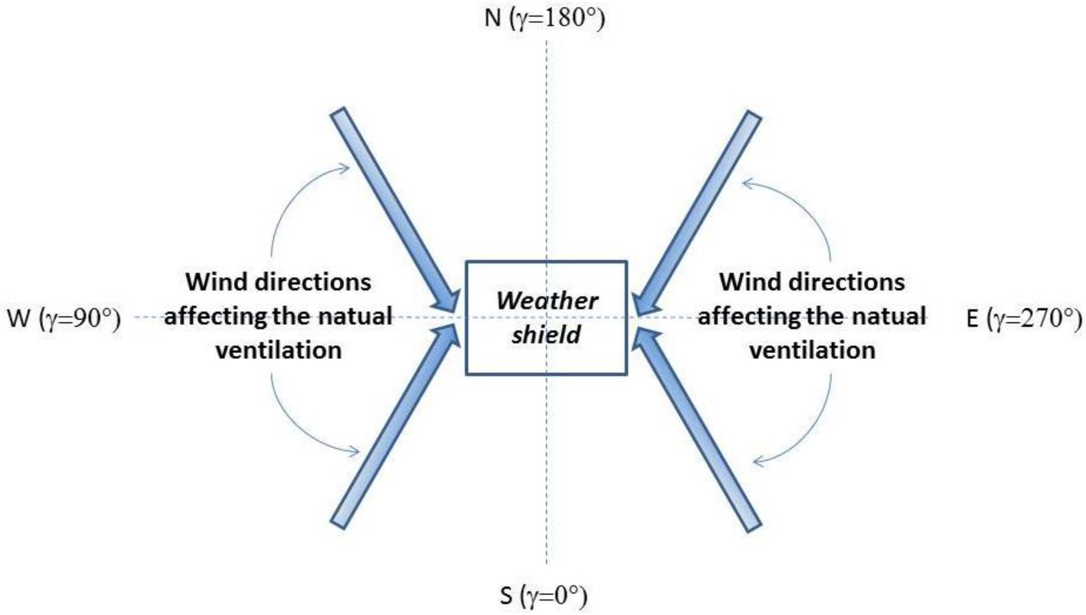


Figure 11: Wind directions affecting the natural ventilation process of the weather shield

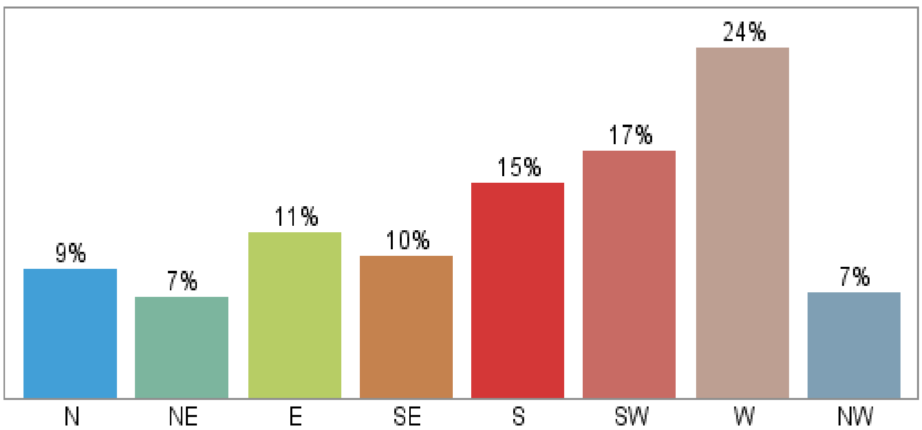


Figure 12: Wind directions over the entire year in Copenhagen. *http://weatherspark.com*

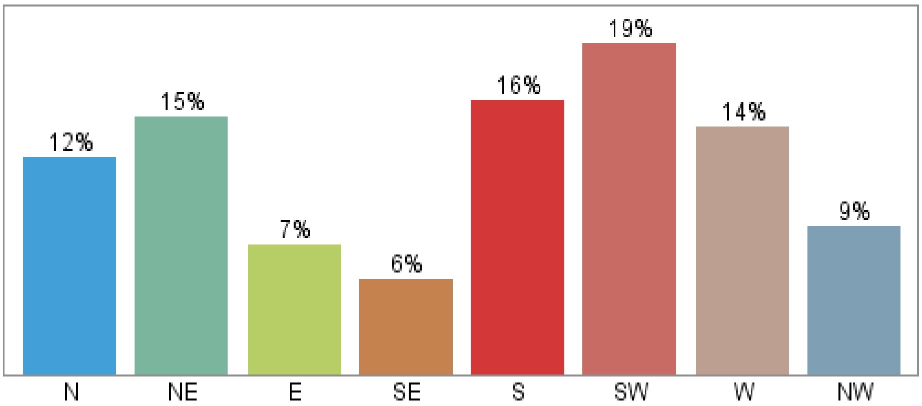


Figure 13: Wind directions over the entire year in Paris. http://weatherspark.com

The weather and therefore the house are oriented according to Figure 11. Simulations were carried out both for Copenhagen and Paris. The purpose is to estimate the temperature inside the weather shield, especially in summer conditions, in order to evaluate the possible overheating. The created TRNSYS model can be seen in the following Figure 14:

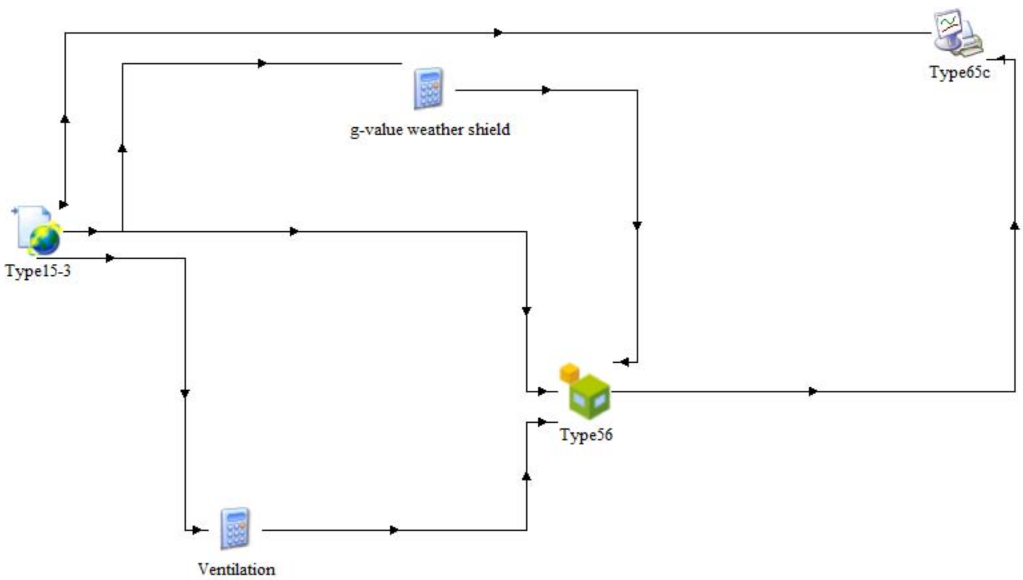


Figure 14: TRNSYS Simulation Studio representation of the model

Figure 14 contains:

- Type 15-3: Energy+ weather file
- Type 56: Model in TRNBuild of the weather shield
- Type 65c: Output component

The hourly temperature inside the weather shield all over the year was obtained from the previously described simulations. This evaluation enables the verification of the design temperature for both cities' climates and highlights that the outdoor temperature for cooling demand calculation (found in ASHRAE Handbook, 2009) has to be increased, due to the presence of the weather shield. According to these results, design temperatures for cooling load calculations have been changed.

Table 1: Design temperature for cooling demands for Copenhagen and Paris, with and without taking into account the weather shield

	$T_{design, no\ weather\ shield}$ (°C)	$T_{design, weather\ shield}$ (°C)
Copenhagen	30	32
Paris	30.9	35

Table 2: Temperatures for average cooling demands calculations for Copenhagen and Paris, with and without taking into account the weather shield

	$T_{average, no\ weather\ shield}$ (°C)	$T_{average, weather\ shield}$ (°C)
Copenhagen	18.1	19
Paris	20.7	22

Regarding the design temperatures for heating load calculations, values provided by ASHRAE Handbook (2009) for Paris and by the notes from (11221: Ventilation and Climatic systems, Load Calculations, 2006) for Copenhagen have been used. Since one of the weather shield's main purposes is to create a warmer environment during winter, for security benefits, outside design temperatures for heating haven't been changed even though this could lead to an overestimation of the heating load. Regarding the temperatures used for cooling calculations, values have been increased according to assumptions based on the TRNSYS simulations of the natural ventilation process inside the weather shield for security benefits.

4.2.2 Hypotheses and input values

Load calculations have been based on two inhabitants and Copenhagen and Paris weather conditions. Due to the lack of information regarding the structure of the external walls of the inner building (thermal envelope), a total coefficient of transmission (U-value) of 0.1 $W/m^2{\cdot}K$ has been used for the calculations, according to the experience from the previous house (Kazanci & Skrupskelis, 2012) The following table presents area and U-value of every surface, floor and ceiling area, conditioned volume.

Table 3: Overview of walls and surfaces

Orientation	Area (m^2)	U-value ($W/m^2 \cdot K$)
South (vertical)	16.0	0.1
South (26° slope)	27.8	0.1
West (facing outside)	41.2	0.1
North (vertical)	20.2	0.1
North (62° slope)	13.5	0.1
East (facing outside)	7.6	0.1
East (facing inside)	28.0	0.1
Doors	13.2	0.7
Roof	18.0	0.1

Floor area	42.8 m^2
Room height	3.0 m

Walls and surfaces are split according to their orientations. In particular, East facade has been divided depending on the environment they face, being it outdoor environment or the one under the weather shield. The purpose of this approach is to provide ease for the calculations. For the same reason, surfaces have been split according to their tilt (tilt angle is defined as the angle between the plan of the surface and the horizontal).

Doors are considered as unique wall surfaces, characterized by a U-value of *0.7 $W/m^2 \cdot K$*. Overall door surface presented in the table above is the sum of every single door surface.

Transmission losses are estimated using the formula:

$$Transmission\ \ losses = \sum_k U_k \cdot A_k \cdot (T_{outdoor} - T_{indoor})$$

Equation 3

Where U_k [$W/m^2 \cdot K$] is the heat transmission coefficient for the k^{th} surface, A_k [m^2] is the area of the k^{th} surface, $T_{outdoor}$ and T_{indoor} [K] are the outside and indoor temperatures, respectively.

The 26° tilted, South facing surface is not taken into account, since it's provided with PV/Ts and it could be very difficult to estimate the exact flow of the heat. Due to these calculation issues, it has not been considered in the transmission losses for load calculations purposes. However, since the TRNSYS component for the implementation of the PV/Ts provides their back-surface temperature, in TRNSYS simulations the surface is taken into account.

Windows have been assumed to be triple glazing (4-15-4-15-4), according to the experience from the previous competition. The following table describes the main features of windows: orientation, area, U-value, solar transmission coefficient, factor of shade, etc.

Table 4: Windows data overview

	South (90° tilted)	South (26° tilted)	North (90° tilted)	North (62° tilted)	East (90° tilted)
Number of windows	1	1	1	1	1
Window area(m^2)	2.5	1.4	1.8	0.8	0.9
Factor for Window Area	1.0	1.0	1.0	1.0	1.0
U-value ($W/m^2 \cdot K$)	0.7	0.7	0.7	0.7	0.7
Transmission coefficient	0.3	0.3	0.3	0.3	0.3
f_{shade} device (ext. blinds)	1.0	1.0	1.0	1.0	1.0
f_{shade}	0.4	0.4	0.4	0.4	0.4
f_{shadow} (no obsticuls)	1	1	1	1	1

Windows affect the thermal balance due to two processes: transmission losses or gains towards or from the outside environment and solar gains. Thus, the formula used for the evaluation of the windows' contribution is the following (11221: Ventilation and Climatic systems, Load Calculations, 2006):

$$\Phi_{solar} = A_{glass} \cdot f_{shade} \cdot (f_{shadow} \cdot I_{t,direct} + I_{t,diffuse})$$

Equation 4

Φ_{solar} is the heat gained due to the glazing surface [W], A_{glass} is the glazing surface [m^2], f_{shade} is the shading coefficient, f_{shadow} is the shadow coefficient $I_{t,direct}$ and $I_{t,diffuse}$[W/m^2] are the direct and diffuse solar radiations transferred through the glazed area of a double pane reference window. f_{shade} is calculated as:

$$f_{shade} = \frac{transmission\ coefficient\ of\ current\ window}{transmission\ coefficient\ of\ reference\ window} =$$

$$= \frac{f_{shade,external\_device} \cdot transmission\ coefficient}{transmission\ coefficient\ of\ reference\ window}$$

Equation 5

The solar transmission coefficient of the double pane reference window is 0.76 (11221: Ventilation and Climatic systems, Load Calculations, 2006).

Assuming that the house is built with respect to the necessary standards and its air-tightness is assured by high quality materials for walls, windows and structures, the infiltration rate has been set as *0.1 ACH*.

Indoor human activities have been based on presence of 2 inhabitants. Following are the technical data used, according to the hypothesis of moderate active office work.

Table 5: Human occupancy overview (11221: Ventilation and Climatic systems, Load Calculations, 2006)

Occupants	2	persons
Activity (sedentary activity)	1.2	met
Total heat released	132	W
Sensible heat at 24°C	73	W
Latent heat at 24°C	59	W

Heat gains from lighting equipment have been estimated as 3 W/m^2 (ceiling surface, equal to floor area).

At this design step there are no defined data about the internal equipment, therefore the values used in the SDE2012 *Fold* house have been considered in the thermal balance. Internal gains of 0.46 kW and 0.30 kW have been used for "maximum" and "average" conditions, respectively. Appliances are supposed to be activated continuously.
Total heat gains contribution derives from the appliances listed in

Table 6 and Table 7. As stated in (11221: Ventilation and Climatic systems, Load Calculations, 2006), 30% of the power has to be considered as turned into heat gain. The appliances included in the calculation are very energy-efficient and particular attention has been given to their low energy consumption.

Table 6: Appliances and respective power – "Maximum" load calculations

PC	35	W
Refrigerator Bosch KGV36VL30G	26.25	W
Freezer	0	W
Clothes washer Bosch WIS24140GB	100	W
Clothes dryer Bosch WTW84360GB	160	W
Dishwasher Bosch SRS55C02GB	400	W
Home Electronics (TV, dvd) 41+11	26	W
Oven, Miele H4412B	790	W

Table 7: Appliances and respective power – "Average" load calculations

PC	35	W
Refrigerator Bosch KGV36VL30G	26.25	W
Freezer	0	W
Clothes washer Bosch WIS24140GB	0	W
Clothes dryer Bosch WTW84360GB	0	W
Dishwasher Bosch SRS55C02GB	400	W
Home Electronics (TV, dvd) 41+11	26	W
Oven, Miele H4412B	790	W

Ground losses are calculated according to UNI EN 12831 (2006). Starting from a total coefficient of heat exchange towards the ground of 0.3 W/m^2·K and according to this standard, the formula used is the following:

$$\Phi_{ground} = H_{T,ig} \cdot (T_{in} - T_{avg,ground}) = f_{g,1} \cdot f_{g,2} \cdot (\sum_k U_{equival,k} \cdot A_k) \cdot G_w \cdot (T_{in} - T_{avg,ground})$$

f_{g1} takes into account the influence of the yearly average temperature; f_{g2} is the temperature reduction factor; $U_{equival,k}$(W/m^2·K) is the equivalent thermal transmittance towards the ground, calculated from the actual thermal transmittance; A_k is the area of the k^{th} floor surface; G_w takes into account the presence of the water in the soil; $T_{avg,ground}$ is the yearly average ground temperature, assumed to be equal to the yearly average air temperature; T_{in} is the indoor air temperature, set as 22°C or 24.5°C; Φ_{ground} (W) is the heat loss to the ground.

4.2.2.1 *Winter calculations*

Heating demand calculations are based of two reference conditions, a design one and an average one, both for Copenhagen's and Paris' climates. The design values are the following:

Table 8: Winter design temperatures for Copenhagen and Paris

	Copenhagen	Paris
Winter design temperature – maximum (°C)	-12 [1]	-5.9 [2]
Winter design temperature – average (°C)	1.8 [3]	0.9 [4]

References:

- *[1]*: (11221: Ventilation and Climatic systems, Load Calculations, 2006)
- *[2]*: (ASHRAE Handbook , 2009) - Design Temperature for heating load in Paris
  [3]: Monthly average temperature for February (Photovoltaic Geographical Information System - PVGIS); as stated in (ASHRAE Handbook , 2009), February is the coldest month in Copenhagen
- *[4]*: Monthly average temperature for January (Photovoltaic Geographical Information System - PVGIS); as stated in (ASHRAE Handbook , 2009), January is the coldest month in Paris

No changes in the design temperatures are applied because of the weather shield, because, as previously stated, it's meant to create a warmer environment outside the inner dwelling and, thus, considering the outdoor temperatures as design temperatures is reasonable for being in the conservative side in the calculations.

The indoor temperature is set as 22°C, since the purpose is the satisfaction of the Category I condition for the indoor environmental quality of buildings (EN 15251, Annex A).

Solar gains are not considered in winter-time, for the "maximum" case and even for the "average" case, since the purpose is to evaluate the heating load in absence of any other thermal gain. Windows, thus, affect the balance only due to transmission losses.

Ventilation contribution to the heat load takes into account the presence of a heat recovery system, which effectiveness is 0.8. The requested power has to cover the temperature gap between the heat exchanger outlet temperature and the indoor temperature.

The final balance is calculated adding up all the thermal losses, ignoring the heat gains. The result (Φ_{losses} [W]) is the addition of the following contributions: ventilation ($\Phi_{ventilation,heat\ recovery}$[W]), windows (no sun radiation considered – $\Phi_{transmission,windows}$[W]), thermal exchange through the walls($\Phi_{transmission,walls}$[W]), thermal exchange due to the ground ($\Phi_{losses,ground}$[W]), infiltration ($\Phi_{losses,infiltration}$[W]), as the following formula describes.

$$\Phi_{losses} = \Phi_{ventilation,heat\ recovery} + \Phi_{transmission,windows} + \Phi_{transmission,walls} + \Phi_{losses,ground} + \Phi_{losses,infiltration}$$

Equation 6

4.2.2.2 Summer calculations

As in the heating case, cooling demand calculations are based on two reference conditions, a design one and an average one, both for Copenhagen's and Paris' climates. Since the presence of the weather shield adds an overheating effect, its influence in temperatures is not negligible anymore. Thus, design temperatures have been increased, according to the TRNSYS simulations of the weather shield. The design values are the following:

Table 9: Summer design temperatures for Copenhagen and Paris – No Weather shield

	Copenhagen	Paris
Summer design temperature – maximum (°C)	30 [5]	30.9 [6]
Summer design temperature – average (°C)	18.1 [7]	20.7 [8]

References:

- *[5]:* Hypothesis
- *[6]:* Design temperature for cooling demand in Paris, (ASHRAE Handbook , 2009)
- *[7]:* Monthly average temperature for July in Copenhagen (Photovoltaic Geographical Information System - PVGIS); as stated in (ASHRAE Handbook , 2009) July is the hottest month in Copenhagen
- *[8]:* Monthly average temperature for July in Paris (Photovoltaic Geographical Information System - PVGIS); as stated in (ASHRAE Handbook , 2009), July is the hottest month in Paris

Table 10: Summer design temperature for Copenhagen and Paris – Weather shield overheating taken into account

	Copenhagen	Paris
Summer design temperature – maximum (°C)	32[9]	35[9]
Summer design temperature – average (°C)	19[9]	22[9]

References:

- [9] : TRNSYS Weather Shield simulations

As previously reported, based on the TRNSYS simulations for the weather shield, design temperatures for outdoor conditions have been increased, in order to take into account the overheating deriving from the presence of weather shield.

The indoor temperature is set as 24.5°C, since the purpose is to satisfy Category I condition for the indoor environmental quality of buildings (EN 15251, Annex A).

Solar radiation data are taken from the PVGIS weather databases (*Photovoltaic Geographical Information System – PVGIS – online calculator for monthly global irradiation data, http://re.jrc.ec.europa.eu/pvgis*), calculating the daily average value for both the beam radiation and the diffuse radiation. The considered month is July, as reported in (ASHRAE Handbook , 2009). As a result, windows affect the final balance due to the transmission losses and the solar gains through the glazed surface. In order to consider the effect of the weather shield, with particular regard for its g-value, data on solar radiation, both beam and diffuse, are multiplied by the g-value of the weather shield, being 0.85 for the 90° tilted surface facing south and for the surfaces facing north, and 0.35 for the 26° tilted surface facing south (as stated, this surface is partly shadowed because of the presence of the semi-transparent PV cells and, thus, it´s the only one being affected by a lower g-value). g-values are taken from TRNSYS library for glazing components or indicated from team DTU Daylight group.

Regarding the transmission losses through the walls in the case the weather shield is taken into account, two different outdoor temperatures are considered, the actual outdoor air design temperature and the air temperature under the solar envelope, calculated as output of the TRNSYS simulations, respectively if the considered surface faces the actual outdoor environment or the environment under the weather shield.

The final balance is calculated as the sum of every loss or gain. Thus, the result is the addition of the following contributions: ventilation, windows (sun radiation considered), equipment, lighting, occupants, thermal exchange through the walls, thermal exchange due to the ground, infiltration.

4.2.3 Results

According to the previously stated information, assumptions and procedures, heating and cooling demand calculation has been performed. The ground floor presents a flexible room, meant to be used by more than one dwelling. It´s reasonable to consider for this space, since it´s not used as often as the rest of the house, different temperature set-points, admitting temperatures lower than 22°C in winter as well as higher than 24.5°C in summer. The calculations presented have been carried out considering the shared space as part of the house, hence with equal temperature set-points, because during the competition it´s highly likely that this space will be conditioned as the rest of the building. Following are the results:

Table 11: Heating and cooling demands for Copenhagen and Paris – No weather shield considered

	Heating/Cooling demand	
Winter maximum CPH	33.1	W/m^2
Winter maximum PARIS	27.8	W/m^2
Winter average CPH	21.1	W/m^2
Winter average PARIS	21.9	W/m^2
Summer maximum CPH	30.2	W/m^2
Summer maximum PARIS	32.6	W/m^2
Summer average CPH	9.5	W/m^2
Summer average PARIS	12.7	W/m^2

Table 12: Heating and cooling demands for Copenhagen and Paris – Weather shield considered

	Heating/Cooling demand	
Winter maximum CPH	33.1	W/m^2
Winter maximum PARIS	27.8	W/m^2
Winter average CPH	21.1	W/m^2
Winter average PARIS	21.9	W/m^2
Summer maximum CPH	27.9	W/m^2
Summer maximum PARIS	31.6	W/m^2
Summer average CPH	8.2	W/m^2
Summer average PARIS	11.7	W/m^2

*Heating demands are exactly the same in both cases because the effect of solar radiation is neglected and, as previously said, winter design temperatures haven´t been changed due to the presence of the solar envelope.
Particular attention is given to the heating demand in “maximum” conditions for Copenhagen and to the cooling demand in “maximum” conditions for Paris. As expected, due to the local climate, their values are higher than the other city´s corresponding values. Thus, they are chosen, respectively, as heating and cooling output design values for all the HVAC system calculations and dimensioning.
It´s evident from the previous tables that the weather shield has a shadowing effect, decreasing the total cooling demands for both climates due to lowered solar radiation/solar gains, despite the higher outdoor temperatures deriving from its presence. It´s thus verified the absence of any increase in power demand for cooling because of the weather shield. The reduction effect is higher for Copenhagen´s summer conditions, probably because of the higher solar azimuth angles reached in Denmark. In fact, the solar path during the day includes longer time over the semi-transparent 26° tilted surface in Copenhagen than in Paris.

5. HVAC SYSTEM

In order to achieve the requested performance and required levels of indoor environmental quality, a crucial component of the house is the HVAC system.

5.1 Component configuration

HVAC system's components are:

- *PV/T panels*: above the roof of Embrace, PV/Ts are installed in order to cover as much as possible of both the electrical and thermal demand of the building (referred to the heating season and cooling season). They will provide both electricity and hot water during daytime. In summer nights the same panels can be used for cooling, due to night-time radiative heat exchange towards the sky.
- *Domestic Hot Water (DHW) tank*: 300 liters capacity, in order to satisfy the required waiting-time for hot water during the competition. DHW tank is not directly connected to the solar panels, but, since the loop for domestic hot water needs to be separated and open, rather contains an intermediate spiral heat exchanger. Since PV/Ts thermal production is not enough to cover the demand all year long, an additional electric heating element is necessary in case the thermal production from the PV/Ts is insufficient or the water temperature is not high enough to satisfy the necessary response quickness requirements established by SDE Organization. Required electrical energy could be provided by either PV/Ts themselves or the grid.
- *Water Storage Tank*: higher capacity (likely 800 liters) in order to store higher amount of water and act as a buffer component for space heating and cooling purposes. Since DHW tank is requested to be filled by higher temperature water, water storage tank is installed after the DHW tank, along the hydronic circuit. It´s the only source of water to feed the embedded radiant system.
- *Air-to-Water Heat Pump (HP)*: this device is installed to support the system, both in heating and cooling cases. The chosen model is an air-to-water device, meaning that the heat is transferred between the hydronic circuit and outside air. Heat Pump is always used for heating in winter and the operation principle is the following: heat is extracted from outside air (thus, in outside conditions) on the evaporator´s side of the device and released to the heat pump´s refrigerant; the fluid is then compressed to the condensation pressure and, through the condenser, it´s able to release heat to an external fluid (the heating medium of the hydronic system), heating it up.

- *Radiant floor*: consisting of a dry-system, a radiant floor is installed all over the floor area of the building, covering both the ground floor and the first floor.
- Pipes, pumps and manifolds
- *Phase Change Materials (PCM)*: installed inside the building's structure or inside the hydraulic system itself, PCMs could smooth the thermal demands. They are not considered in this report. Their use could be implemented in further investigations.

5.2 Heating mode

5.2.1 Operation without heat pump

The system is required to cover two thermal demands of hot water: domestic hot water and hot water for space heating. It's possible to use a single hydraulic loop for both, being 55°C the requested temperature for the DHW tank and within 35÷40°C the requested temperature for space heating.

PV/Ts outlet water stream (see chapter 2 for detailed explanation of the solar radiation collection process) is firstly driven to the DHW tank and afterwards through the water storage tank. After transferring heat to both of the tanks, water is colder and ready to be circulated again in the PVTs water pipes. DHW is taken from the top part of DHW tank (due to the stratification process, this part has the highest temperature) and hot water for space heating is extracted from the upper part of the SHW tank as well; after circulating through the radiant floor loop, water is driven back to the water storage tank, flowing towards the lower part of it.

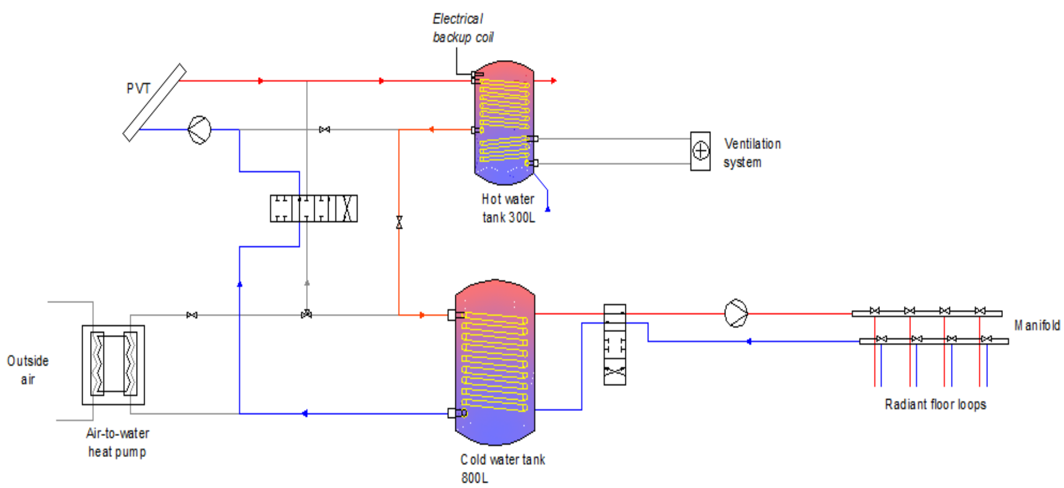


Figure 15: Heating season operation, without heat pump. ***Team DTU for SDE 14, Comfort Conditions group***

5.2.2 Operation with heat pump

In case the thermal production of the PVTs is absent or insufficient (due to the absence of sun, maintenance, etc.), the heat pump could act as heat source for the system. Extracting heat from outside air and releasing it to the hydraulic circuit, the heat pump is able to feed both the DHW and the SHW tanks in series (first the DHW tank and then the water storage tank), in the same way described in the previous sub-chapter. Return flows (cold water) are collected, merged and driven through the heat pump's condenser for the next loop to be started.

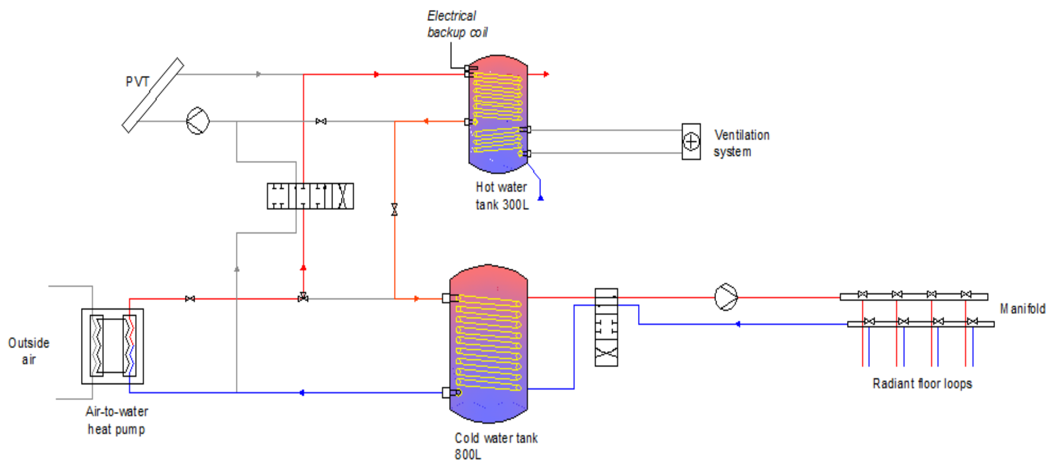


Figure 16: Heating season operation, with heat pump. ***Team DTU for SDE 14, Comfort Conditions group***

5.3 Cooling mode

5.3.1 Daytime operation

During the cooling season, tasks regarding the satisfaction of DHW and water storage tank requirements are split. PVTs are requested to provide the necessary amount of hot water to suit DHW tank requirements, heat pump is demanded to extract heat from the hydraulic circuit (the amount of heat that was extracted previously from the conditioned room due to the embedded system). In case the system is equipped with PCM, during daytime PCM layer absorbs heat from the conditioned room, storing it before being discharged during night-time.

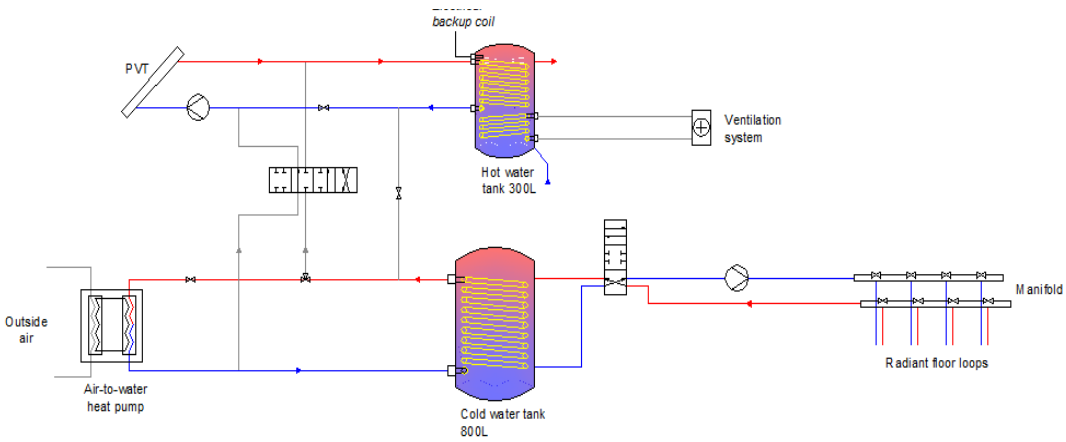


Figure 17: Cooling season operation, day-time. ***Team DTU for SDE 14, Comfort Conditions group***

5.3.2 Night-time operation

During night-time the main tasks to accomplish are the continuous cooling of the house and the discharge of the water storage tank. Sky is the heat sink to release the heat extracted. In fact, thanks to the night-time radiative effect, PVTs are able to cool down the water coming from the hydraulic loop. Cold water produced is driven towards the water tank, in order to decrease its temperature and face next daytime cooling needs. In the meanwhile, cold water is extracted from the bottom part of this tank and pumped to the embedded system, in order to cool down the space and the structure and, in case the system is equipped with PCM, to the PCM layer, in order to extract the heat they absorbed during the previous daytime and thus discharging them.

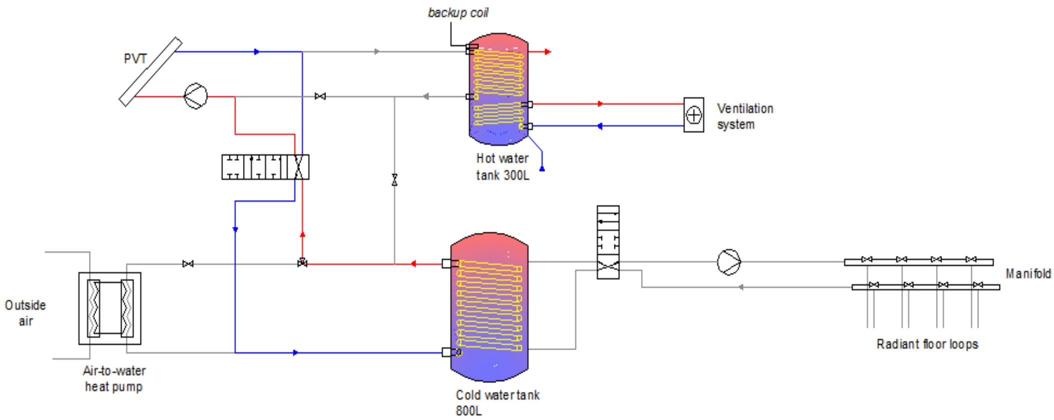


Figure 18: Cooling season operation, night-time. ***Team DTU for SDE 14, Comfort Conditions group***

6. EMBEDDED PIPE SYSTEM

Moving on towards the final purpose, i.e. the analysis of the interaction and the coupling of PVTs with the radiant system provided with thermal storage solution, the next step is the design of the radiant floor for *Embrace*.

6.1 Radiant system description and overview

The cases involving the weather shield have been taken into account as heating and cooling demands to satisfy. Using the previously calculated heating and cooling demands as input values, the radiant system has been dimensioned. As reported in the previous tables, the design heating need is *33.1* W/m^2 (referred to Copenhagen "maximum" winter conditions) and the cooling need is *31.6* W/m^2 (referred to Paris "maximum" summer conditions). Since at the present stage of the design process no detailed information about the actual structure of the floor and its layers are available, a structure similar to the one used in *Fold* (team DTU's house for SDE 2012) has been taken into account. Figure 19 shows the structure of the ground floor; image is taken from MIRAGE simulation software (see cub-chapter 6.2.4.1 for detailed information). Following is a brief description of the floor's structure:

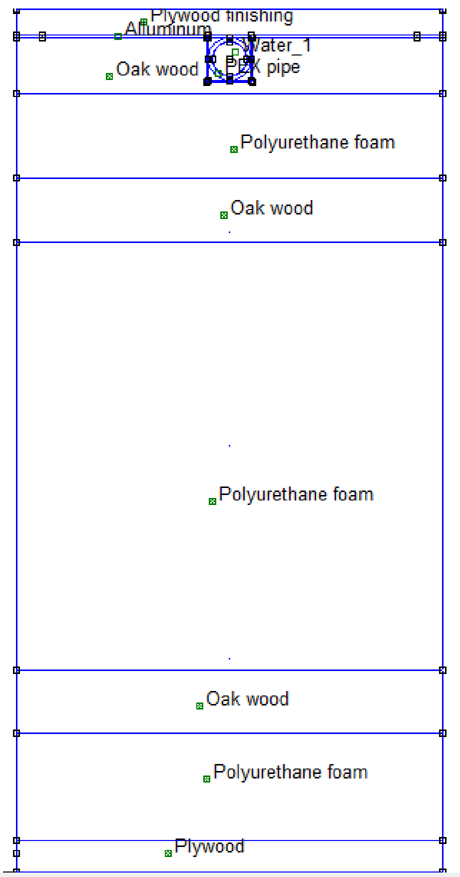


Figure 19: Floor structure overview. *Source: MIRAGE*

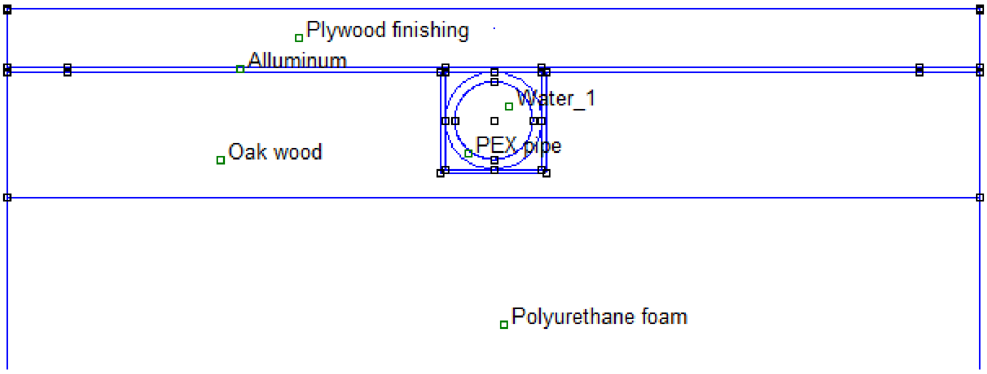


Figure 20: Floor structure, detailed view of the layers underneath the floor surface. ***Source: MIRAGE***

Table 13: Floor structure – Thickness and thermal conductivity of each layer

LAYER	NAME	s (m)	λ (W/m·K)
0	Plywood	0.015	0.125
1	Polyurethane foam	0.05	0.03
2	Oak wood	0.03	0.16
3	Polyurethane foam	0.2	0.03
4	Oak wood	0.03	0.16
5	Polyurethane foam	0.04	0.03
6	Oak wood	0.03	0.16
7	Pipe	0.02	0.35
8	Aluminum	0.001	226
9	Plywood finishing	0.012	0.15

Referring to Table 13, layers are numbered from the lowest (0 - in contact with the ground) to the highest (9 - in contact with the conditioned space). σ [m] is the thickness of the layer and λ is its thermal conductivity [W/m·K].

6.2 Radiant system dimensioning

Among the international standards' overview, two of them particularly deal with the problem considered, the (EN1264 - Water based surface embedded heating and cooling system, 2008), and the (EN15377 - Heating system in buildings - Design of embedded water based surface heating and cooling system, 2008). Both of them belong to the European regulation and have European validity.
The purpose of the performed parallel analysis of two different and alternative calculation methods is to examine the differences and the similarities, likely providing suggestions and guidelines to the final-user.

The first step of the dimensioning procedure is to fix several features of the system; the most important parameters in this group are pipe material, pipe spacing and pipe diameter. Regarding the choice of the pipes to install, *Uponor* catalogue (Uponox PEX catalogue 2013) has been taken as reference. The chosen pipe is *Uponor EvalPEX Q&E 20 x 2* mm. The main characteristics are reported below:

Table 14: Main features of choses pipe type – Uponor EvalPEX Q&E 20 x 2. *Source: Uponor PEX catalogue 2013*

Outer diameter	0.02	m
Inner diameter	0.016	m
Pipe wall thickness	0.002	m
λ (Pipe wall thermal conductivity)	0.35	W/m·K

Regarding the pipe spacing, a distance between the pipes of 0.20 m has been considered.

6.2.1 EN 1264 - Water based surface embedded heating and cooling system

EN 1264 (2008) describes a calculation method involving the power function which is the product of factors a_k, having m_k as exponents. The procedure applies to four standard types of embedded structures, *Systems with pipes installed inside the screed (type A and type C), Systems with piped installed below the screed or timber floor (type B), Systems with surface elements (type D)*. Due to construction constraints, type B system has been selected, but it has been modified in order to narrow the difference between the actual system and the fictitious standard system described in EN 1264. The modification consists in setting layer 2 - weight bearing layer, screed (see Figure 21 below)–as the interior cladding; therefore layer 1 thickness is set as null.

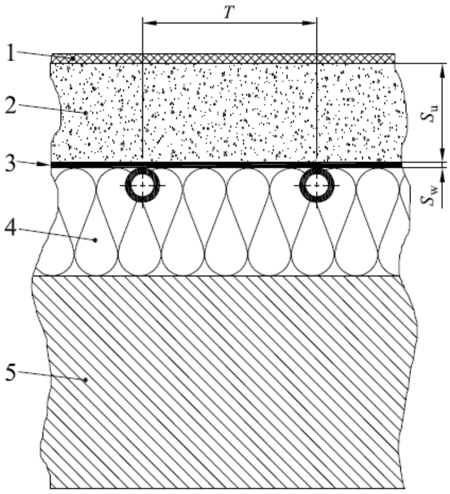


Figure 21: Type B floor structure – Systems with pipes installed below the screed. ***Source: EN 1264-2 (2008)***

The basic formula provided for type B structures is the following:

$$q = B \cdot \prod_k a_k^{m_k} \cdot \Delta\vartheta_H = B \cdot a_B \cdot a_T^{m_T} \cdot a_U \cdot a_{WL} \cdot a_K \cdot \Delta\vartheta_H = k_H \cdot \Delta\vartheta_H$$

Equation 7

B [W/m^2·K] is a system-dependent coefficient; $\prod a_k^{mK}$ is the power product, depending on several features of the system (pipe diameter, pipe spacing, floor covering for instance); $\Delta\theta_H$ is the logarithmic temperature difference between the heating medium and the room temperature calculated as follows:

$$\Delta\vartheta_H = \frac{\vartheta_V - \vartheta_R}{\ln \dfrac{\vartheta_V - \vartheta_i}{\vartheta_R - \vartheta_i}}$$

Equation 8

θ_V [K] is the system's supply temperature, θ_R [K] is the system's return temperature (the difference $\theta_V - \theta_R$ is usually called σ [K]) and θ_i [K] is the indoor temperature.

Following the guidelines and the calculation procedure described in EN 1264-2, k_H coefficient has been calculated. Once k_H is evaluated, it's possible to obtain the design supply water temperature θ_V. Being the design heat output equal to *33.1 W/m^2* and considering a temperature drop σ (difference between supply and return temperature) equal to 5°C (EN1264, 2008), the design value of θ_V is *32.3°C*.

A necessary check that must be done afterwards is about surface temperatures limitations. According to ISO 7730, floor surface temperature should be within the range 19÷29°C. MIRAGE simulation (described in subchapter 6.2.4) is the base for this check.

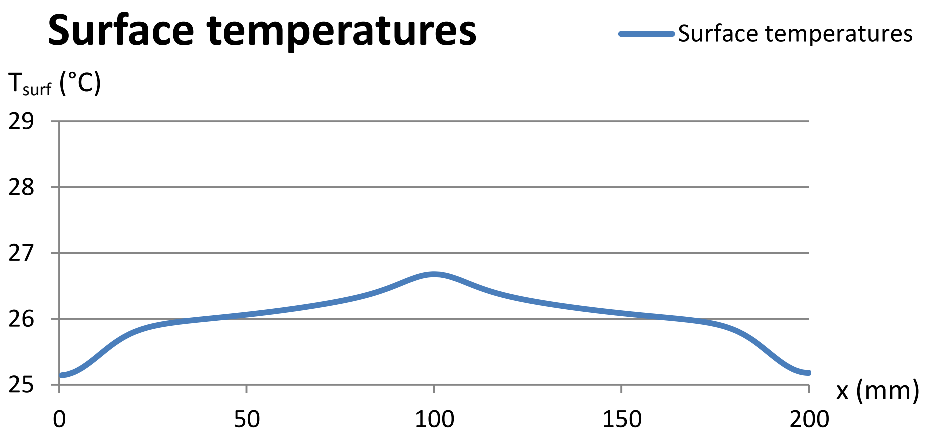


Figure 22: Surface temperature distribution along a pipe spacing T – EN1264. Source: MIRAGE

$T_{surface}$ [°C]: surface temperature
x [mm]: horizontal distance, pipe center corresponds to x=100 mm

Since the maximum temperature, reached above the water pipe, is *26.7°C* the constraints regarding the maximum surface temperature are respected.

Hereinafter the calculations the calculations regarding the design flow-rate are presented. According to EN 1264-3 (2008), the design flow-rate for heating season has to be calculated as follows.

$$m_H = \frac{A_F \cdot q_H}{\sigma \cdot c_W} \cdot \left(1 + \frac{R_o}{R_u} + \frac{\vartheta_i - \vartheta_u}{q \cdot R_u}\right)$$

Equation 9

Being A_F the floor area [m^2]; q the specific heat output the system is requested to provide [W/m^2]; σ the temperature drop between supply temperature and return temperature [K]; c_W the specific heat capacity of water [J/kg·K]; R_o the upwards partial heat transmission resistance of the floor structure [m^2·K/W]; R_u the downwards partial heat transmission resistance of the floor structure [m^2·K/W]; θ_i the indoor temperature [K]; θ_u the indoor temperature of a room under the floor heated room [K].
R_o and R_u detailed calculation method can be found in Section 4.1.3.3 of EN 1264-3.
Since the floor structure is in contact with the floor, it is hypothesised that θ_u is the ground yearly average temperature for Copenhagen (9.3°C), assuming it equal to the yearly average outdoor temperature. The calculation of the flow-rate has been split among ground floor and first floor, since both of them are intended to be used as active structures. Total design flow-rate in heating season is therefore 225 kg/h.

6.2.1.1 Calculation for cooling demand

The reference standard text for cooling demand calculations is EN 1264-5 (2008). The basic formula to obtain coefficient k_H in cooling mode is the following:

$$k_H = k_H(\Delta R_\alpha, R_{\lambda,B}) = \frac{k_{H,floor}}{1 + \frac{\Delta R_\alpha + R_{\lambda,B}}{R^*_{\lambda,B}} \cdot (\frac{k_{H,floor}}{k^*_{H,floor}} - 1)}$$

Equation 10

- $k_{H,floor}$[W/m^2·K]: is the gradient of the characteristic curve of the embedded system, obtained from EN 1264-2 with a thermal resistance of the covering $R_{\lambda,B}=0$.
- $k^*_{H,floor}$ [W/m^2·K]: is the gradient of the characteristic curve of the embedded system with a higher thermal resistance of the covering $R^*_{\lambda,B} > R_{\lambda,B}$. Generally, $R^*_{\lambda,B} = 0,15\ m^2 \cdot K/W$ applies.

ΔR_α[m^2·K/W]: is the additional resistance to be calculated for the surface in question. $\alpha_{floor_cooling}$is, according to EN 1264-5 (2008), *6.5 W/m^2·K*.

$$\Delta R_\alpha = \frac{1}{\alpha_{floor\_cooling}} - \frac{1}{10{,}8}$$

Equation 11

More detailed explanations regarding the calculation of the parameters mentioned above can be found in EN 1264-5, Section 4.

Since the structures which are requested to provide the cooling energy are the floors, according to Table A.1 of EN 1264-5 $\alpha_{cooling}$ to 6.5 W/m^2·K is involved. Temperature drop of 4°C has been considered. Results show that the supply temperature in cooling season is 13.3 °C.

The calculation of the design flow-rate is referred to Section 5.2.3 of EN 1264-2. The basic formula involves the same parameters presented in subchapter 6.2.1.1.

$$m_C = \frac{A_F \cdot q_C}{\sigma \cdot c_W} \cdot \left(1 + \frac{R_o}{R_u} + \frac{\vartheta_i - \vartheta_u}{q \cdot R_u}\right)$$

Equation 12

The calculation has been split among ground floor flow-rate and first floor flow-rate, as well as in the heating case. The temperature drop σ has been considered equal to 4°C. Design flow-rate has been obtained according to the implementation of this formula, resulting equal to 315 kg/h.

6.2.2 EN 15377 - Heating systems in buildings, Design of embedded water based surface heating and cooling system

EN 15377 (2008) introduces another calculation method. Since it's based on thermal resistances, it's known as the *Resistance Network Method*. It has general validity and it's an alternative to the use of EN 1264 (2008) for type A,B,C,D. Moreover, it extends the calculation's applications to three more types of structure, *Embedded radiant systems in concrete slab (type E)*, Embedded radiant systems with capillary tubes in concrete surface (type F) and *Embedded radiant systems in wooden constructions, pipes in sub-floor or under sub-floor, conductive devices (type G)*.Since the actual structure of Embrace will most likely be a wood-based system, type G is chosen as reference. The calculation method is described in *EN 15377-1, Annex C – Pipes embedded in wooden construction*.

The procedure involves three thermal resistances:

- R_i: thermal resistance above the heat conducting layer, from the heat conducting layer to the conditioned room (chosen value for contact resistance $R_{con,i}$ is 0.10 m^2·K/W, because it´s assumed that the heat conducting plate is carefully shaped and bonded to the floor materials)
- R_e: thermal resistance beneath the heat conducting layer, from the heat conducting plate to a neighbour room or outside
- R_{HC}: thermal resistance from the heating medium to the heat conducting layer

The structure has to fulfill some conditions, represented as follows:

$$\lambda_{WL} \geq 10 \cdot \lambda_{surroundig \ \ materials}$$
$$s_{WL} \cdot \lambda_{WL} \geq 0.01$$
$$L_{WL} \leq T$$

Being λ_{WL}[W/m·K] the thermal conductivity of the heat conducting plate, s_{WL}[m] the thickness of the heat conducting plate, the product $s_{WL}\cdot\lambda_{WL}$ [W/K] the heat performance of the heat conducting device (EN 15377-1, Section C), L_{WL}[m] the width of the heat conducting plate, T [m] the pipe spacing. L_{WL} is chosen equal to 0.175 m.

Restrictions are fully satisfied in the examined case.

The reference formula to obtain the design supply water temperature is:

$$\vartheta_i - \vartheta_m = R_i \cdot q_i$$

Equation 13

θ_i [K] is the indoor temperature of the conditioned room; R_i [m^2·K/W] is the thermal resistance above the heat conducting plate; q_i [W/m^2] is the design heat output the system is required to provide; θ_m [K] is, according to EN 15377-1:

$$\vartheta_m = \vartheta_i + \Delta\vartheta_H = \vartheta_i + \frac{\vartheta_V - \vartheta_R}{\ln \frac{\vartheta_V - \vartheta_i}{\vartheta_R - \vartheta_i}}$$

Equation 14

It's therefore possible to obtain θ_V for a heat output of *33.1 W/m²*: thus, considering a temperature drop σ of 5 K, the design supply water temperature, according to EN 15377, is *33.8°C*. Using this supply value, surface temperatures are checked. According to ISO 7730, the admissible range is 19÷29°C.

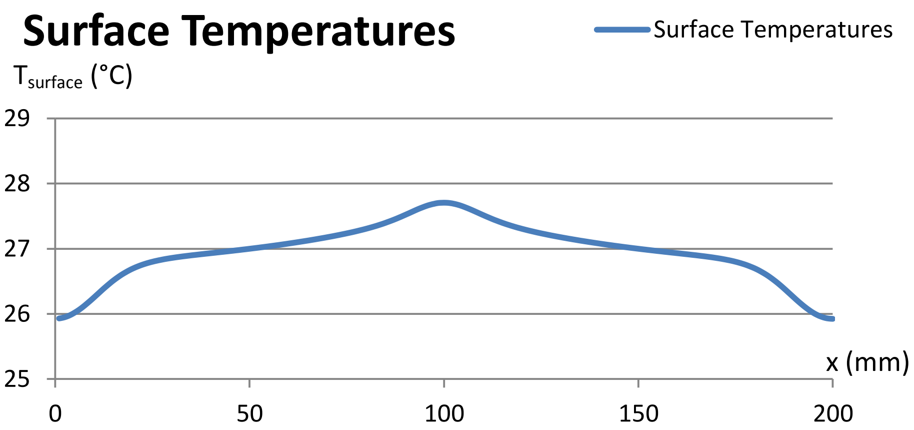


Figure 23: Surface temperature distribution along a pipe spacing T – EN15337. Source: MIRAGE

$T_{surface}$ [°C]: surface temperature
x [mm]: horizontal distance, pipe center corresponds to x=100 mm

Maximum temperature is 27.8°C and therefore it´s within the required range.

Total design flow-rate is calculated according to the formula presented in subchapter 6.2.1 (Equation 12) for the heating case according to EN 1264-3. Therefore the result is identical, corresponding to 225 kg/h.

6.2.2.1 Calculation for cooling demand

The calculation procedure for heating dimensioning can apply for cooling calculations as well, but it's obviously necessary to consider a different design heat flux output and a different design indoor temperature.

Results highlight that the supply water temperature has to be equal to 13.2°C. Regarding mass flow-rate, the calculation method presented in subchapter 6.2.1.1 applies (Equation 12). The results are therefore identical.

6.2.2.2 Influence of thermal contact resistance $R_{con,i}$

Calculation method described in EN 15377 provides predefined values for the thermal contact resistance between the heat conducting plate and the floor materials ($R_{con,i}$); the value has to be chosen among two possibility, 0.15 or 0.10 m^2·K/W, depending on the quality of the thermal connection. It's obvious that the better this connection is, the lower the thermal contact resistance is. In particular $R_{con,i}$ = 0.10 m^2·K/W is suggested if "*the heat conducting plate is carefully shaped and bonded to floor materials*" (EN 15377 - Annex C, 2008). The chosen value for $R_{con,i}$ can significantly affect the final result in terms of supply temperature, therefore its effect has been investigated from a quantitative point of view.
Thermal contact resistance $R_{con,i}$ is involved in the calculation of resistance R_i. As previously described, calculations presented above are based on a value of 0.10 m^2·K/W for this resistance, according to the guidelines of the standard. In order to study the influence of $R_{con,i}$ on the final result, using fixed values of this contact resistance as inputs, design supply water temperature has been calculated according to EN 15377. Following are the results:

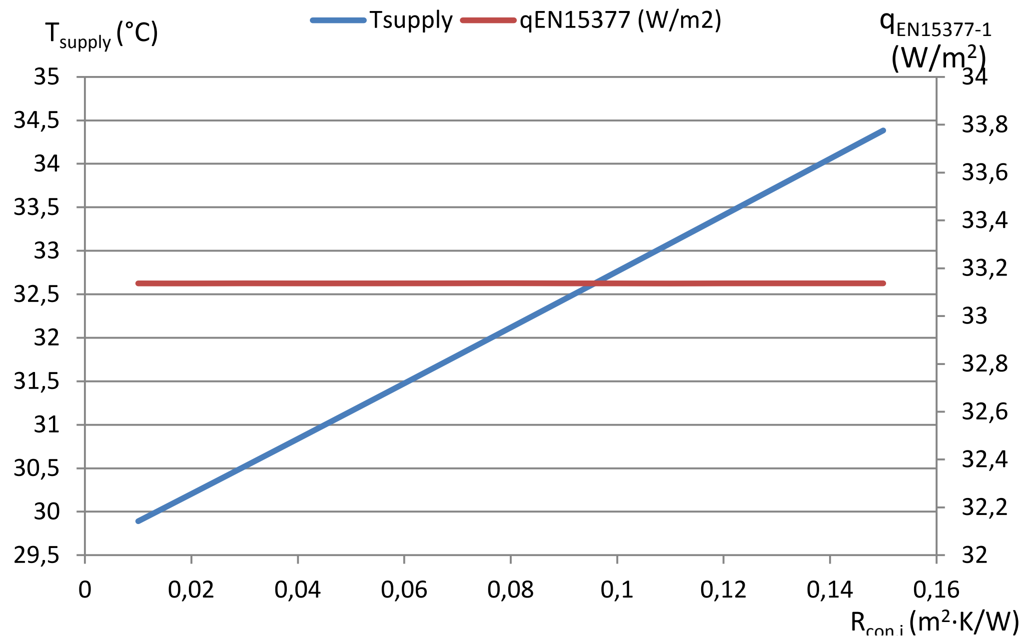


Figure 24: Influence of $R_{con,i}$ on the design supply temperature and heat output calculation

As the figure above shows, contact thermal resistance $R_{con,i}$ affects the output (design supply water temperature) in a considerable way. According to the EN 15377 standard, $R_{con,i}$ should be chosen among 0.10 – 0.15 m^2·K/W, depending on the quality of the thermal connection between the heat conducting device and floor finishing materials. This leads, in the examined case, to the supply temperatures of:

Table 15: Influence of $R_{con,i}$ on the design supply water temperature and heat output

$R_{con,i}$ ($m^2 \cdot K/W$)	$\theta_{V,des}$ (°C)	$q_{EN15377}$ (W/m^2)
0.1	32.8	33.1
0.15	34.4	33.1

According to the previous observations, it´s possible to conclude that the choice of contact thermal resistance value affects the final design supply water temperature in a remarkable way. This, obviously, could lead to remarkable differences in the final dimensioning of the system and to different energy consumptions.
Table 15 also underlines that the heat output remains the same with different values of $R_{con,i}$ (however it has to be kept in mind that the heating demand is not particularly high, therefore different applications with higher heating demands could lead to different conclusions).

6.2.3 EN 1264 - EN 15377 Comparison

Previously described calculations highlight that from the same required heat output, it's possible to obtain two different design supply water temperatures, depending on the standard procedure followed, EN 1264 or EN 15377. The difference is close to 1.5 K, and it's not negligible in a low energy building as *Embrace* is supposed to be. In fact, this kind of dwelling relies on the concept of Low Temperature Heating (LTH) and High Temperature Cooling (HTC) and it´s very important to keep the embedded system´s supply temperature as close as possible to the indoor temperature. Moreover, EN 15377 provides another suggested formula to calculate the design supply temperature (in a "somewhat simplified" way, as stated in EN 15377-1, 2008), which gives a totally different result, disagreeing with both EN 1264 and the previous formula of EN 15377.

In order to sum up the previous observations and highlight, from a numerical point of view, the difference between the two standards, a supplementary calculation has been performed, fixing as inputs a range of values for the water supply temperature (25÷50 °C) and the temperature drop σ (θ_V-θ_R = 5°C) and calculating the heat output using the three different ways:

- $q_{EN1264\text{-}2}$: Equation 7 described in sub-chapter 6.2 is used. K_H is calculated for the examined radiant system taken into account.
- $q_{EN15377\text{-}1,1}$: Equation 13 and Equation 14 described in sub-chapter 6.2.2 are used.

- $q_{EN15377\text{-}1\_simplified\_1}$: following formula is used, even if the calculation is "somewhat simplified", as stated in (EN 15377-1, 2008):

$$q_{EN15377-1\_simplified\_1} = k_H \cdot \Delta\vartheta_H$$
$$k_H = \frac{1}{R_{HC} + R_i}$$

Equation 15

- $q_{EN15377\text{-}1\_simplified\_2}$: a different formula is considered but the calculation is simplified as well. Following is the formula:

$$q = k_H \cdot \Delta\vartheta_H$$
$$k_H = \frac{q_{i,\max}}{\vartheta_H^{\max} - \vartheta_i}$$
$$\vartheta_H^{\max} = \vartheta_i + \frac{q_{i,\max}}{R_{HC} + R_i}$$

Equation 16

Clearly, $q_{EN15377\text{-}1\_simplified\_1}$ and $q_{EN15377\text{-}1\_simplified\_2}$ coincide.

Starting from a range of water supply temperatures within 25÷50°C and using data and features of the examined system, calculations are performed. Results are reported in Figure 27. In EN 15377 calculations, $R_{con,i}$ (whose value's choice can affect the result in a significant way as discussed in the next sub-chapters) is considered equal to 0.10 m^2·K/W.

Table 16: Comparison of heat ouput calculated using different standard´s methods. In $\theta_{V,EN15377}$ cases $R_{con,i}$ = 0.10 W/m^2·K

q_{des} (W/m^2)	$\theta_{V,des,EN1264-2}$ (°C)	$\theta_{V,des,EN15377-1}$ (°C)	$\theta_{V,des,EN15377-1\_simplified}$ (°C)
33.1	32.4	33.8	40.0

It is observed that the difference between the two standards is not negligible (supply temperature calculated according to (EN15377-1, 2008) is 5% higher than the one calculated with (EN1264-2, 2009).
In the case "simplified" formulas are applied, the difference becomes very wide and this could lead to huge differences in the system's final design and dimensioning as well.

6.2.4 MIRAGE simulations

6.2.4.1 MIRAGE software

Mirage is a program for solving steady-state heat conduction problems on two dimensional planar and axisymmetric domain, based on the finite element method. The software is divided into three parts: the interactive shell, the mesh generator, the equation solver. The interactive shell contains a CAD-like interface for laying out the geometry of the problem to be solved and for defining material properties and boundary conditions. The mesh generator is the tool in charge of creating the mesh, which is essential for the finite element method. The solver takes a set of data files that describes a heat flow problem and solves the relevant partial differential equations to obtain values for the temperature throughout the solution domain. (MIRAGE Steady State Finite Element Heat Conduction Solver - User Manual 1.0, 2005)

6.2.4.2 Simulation parameters and inputs

In order to verify the values obtained from EN 1264 (2008) and EN 15377 (2008) and determine which calculation procedure reflects in the closest way the behavior of the actual system, simulations using a commercially available software, *MIRAGE,* have been performed. After defining the structure of the layers and their thermal properties, MIRAGE requires certain user-defined boundary conditions.

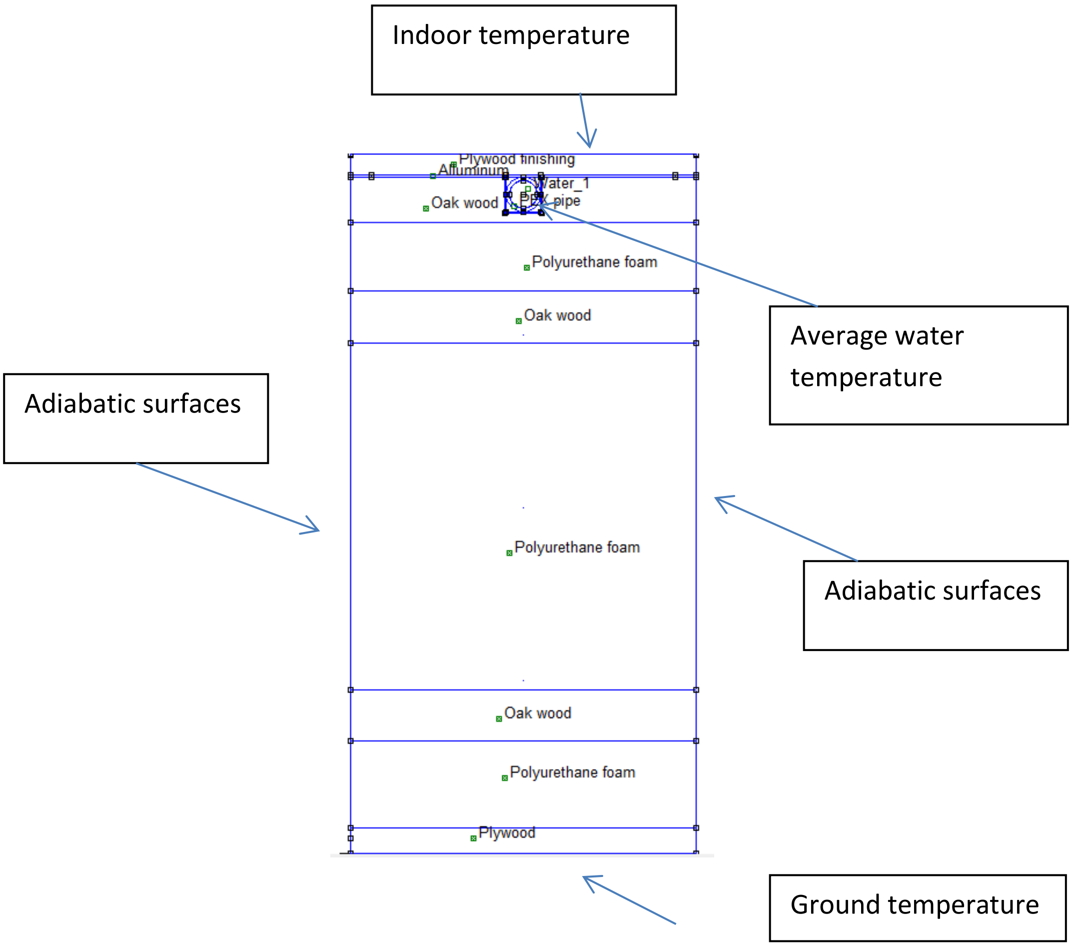


Figure 25: Floor structure. ***Source: MIRAGE***

Boundary conditions:

- *Adiabatic*: applied to the external vertical surfaces, both on the left side and the right side. Since only one pipe has been considered and it has half of the pipe spacing distance on each side, it´s reasonable to assume that those surfaces are adiabatic (no heat flux passing through) due to the presence of other pipes aside.
- *Ground temperature*: the deepest surface is assumed to be in contact with "ground boundary condition", which means a temperature equal to the average soil temperature in Copenhagen (9 °C) and an equivalent coefficient of heat exchange (0.2 $W/m^2 \cdot K$) calculated according to UNI EN 12831 (2006).
- *Indoor temperature*: it means a bulk temperature (indoor environment set point temperature) of 21°C and a coefficient of heat exchange of 10.8 $W/m^2 \cdot K$ (EN 1264-2, 2008). It´s applied to the upper surface of the floor finishing.
- *Water temperature*: applied to the circular inner surface of the water pipe. It consists of a fixed bulk water temperature and a coefficient of convective heat exchange of 2400 $W/m^2 \cdot K$, calculated according to Gnielinski empiric correlation (validity field: $0.5 \leq Pr \leq 2000$ and $3000 \leq Re \leq 5 \cdot 10^6$ – water velocity of 0.5 m/s has been considered). The bulk water temperature is calculated as the average between supply and return temperatures.

The result of MIRAGE simulation is the temperature distribution with the possibility to integrate the values and obtain the heat flux output towards the indoor environment.

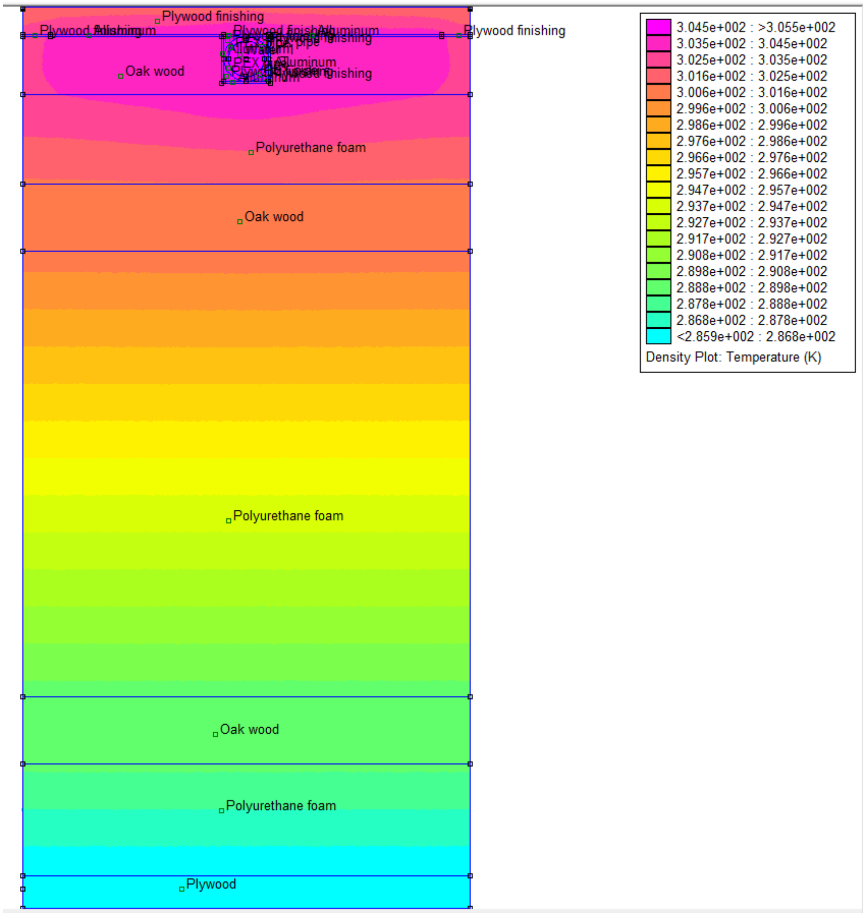


Figure 26: Example of temperature field distribution. Source: MIRAGE

6.2.5 Results

Starting from fixed values, among the range 25÷50°C, for water supply temperature, heat flux output has been calculated according to the different methods previously described and finally using MIRAGE simulation tool. Following are the results, which underline the close agreement between EN 1264 calculations and Mirage simulations. Thus, due to the presented results, EN 1264 appears to be the standard with the best correspondence to MIRAGE software. Temperature drop σ of 5°C has been considered. Figure 27 shows the results.

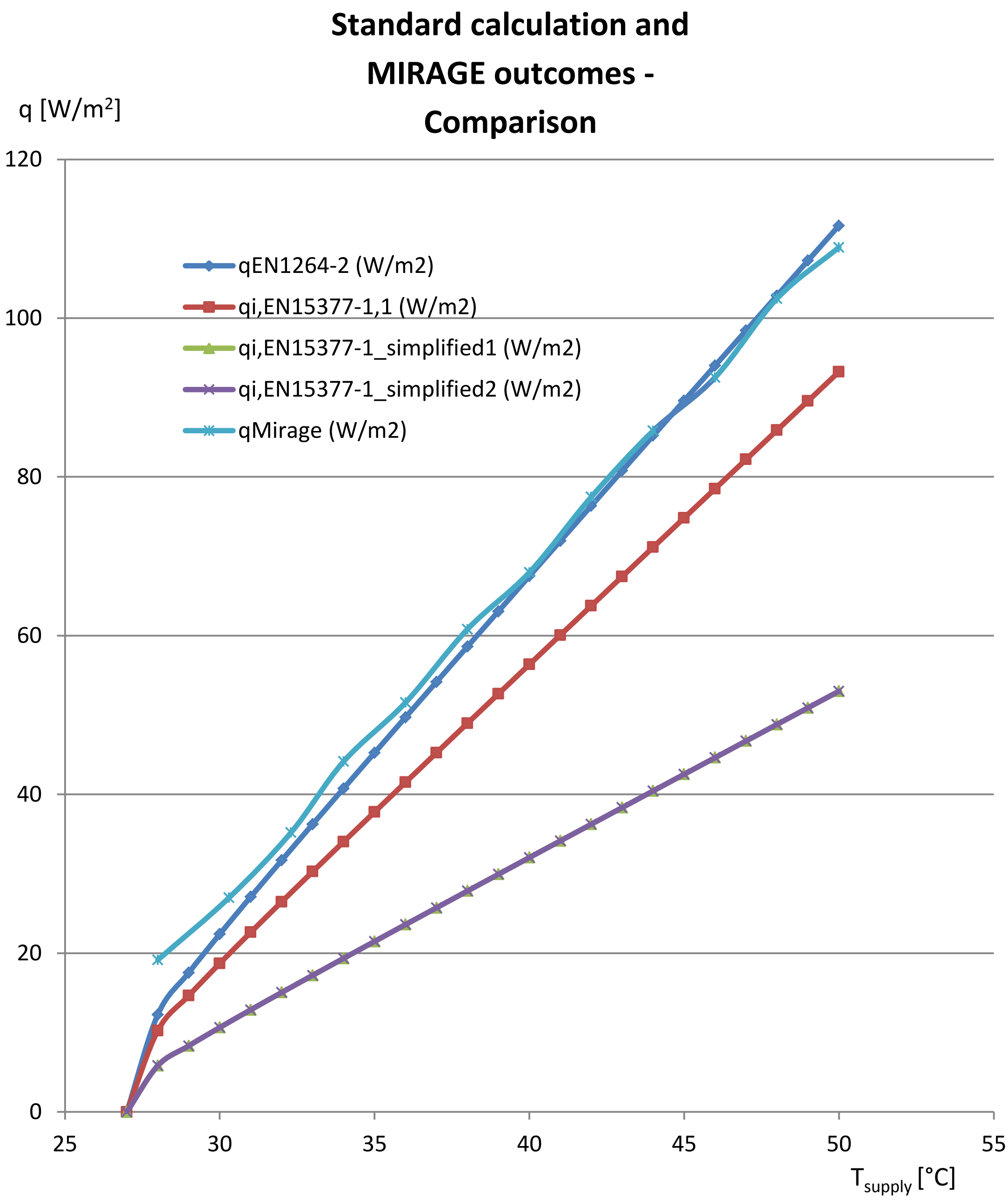


Figure 27: Heat output calculated according to different standard methods and using MIRAGE simulation tool

Results of different standard calculation methods present remarkable differences. MIRAGE heat output are in good accordance with the heat output calculated according to EN 1264, while they differ from the ones calculated according to EN 15377-1. The simplified methods suggested in EN 15377-1 introduce many approximations that result in discordant results. The difference between the three calculation methods increases with higher heat output. For this reason, all further considerations will be based on (EN1264-2, 2008).

7. TRNSYS SIMULATIONS

Once the embedded system has been designed, the following step is to simulate the performances of all the devices, connected together as it will likely be in the future *Embrace*. This process is very important and could be very beneficial to provide the team's members in charge of the design configuration with some feedback about the energy performance of the building. Nevertheless, Embrace's design hasn't reached an ultimate form at the present time, therefore an intermediate design proposal (14th November 2013) has been chosen for this report's purposes. Following are the drafts of the considered model (detailed drawings can be found in the appendix).

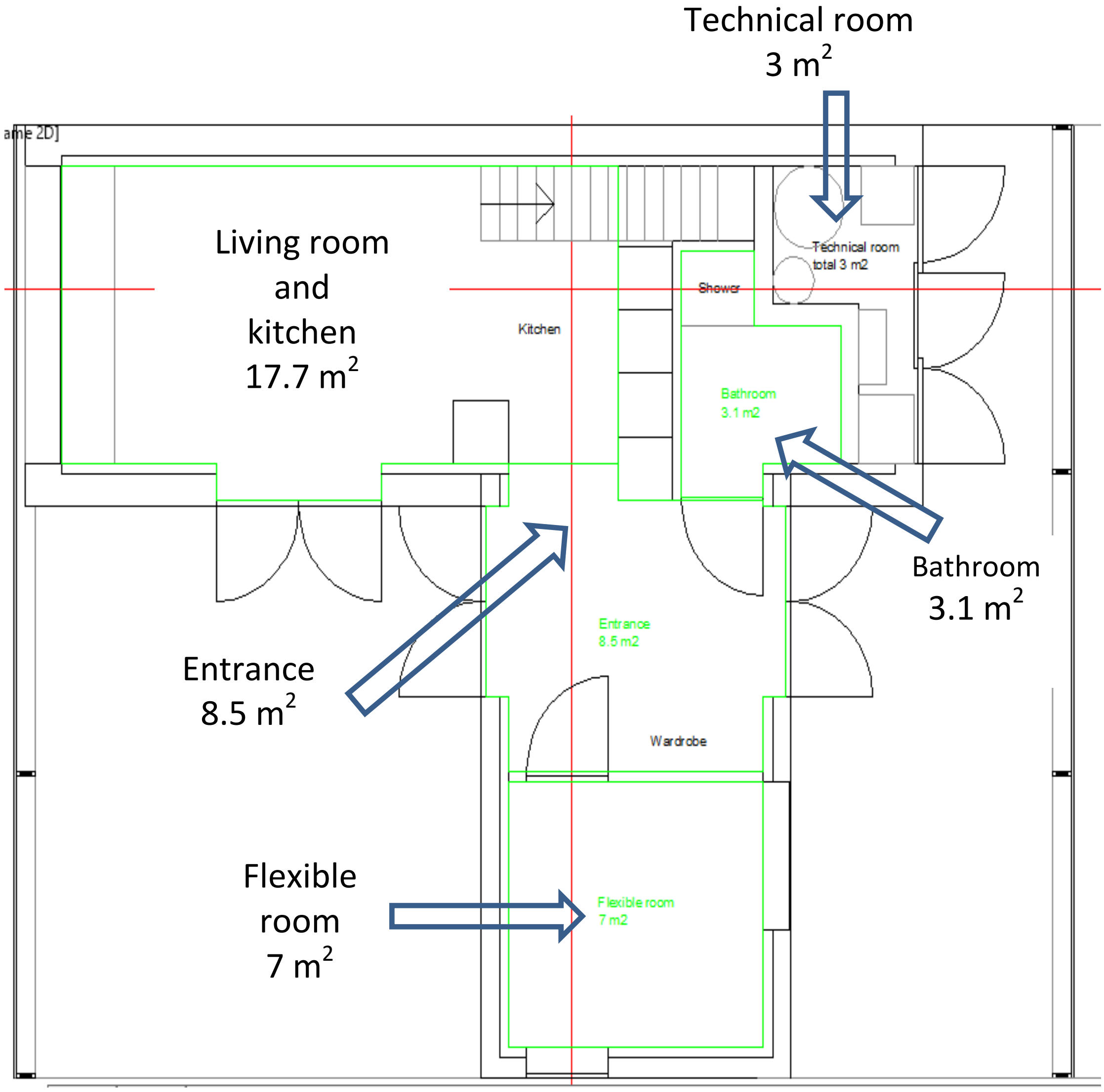


Figure 28: Map of Embrace, ground floor. ***Entrance, living room, bathroom, flexible room or shared space, technical room***

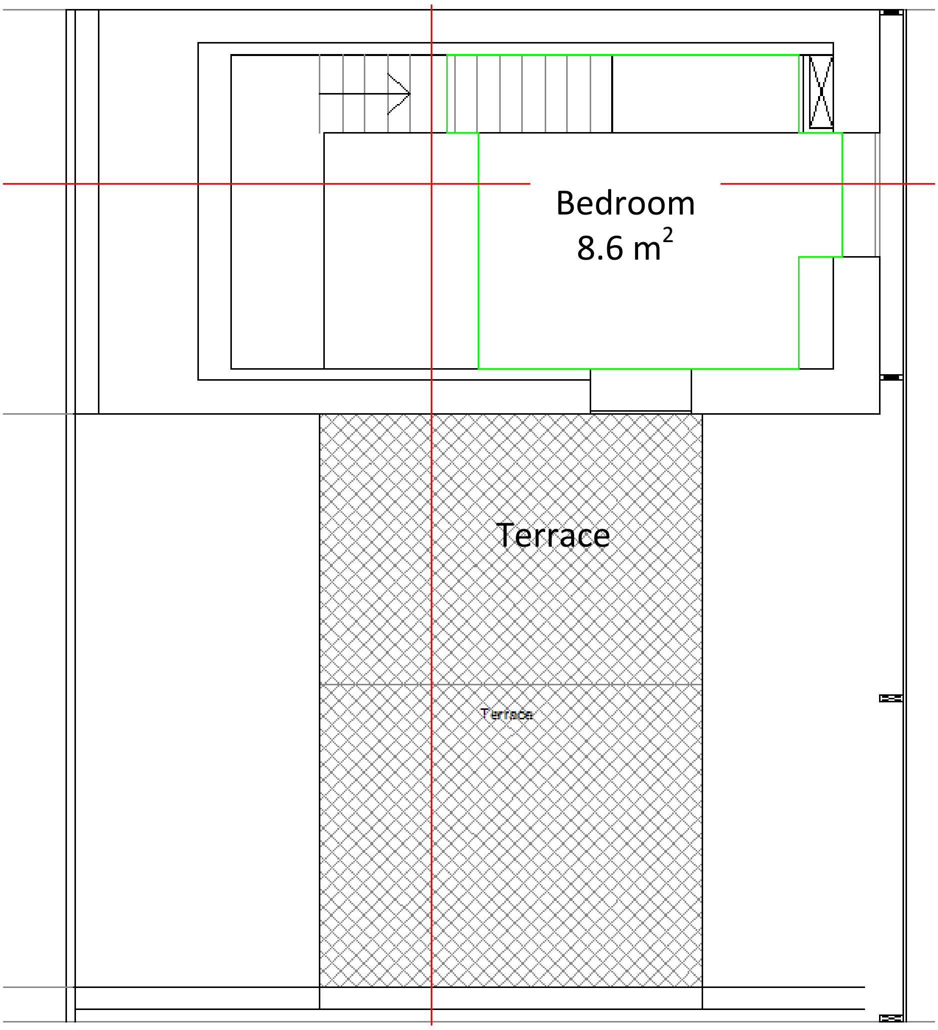
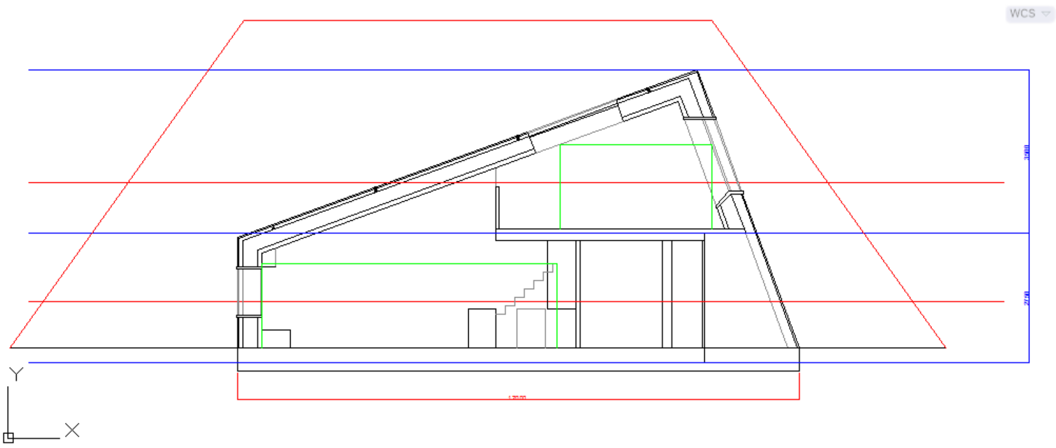


Figure 29: Map of Embrace, first floor. Bedroom and terrace



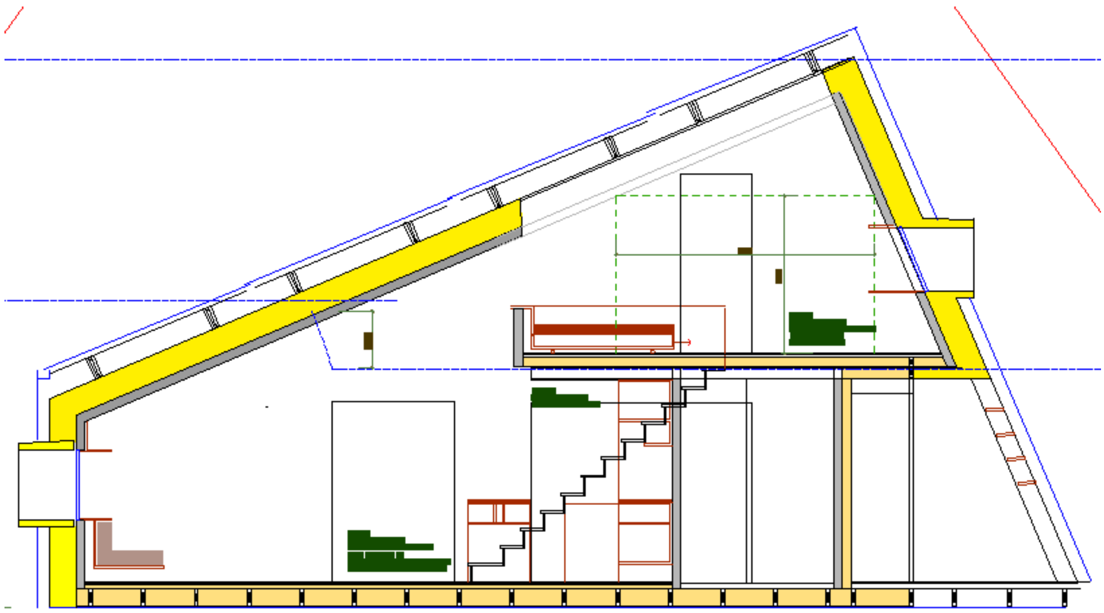


Figure 30: Embrace, lateral views

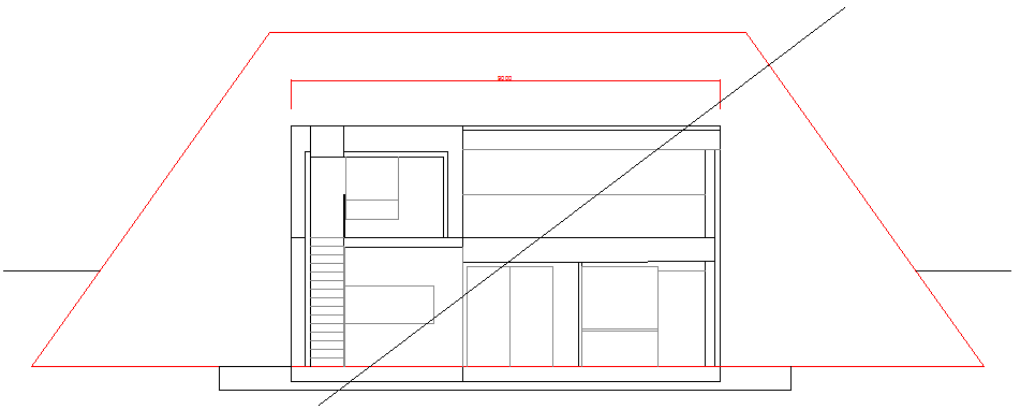


Figure 31: Embrace, front view

Taking into account the presented geometry, a commercially available systems and buildings simulation software TRNSYS (Klein & al., 2009) is used to simulate *Embrace*. TRNSYS is a flexible and graphically based software environment used to simulate the behavior of transient systems; it is able to simulate thermal-electrical energy systems and other types of systems as well, such as traffic flows and biological processes. TRNSYS is made up of two parts. The first is the software "engine" (also called the *kernel*) that reads and processes the input file, iteratively solves the system, determines convergence, and plots system variables. The *kernel* also provides utilities that, among other things, determine thermo-physical properties and interpolate external data files. The second part of TRNSYS is a library of components, each of which models the performance of one part of the system. The standard library includes approximately 150 models ranging from pumps to multi-zone buildings, weather data processors to basic HVAC. Models are

constructed in such a way that users can modify existing components or write their own. An additional set of libraries (TESS libraries) is also used in the simulations.
The main structure of the software consists of two interacting tools, *Simulation Studio* and *TRNBuild*. First the building and its structures are defined in TRNBuild (according to their orientations, properties and materials) and then all the system's components and the building itself are included in the Simulation Studio model. Each element requires some specifications and parameters to be defined. Interacting devices are connected to each other through parameters of interest.

7.1 General structure of the model

The main efforts in simulating the building are focused to the HVAC system and its dynamic behavior all over the year. Its structure is described in chapter 5 and subchapters 5.1, 5.2, 5.3 and in figures Figure 15, Figure 16, Figure 17, Figure 18.
All the components involved in the mentioned drawings are taken into account in the TRNSYS model. Obviously, the HVAC system has to be integrated and "connected" to *Embrace*, which is "reconstructed" in the software through the TRNBuild tool of TRNSYS. Further explanations are given in the next sub-chapters.

7.2 TRNBuild model of Embrace

TRNBuild is the interface of TRNSYS tools for creating and editing all of the geometrical information required by the TRNSYS model of the building. TRNBuild allows editing the walls and layer's materials and properties, to create ventilation and infiltration profiles, to add internal gains (with a certain profile) and define radiant surfaces. Embedded system is therefore specified in TRNBuild.
Since *Embrace* is a two-storey building, the model is based on two principal air-nodes (*GROUND_FLOOR* and *FIRST_FLOOR*), each represents one floor. A supplementary air-node (*TECH_ROOM*) has been added to evaluate roughly the temperatures inside the technical room and its effects on the conditioned space.
Apart from the standard and usual input parameters for a multi-zone building (atmospheric data, temperatures, solar radiation data for each surface, etc.), external inputs are: radiant system supply temperature and flow rates (ground floor flow rate and first floor flow rate), supply temperature and relative humidity for the ventilation system and PVTs back surface temperature.

Embrace is oriented according to the four main cardinal axes and thus the defined orientations are:

Table 17: TRNBuild Surfaces' orientation

Surface n°	Orientation	Angle of inclination from the horizontal
1	South	90° (vertical)
2	South	26°
3	West	90° (vertical)
4	North	90° (vertical)
5	North	62°
6	East	90° (vertical)

Regarding general properties of the simulation and the internal calculation of heat transfers properties, TRNSYS default values were used; they are shown Figure 32.

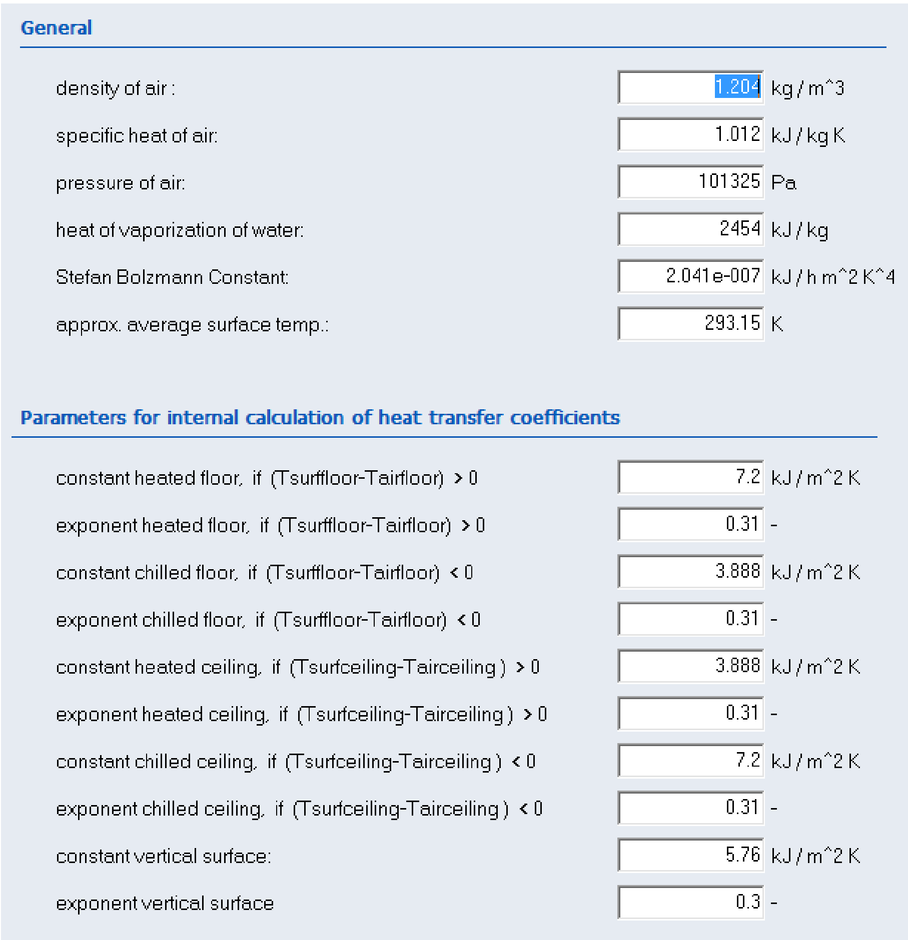


Figure 32: Simulation's general inputs and parameters. ***Source: TRNBuild***

For the heated and cooled surfaces, TRNSYS performs the calculations based on the formula:

$$\alpha_{conv} = (T_{surf} - T_{air})^{\exp}$$

(TRNSOLAR Energietechnik GmbH, 2010)

Initial values of temperature and relative humidity for the air-nodes are 20°C and 50%. Humidity model utilized is *Simple Humidity Model* (Capacitance Humidity Model), details can be found in (TRNSOLAR Energietechnik GmbH, 2010).

7.2.1 General and geometrical inputs

For the unheated surfaces, default values of 3.06 W/m^{2}K and 17.8 W/m^{2}K were used for front and back surfaces convective heat transfer coefficients, respectively (where front refers to internal and back refers to external surfaces). "Internal calculation" option was activated for floor and ceiling. When the U-value of the walls is being calculated, total heat transfer coefficients of 7.7 W/m^{2}K and 25 W/m^{2}K are taken for inside and outside surfaces, respectively.
Default values for solar absorptance of the wall (0.6) and long-wave emission coefficient (0.9) have been considered.
For the beam radiation, diffuse radiation distribution and long-wave radiation exchange within a zone, standard models were used, details can be found in (TRNSOLAR Energietechnik GmbH, 2010).
Walls and surfaces structure are shown in the tables below:

Table 18: External walls structure

Layer	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.03	0.11
Glass fiber	0.3	0.032
Plywood	0.03	0.11

Total thickness is 0.36 m and U-value is 0.099 W/m^{2}K (including α_{int}=7.7 W/m^{2}K and α_{ext}=25 W/m^{2}K). Since no detailed information were given (apart from the insulation material which was fixed as glass fiber due to a certain company's sponsorship), TRNBuild model of *Embrace* has been based on this simple structure. Nevertheless, the overall U-value is in good accordance with the one hypothesized in the steady-state load calculations of the dwelling (see subchapter 4.2 for further explanations). Structure described in Table 18 has been considered also for roof coverings, due to the lack of clear information. PV/T's back surface temperature has been considered as the boundary condition for the 26° tilted roof facing South.

Internal walls have simpler structure and they are assumed to be constituted of exterior plywood finishing and an internal core of oak wood, as the table below shows:

Table 19: Non-insulated internal walls structure

Layer	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.02	0.11
Oak wood	0.06	0.16
Plywood	0.02	0.11

Total thickness is 0.10 m.
Since the ground floor area internally borders on the technical room too, an insulated internal wall is assumed to divide the zones, providing higher thermal insulation in order to prevent excessive and unwanted losses (or gains) towards the non-conditioned zone. Its structure is hypothesized as follows:

Table 20: Insulated internal walls structure

Layer	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.015	0.11
Oak wood	0.02	0.16
Glass fiber	0.08	0.032
Oak wood	0.02	0.16
Plywood	0.015	0.11

Total thickness is set as 0.15 m.

Ground floor structure has been considered identical to the one used for load calculations (subchapter 4.2) and Mirage simulation (subchapter 6.2.4). Despite the radiant system is actually integrated in the middle of an oak wood layer, due to TRNBuild requirements the wooden layer is split in two identical layers, each of which is 0.06 m thick (thus satisfying minimum thickness requirement for the layer above the radiant system, which has to be ≥ 0.3·T, T being the pipe spacing equal to 0.2 m). The layer underneath the active one is included automatically and it's defined as identical to the one above the pipes. The overall effect of the layers named *Oak wood (fictitious)* is set as the same of the actual wooden layer in terms of thermal properties. Due to numerical constraints of the software in terms of maximum allowed thickness and wall's transfer function calculation, an additional fictitious layer (*Fictitious back layer*) has to be defined. It is 0.2 m thick and has equivalent thermal behavior to the actual materials below the embedded pipes. Therefore, ground floor's total thickness has no physical meaning.

Table 21: Ground floor structure

Material	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.012	0.11
Aluminum	0.001	200
Oak wood (fictitious)	0.06	0.64
Pipe	active layer	
Oak wood (fictitious)	0.06	0.64
Fictitious back layer	0.2	0.020

Ground floor structure described in the previous paragraph applies to the overall ground floor area but the one included in the technical room, which is not supposed to be conditioned in any way. In this case the structure is simpler and, once more, due to the lack of information, it has been hypothesized as follows:

Table 22: Technical room's ground floor structure

Layer	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.012	0.11
Oak wood	0.05	0.16
Fictitious back layer	0.2	0.020

The structure is mostly identical to the one constituting the rest of the ground floor, obviously apart from the embedded pipe system. Similarly to the rest of the ground floor, the fictitious back layer has been included in the structure underneath the technical room as well. Bulk temperature boundary condition for both of the ground floor structures is set as the approximate yearly average air temperature for Copenhagen (9°C), which is consider almost equal to the yearly average ground temperature. This choice is due to several lacks of information about structures and positioning.

Embedded pipe system is identical in both of the floors. It is defined as *Active Layer* in TRNBuild and standard mode for the parameters is applied. Specific heat coefficient of the fluid is set as 4.18 kJ/kg·K being it water, pipe spacing is 0.2 m, pipe external diameter is 0.02 m, pipe wall thickness is 0.002 m, pipe wall conductivity is 0.35 W/mK. Pipe features are the same used in chapter 6. In the *Active Layer Specification*, it is required to specify the number of pipe loops: since the “Autosegmentation” option is not possible to apply (this is probably due to the small floor area of *Embrace*), ground floor is assumed to be constituted of two loops and the intermediate floor of a single loop. Default values are kept in the other parameters of *Active Layer Specification.*

According to some rough design guidelines, approximate thickness of the first floor had to be around 0.30 m. Considering a required space for ventilation ducts and other installations of 0.10÷0.15 m, available space is thus of 0.15÷0.20 m. Total thickness of 0.18 m has finally been considered and the structure is the following (it should be kept in mind that the total layer 0.12 thick of oak wood (fictitious) is equivalent to a layer of actual oak wood 0.03 m thick):

Table 23: Intermediate floor structure

Layer	Thickness [m]	Thermal conductivity λ [W/m·K]
Plywood	0.01	0.11
Oak wood (fictitious)	0.06	0.64
Pipes	Active layer	
Oak wood (fictitious)	0.06	0.64
Glass fiber	0.07	0.032
Oak Wood	0.06	0.16
Plywood	0.01	0.11

Pipe system has been assumed as identical to the one embedded in the ground floor. First floor surface facing the technical room has been excluded from the overall active surfaces because it is in contact with a non-conditioned space and it would cause excessive temperature drops in the radiant system.

Doors are modeled as a different wall structure, constituting of a massless material (since door's additional thermal mass in building's overall is negligible) of a given thermal resistance of 1.276 m^2K/W (roughly estimated as a layer of plywood 0.14m thick). The overall U-value considering α_{int}=7.7 W/m^2K and α_{ext}=25 W/m^2K is approximately equal to 0.7 W/m^2K and it's therefore in accordance with the one hypothesized in the load calculations subchapter 4.2.

Windows properties have been taken from TRNBuild standard components library. Chosen window is assumed to be *Pilkington OPTITHERM* triple glazing (*4/16/4/16/4*), which air gaps are filled with insulating gas (Argon). Its U-value and g-value are 0.7 W/m^2K and 0.501 respectively (U-values are in accordance with the one hypothesized in load calculations – see subchapter 4.2 for further references).

Table 24: Windows orientations and area. Table 17 is the reference for the orientations

Orientation n°	Window area [m^2]	Zone
1	2.52	Ground floor
2	1.39	First floor
4	1.82	Ground floor
5	0.8	First floor
6	0.9	Ground floor

Since the interior of the dwelling is split into two different zones, it is necessary to introduce fictitious windows among the internal space, in order to simulate the actual openings and take them into account when calculating the thermal balances. TRNBuild library provides a particular window for this purpose called *No Glazing*: its U-value is 5.68 W/m^2K and g-value is 1.

Materials included in the structures previously described are characterized by the following properties:

Table 25: Materials properties

Material	Thermal conductivity [W/m·K]	Specific heat capacity [kJ/kg·K]	Density [kg/m^3]
Aluminum	200	0.86	2700
Glass fiber	0.032	0.9	35
Oak wood	0.16	2.7	700
Oak wood (fictitious)	0.64	2.7	175
Plywood	0.11	1.6	560
Fictitious back layer	0.020	2.09	300

Table 26 reports the volumes of the air-nodes taken into account.

Table 26: Air volume for each air-node in the model

Air-node	Air volume [m3]	Capacitance [kJ/kg]
Ground floor	96	115.2
First floor	30	36
Technical room	8	9.6

7.2.2 Other inputs

Apart from the structures, TRNBuild requires the definition of other features of *Embrace*.

Infiltration is set as constantly equal to 0.1 ACH. This value is the same as the one hypothesized in load calculations section (chapter 4.2).

Ventilation load calculation has been based on sensory load removal, avoiding the consideration of chemical load removal. Design value of 1.25 ACH is thus obtained, assuming to use mixing ventilation (which effectiveness is considered equal to 1.0 if temperature difference between supply air and room air temperature is less than zero - (CR 1752, 1998)) This choice has been made since no deeper details about ventilation system had been fixed. Supply air temperature and relative humidity are obtained from external components in the Simulation Studio environment (further explanations are given in the following chapters).

Internal gains have been split into three different types:

- *Occupants*: Occupancy is modeled in TRNBuild according to ISO 7730. The metabolic activity of two people, seated and doing a very light writing activity has been considered (source: TRNBuild – Internal gains). Total effect is the product of metabolic activity of two people and the occupancy schedule (sum of weekday schedule and weekend schedule), modeled for the ground-floor air-node as follows:

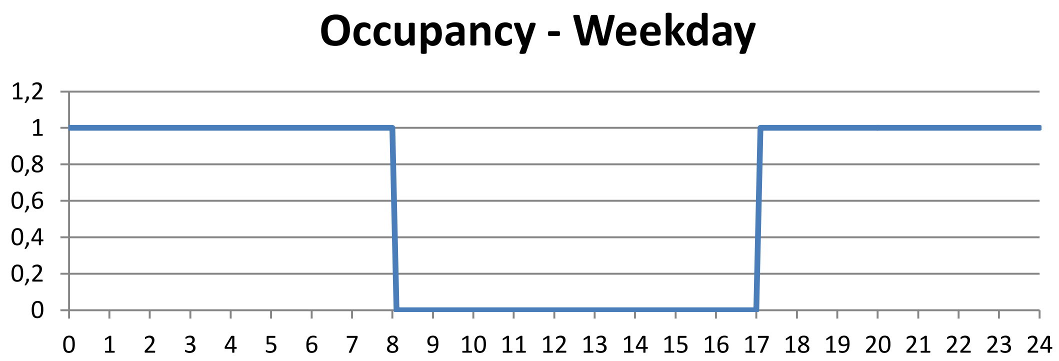


Figure 33: Occupancy schedule - Weekday

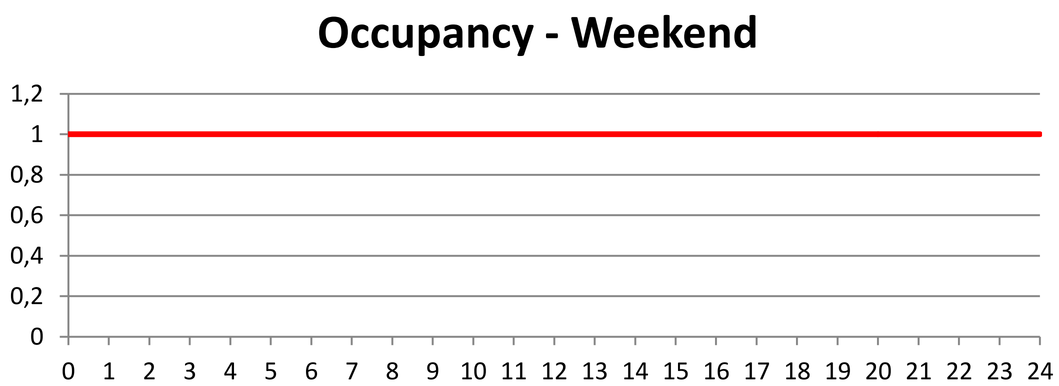


Figure 34: Occupancy schedule – Weekend

Since first floor is meant to be used as bedroom, occupancy schedule for *First floor* air-node is different and it is shown in Figure 35.

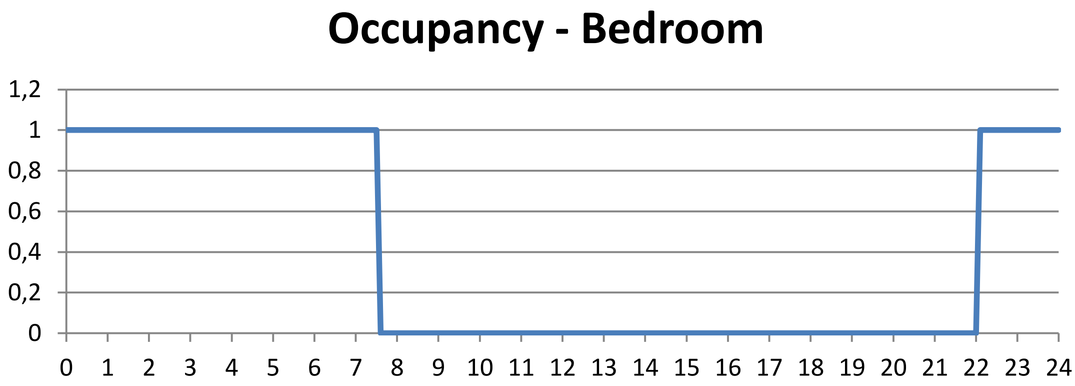


Figure 35: Occupancy schedule for the bedroom (first floor air-node)

Other parameters have been implemented.

- *Artificial Lighting* is considered. Total heat gain of 5 W/m^2 is considered, with a floor area of approximately 43 m^2. The value is higher than the one considered in the load calculations (3 W/m^2) but it has been chosen since it is the lowest possible in TRNBuild option list. Lighting schedule is identical for each day of the week and it is defined in Figure 36. Total effect is the product of internal gain due to artificial lighting and lighting schedule. Convective part is considered 30% of the total gain (11221: Ventilation and Climatic systems, Load Calculations, 2006).

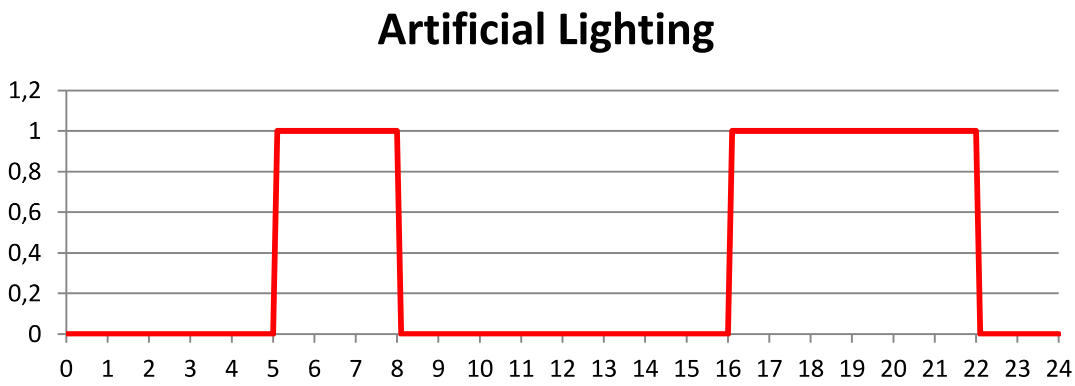


Figure 36: Artificial Lighting schedule

- Other internal gains have been assumed as explained in chapter 4.2. Power of 0.46 kW is considered, 30% of which being convective and 70% being radiative (11221: Ventilation and Climatic systems, Load Calculations, 2006). Overall effect is the product of internal gain values and internal gains schedule, which is the following:

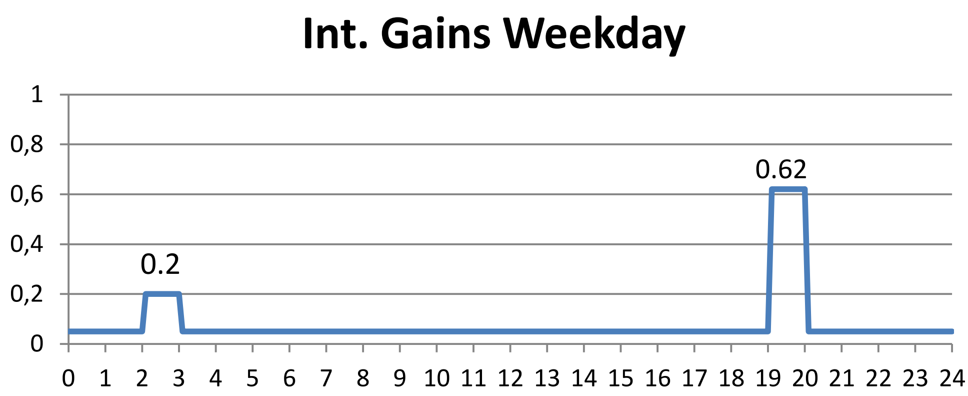


Figure 37: Internal gains schedule - Weekday

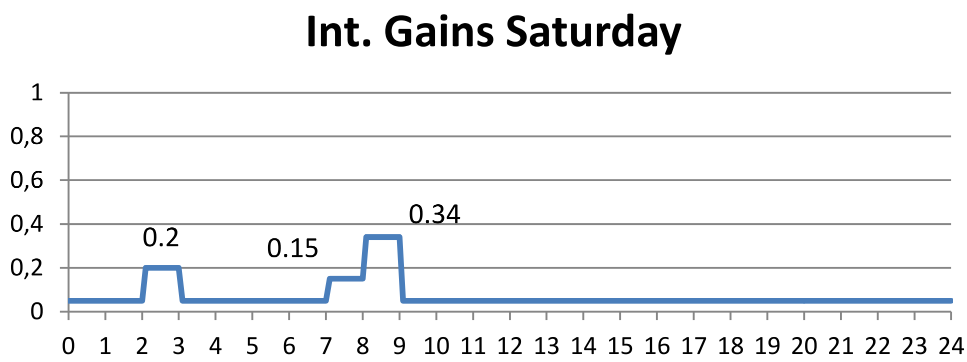


Figure 38: Internal gains schedule - Saturday

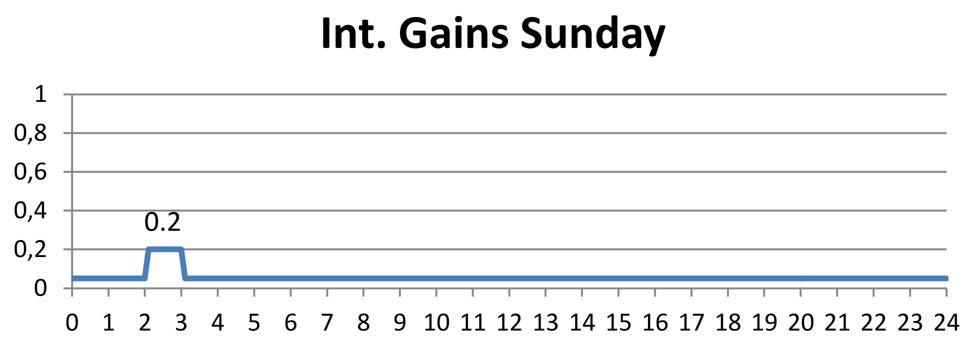


Figure 39: Internal gains schedule – Sunday

In particular, the peak shown in Figure 37, Figure 38 and Figure 39 between 2 AM and 3 AM is associated with the activation of the dish washer. The evening peak in weekdays schedule is associated with the use of the oven. The morning peak on Saturday morning is associated with the use of the clothes washer and the clother dryer afterwards.

Once Embrace is completely defined in TRNBuild, it represents a single component in the *Simulation Studio* environment. It requires other components to feed it with the necessary inputs and it provides outputs to be used by other components. Standard output list is extended to make it include:

- Outlet ground floor's embedded system water temperature
- Outlet first floor's embedded system water temperature
- Ground floor surface temperature
- First floor surface temperature

Windows are considered fully closed at any time. Furthermore, no natural ventilation effect has been included in the simulations.
Further explanations of Simulation Studio model are given in the following chapters.

7.3 Simulation Studio

Simulation Studio is the TRNSYS environment where the simulation actually takes place. It requires the house to be implemented and connected to other external components, which interact among each other. Every type of component is characterized by a unique type number which identifies the element. *Embrace* is modeled as a *Multi-zone building* (type 56), thus allowing using a conditioned air zone (*Ground floor* and *First floor*) and a non-conditioned air zone (*Technical room*). Type 56 requires many default inputs from the weather file and moreover other supplementary inputs have been added as explained in subchapter 7.2.2.
Among control cards inputs, default values are used and in particular simulation time-step of 1 hour is applied during yearly simulations (from 0 to 8760 hours).

7.3.1 Weather file and season schedule

Weather data are essentials in building simulations, especially in *Embrace*'s case where the same house is required to achieve certain energy performances in Copenhagen's climate and Paris' climate. The component to include weather data in Simulation Studio is *Type 15-3, Energy Plus weather file, EPW* (Weather data reading and processing, standard format). Therefore Energy+ weather file are considered for both of the climates.
Season schedule component to split the calculation in winter conditions and summer conditions has been added. The necessary component is *type 14h, Time dependent forcing function*. Winter period was initially considered including January, February, March, April, October, November and December and summer period the remaining months, which meant that winter conditions hours were from 0 (1st January) to 2880 (30th April) and from 6550 (1st October) to 8760 (31st December) (Olesen, Sommer, & Duchting, 2002). First simulations showed several problems in indoor environment quality (due to the indoor temperature, which was often too low to meet the comfort requirements), especially in May and September, which were supposed to be part of summer period. The problems were due to the difficulty of the model to switch between heating conditions and cooling conditions

in summer case quickly enough to obtain satisfying indoor temperatures in the first and the last summer months, when outdoor temperatures are not very high yet. The problem has been solved extending the winter season and making it include hours from 0 to 3500 and to 6000 to 8760. This means that winter season includes day within the period 1st January – 20th May (approximately) and within the period 11th September (approximately) – 31st December. Summer season therefore includes day within the period 21st May – 10th September.
Regarding the presence of the weather shield, decreased solar radiation values are considered for the surfaces underneath it. The scale-factor is assumed to be the g-value, being 0.35 for the 26° tilted surface facing south and 0.55 for the horizontal surface (as indicated by DTU’s SDE 14 daylight group), 0.855 for the remaining surfaces, which are the vertical walls facing south and north and the 62° tilted wall facing north (clear single glazing – TRNSYS library for glazing surfaces). For the sake of ease and due to problems in the implementation in TRNSYS simulation studio, temperature taken from the weather file has been considered, regardless of the presence of the weather shield and its influence.

7.3.2 Hydronic system

Hydronic system has been modelled according to Figure 15, Figure 16, Figure 17, Figure 18 drawings. Detailed explanations about the operating modes are given in chapter 5. It includes two separate tanks, the first for domestic hot water purposes and the second for space heating and cooling purposes, PV/T panels, a heat pump, pumps, tee-pieces, diverters and all the necessary devices to make the system work.
The system is fed (both of heating and cooling energy) by the PV/Ts and the heat pump. Operating modes, as explained in chapter 5, can be summarized as follows:

- In winter-time, PV/Ts provide the necessary heating energy for domestic hot water and space heating. If PV/Ts outlet temperature is not high enough, heat pump is activated. Heating energy is provided to *Embrace* through the radiant floor.
- In summer-time, PV/Ts are in charge of feeding the domestic hot water tank and the heat pump is required to keep the water storage tank cooled to provide enough cooling energy to the house through the radiant floor. If PV/Ts outlet temperature is not high enough to feed the DHW tank, it can rely on its auxiliary heater. No heating control strategy in summer-time has been implemented.

7.3.2.1 PV/Ts hydronic system

The main component of this system is the PV/Ts. Their features were a controversial issue among DTU’s SDE team at the time the building of the model was started. Therefore, for the purposes of the simulations, WIOSUN PVT-P series products have been chosen (Wiosun - Combined Modules - PVT-P series, Datasheet, 2013). Default values are used for data not included in the datasheet.

Table 27: PV/Ts main features and properties

Collectors length	10	m
Collectors width	3.6	m
Absorber plate thickness	0.001	m
Thermal conductivity of the absorber	360	W/m·K
Number of tubes	20	
Tube diameter	0.015	m
Bond width	0.01	m
Bond thickness	0.001	m
Bond thermal conductivity	1300	W/m·K
Resistance of substrate material	3	m^2·K/W
U-value for roof material	0.2	W/m^2·K
Fluid specific heat	4.19	kJ/kg·K
Reflectance	0.15	
Emissivity	0.9	
1st order IAM	0.1	
PV cell reference temperature	25	°C
PV cell reference radiation	1000	W/m^2
PV efficiency ate reference condition	0.163	
Efficiency modifier - temperature	-0.0034	1/°C
Efficiency modifier - radiation	0.00009	m^2/W
Collectors slope	26°	
Top loss convection coefficient	8.7	W/m^2·K
Fluid heat transfer coeffiecient	390	W/m^2·K

PV/Ts component is *type 563, PV/T collector, interacting with detailed zone models*, taken from the TESS additional libraries.

Top loss convection coefficient has been calculated according to the following equation, taking into account 2 m/s as wind speed (Rahbek, 1995).

$$h_{top\_loss\_convective} = 2.8 + 3 \cdot v_{wind}$$

The figure below shows the outline of the PV/Ts hydronic system.

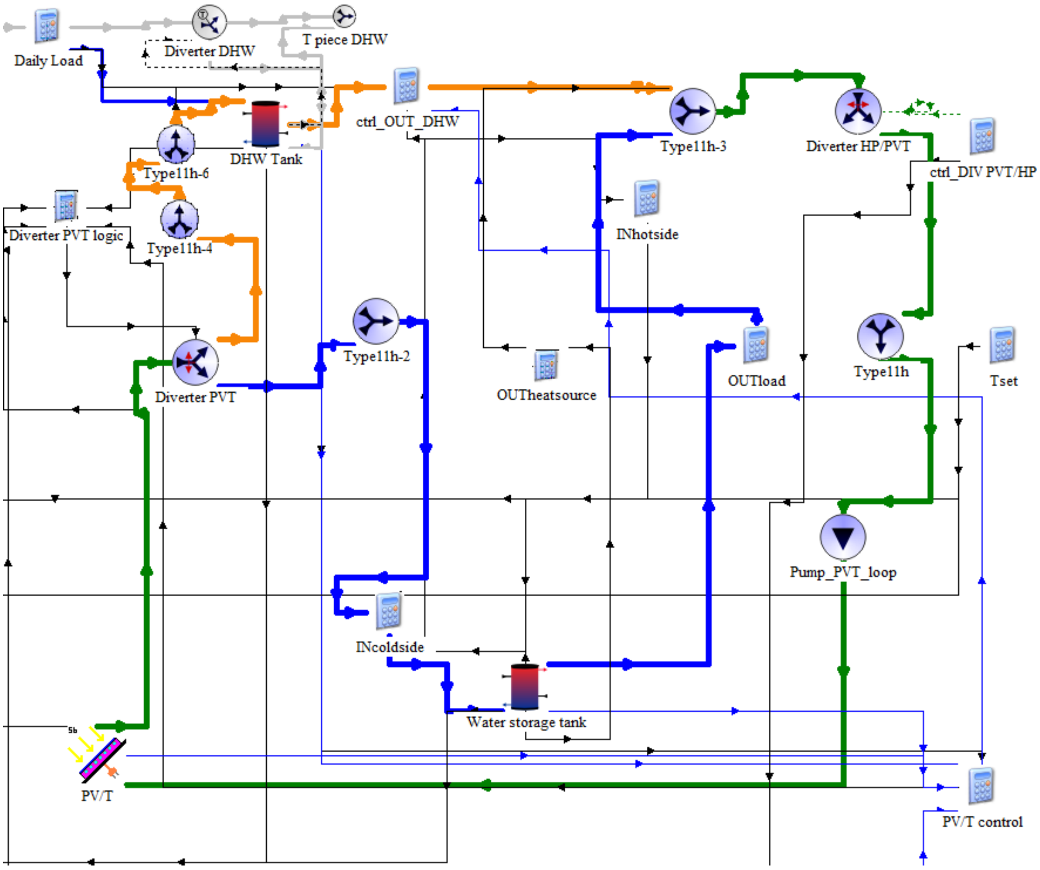


Figure 40: Visualization of the PV/Ts hydronic loop. Orange arrows indicate day-time operation; blue arrows indicate nocturnal operation; green arrows are common. Source: TRNSYS Simulation Studio.

On the outlet side PV/Ts are connected to the domestic hot water tank, except during night-time in summer season, when they are in charge of cooling down the water storage tank due to the combined effects of convective and radiative night-time cooling. PV/T loop pump is supposed to be characterized by a maximum flow rate of 320 kg/h, which is the design flow rate of the radiant floor in cooling season (higher than the design value in heating conditions, equal to 220 kg/h). Although the flow rate is not constantly maintained at the maximum value:

- Flow rate is equal to 220 kg/h in winter-time
- In summer season, flow rate of 320 kg/h is circulated during day-time
- Flow rate of 260 kg/h is used during night-time in summer season.

The pump feeding the PV/Ts has a maximum power of 35 W (Grundfos UPS2 brochure, 2013).
The system has been implemented as an unique big loop, despite it will not be the case in reality because a decoupling heat exchanger will separate the hydronic loops.

The idea of using different flow rates between night-time and day-time in summer conditions derives from the need to cool down the water storage tank during the night allowing the water to lose heat via being circulated through the PV/T collectors. Since the heat able to be exchanged is fixed (it depends on weather conditions and collector features), lower flow rates generate higher temperature differences which means lower outlet temperatures. This gives more stability to the model and results in better temperature distribution all over the water storage tank. The use of flow rates lower than the design value in cooling conditions doesn't affect significantly the indoor comfort, as explained in subchapter 9.1.

PV/Ts pump is operated according to a control, implemented in an equation component. Conditions to be satisfied to operate the PV/Ts pump during the day are:

- Average PV/T surface temperature lower than 95°C for safety reasons
- Outlet PV/T temperature lower than 90°C to prevent boiling inside the collectors
- Outlet PV/T temperature higher than 55°C, which is assumed to be the lowest permissible temperature to feed the domestic hot water tank
- Solar global radiation on PV/Ts tilted surface higher than 50 W/m^2 (if this conditions is satisfied it's assumed to be in day-time)
- Temperature of the outlet flow directed "to load" lower than 55°C.

These conditions need to be satisfied both in winter and in summer. If not, heat pump is activated to substitute the collectors. Further studies could lead the optimization of the set-points mentioned above in order to decrease the operation of the heat pump.

During night-time (and therefore only in summer case), the pump that feeds the PV/Ts is operated when the average temperature in the water storage tank is higher than 15°C, thus guaranteeing an appropriate temperature level inside the tank which is in charge to feed the radiant system.

After the water has circulated into the PV/Ts, a diverter, named *Diverter PVT* controlled by an external equation called *Diverter PVT logic*, is required. It directs the PV/Ts outlet flow towards DHW tank or water storage tank, according to the control logic:

- The flux is directed to the DHW tank if the following conditions are satisfied:
  - In winter season, the pump feeding the PV/Ts needs to be operating
  - In summer season, during day-time, the same condition applies
- The flux is directed to the water storage tank in every other case, which means when the pump is operating during night-time, in summer season.

7.3.2.2 Heat pump hydronic system

Heat pump and PV/Ts are operated alternatively in winter season, simultaneously during summer season. Heat pump component is *type 941, Air to water heat pump* (taken from TESS libraries).

Heat pump hydronic loop is presented in the following figure.

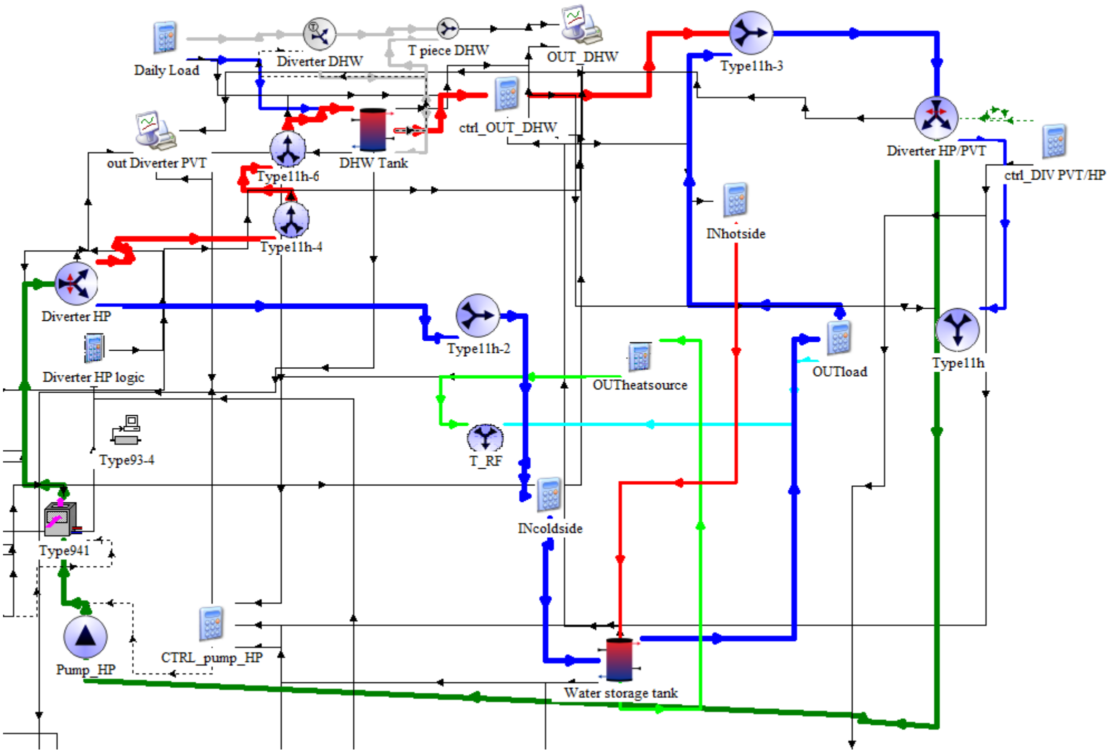


Figure 41: Visualization of the heat pump´s hydronic loop. Red arrows indicate the winter-operation mode; blue arrows indicate the summer-operation mode; green arrows are common for both of the seasons. Source: TRNSYS Simulation Studio

At the time the simulation model was defined, heat pump hadn't been specified. Based on the results of initial simulations, a heating capacity of 10 kW has been chosen, as well as an identical cooling capacity. It should be kept in mind that the heat pump is a crucial component in the balance of the system; since no technical data had been specified at the time the simulations were carried out, the choice of the heating and cooling capacities has been based on assumptions and on the results of initial simulations. However, it should be kept in mind that these considerations were observed simulating the system in the case it is equipped with the buffer tank. Therefore, despite relatively low cooling demands, the heat pump is characterized by a capacity of 10 kW. Default values have been considered for the other parameters. Further simulations implementing more detailed data about the heat pump could lead to more precise results.

It requires as input a couple of control values to indicate whether the device is operating in cooling mode or heating mode. For this purpose an external equation

(named *HP_control*) has been added, indicating winter as the heating season and summer as the cooling season.
After the heat pump, depending on its operating mode and the season, the flux needs to be directed towards DHW tank or water storage tank. Heat pump outlet flow feeds DHW tank in winter season; in the other cases the flux is directed towards the water storage tank.
The pump that feeds the heat pump loop (*Pump_HP* in *Simulation Studio*, Figure 41) is activated according to the following conditions and has a maximum power of 35 W (Grundfos UPS2 brochure, 2013):

- In winter season and in particular periods of summer season when heating is required:
  - Heat pump heating control input has to be activated (it is season-dependant)
  - PV/Ts pump has to be deactivated (this additional check condition isn't directly connected to the heat pump but it has been included to avoid having a heat pump's outlet flow and a PV/Ts outlet flow and the same time)
  - DHW tank outlet temperature "to load" has to be lower than 55°C.
- In summer season, during daytime, when cooling is required:
  - Heat pump cooling control input has to be activated
  - Average water storage tank temperature has to be higher than 15°C.

The pump is connected to the system before the water storage tank. The tank is meant to be a buffer component therefore the heat pump will not be harmed by an excessive number of ON/OFF switches.

7.3.2.3 PV/T – HP diverter

Since, according to the season and other circumstances, both the heat pump outlet flux and the PV/Ts outlet flux are feeding the domestic hot water tank, diverters and T-pieces are necessary. The diverting element named *Diverter HP/PVT*, controlled by an external equation called *ctrl_DIV_PV/HP*, directs the water storage tank outlet flux (*to heat source* in winter season and *to load* in summer season) either towards the PV/Ts or the heat pump. The control logic is:

- To have a flux towards the PV/Ts the following conditions need to be satisfied:
  - In winter-time, the pump operating the PV/Ts needs to be activated (and thus satisfying the conditions in subchapter 7.3.2.1)
  - In summer season, during daytime, when the pump operating the PV/Ts is activated
  - In summer season, during night-time, in any case
- In any other case the flux is directed to the heat pump

7.3.2.4 Domestic hot water

Domestic hot water tank is modelled through *type 4e, Water storage tank, user-designated inlets, uniform losses* (as well as water storage tank).
DHW hydronic loop is presented in the following figure.

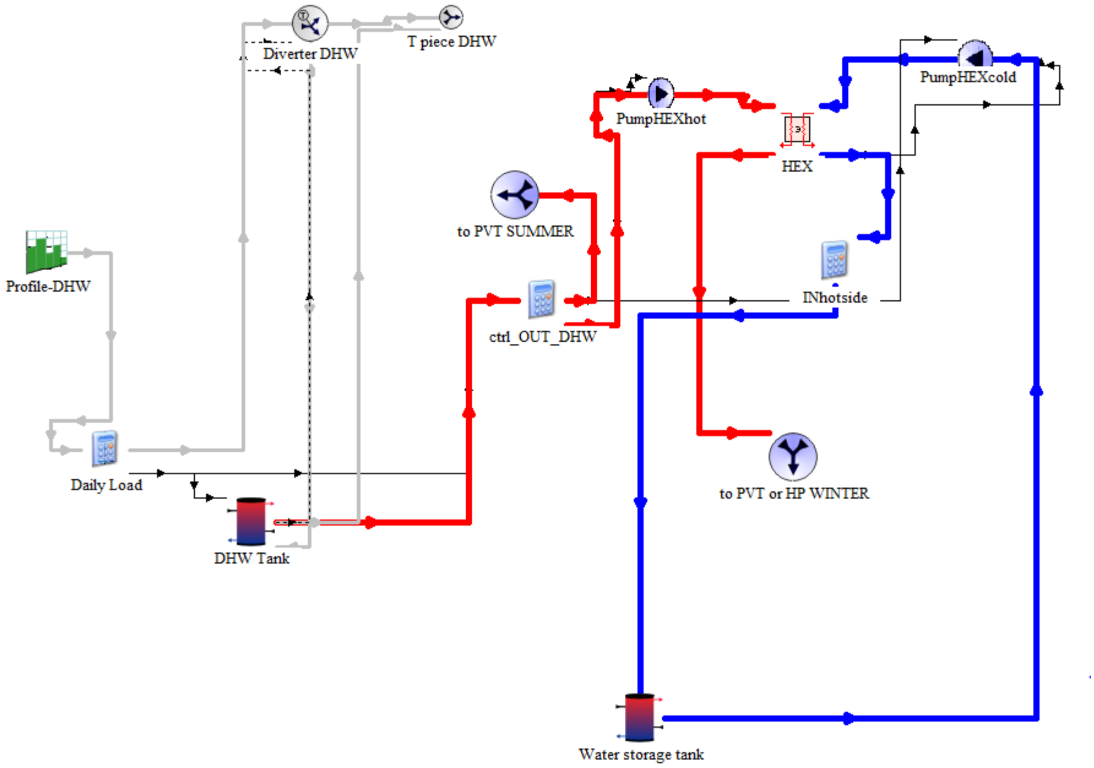


Figure 42: Visualization of the DHW hydronic loop. Red arrows indicate heat exchanger hot-side flows; blue arrows indicate heat exchanger cold-side water flows. Source: TRNSYS Simulation Studio

Among tank´s parameters, due to some constraints regarding tapping response and utilization fixed by SDE organization, tank volume has been indicated to be 300 liters (0.3 m^3). Since it contains water, specific heat and density are set as 4.190 kJ/kg·K and 1000 kg/m^3 respectively. Tank loss coefficient is 0.87 W/m^2·K: no accurate data about the tank were available, therefore this values has been taken from DTU´s SDE 2012 house (*Fold*) TRNSYS model (Kazanci & Skrupskelis, 2012). The component requires the definition of a certain number of nodes (or temperature levels) to use in the calculation on thermal balances and temperatures distribution: 10 nodes have been implemented, each of which is 10 cm (0.10 m) high; therefore the total height of the tank is 1 m. One auxiliary heater is defined (despite the possibility of having two of them, one has been avoided): it is positioned in node number 2, as well as its thermostat. Set point temperature for this device is 55°C, with a dead-band of ±5°C. Auxiliary heater maximum power is set equal to 1.5 kW according to (Kazanci & Skrupskelis, 2012); second auxiliary heater is deactivated by

setting its maximum heating rate as null. Hot-source fluid is assumed to enter at node number 4 while cold-side fluid is supposed to enter at node number 9.
Among tank´s inputs, flux coming from PV/Ts or heat pump feeds hot-side temperature and flow rate, while cold-side inputs are represented by the return flow from the component named *Diverter DHW*. Environment temperature is assumed to be technical room´s temperature, where the tank is supposed to be installed.

Among tank´s outputs, the flux directed to heat-load feeds the component named *T-piece DHW*, while the flux directed to heat-source feeds:

- A heat exchanger decoupling DHW tank and water storage tank in winter-time and during summer season when heating is required
- The PV/Ts loop bypassing the intermediate heat exchanger in summer when heating is not required.

DHW tapping profile is based on (EN15316-3-1, 2007) – tapping profile for single family dwellings – tapping program n°2 (European use). Tapping profile is adjusted according to the desired draw-off temperature of 55°C and corresponds to the one used in (Kazanci & Skrupskelis, 2012).

As previously mentioned, a heat exchanger is inserted between the DHW tank and the water storage tank, in order to decouple the fluxes and avoid domestic hot water from flowing directly into the other tank. Heat exchanger is modelled as *type 91, Heat exchanger with constant effectiveness*, set equal to 0.9. On its source-side the flow is ensured by a single-speed pump (type 3), named *PumpHEXhot*, while the load-side is fed by an identical pump named *PumpHEXcold*. *PumpHEXhot* receives the flux “to heat-source” from DHW tank and feeds the source-side of the heat exchanger; its source-side outlet flux is mixed with the water coming from PV/Ts or heat pump and returns into the DHW tank. *PumpHEXcold* draws water from a low-level node of the water storage tank (node 40 out of 50, where node 1 is the top-most), pumps it towards the load-side of the heat exchanger and after gaining a certain temperature sends it to the water storage tank´s hot side inlet. The main control parameter is called *ctrl_pumpHEXhot*, which activates the pumps in the following cases:

- In winter season, when either the PV/T´s loop pump or the heat pump´s loop pump is operating. The flow-rate is set equal to 220 kg/h. An additional condition has been included: average water storage tank temperature has to be lower than 30°C. This requirement prevents the temperature in the tank from being too high and thus satisfying the limits for the radiant system supply temperature.

- In summer season, when the average water storage tank temperature is lower than 13°C. This condition avoids having too low temperatures in the flow feeding the embedded system. Flow rate is once more set as 220 kg/h.

Whenever the previously explained set of circumstances is satisfied, *PumpHEXhot* (and consequently *PumpHEXcold*) are activated, heating up the water in the storage tank. Pumps has a maximum power of 35 W (Grundfos UPS2 brochure, 2013).
Since DHW tank and water storage tank are connected only in certain conditions, whenever the conditions are not fulfilled, DHW tank outlet flux "to heat source" is made flow through the PV/Ts.
DHW auxiliary heater operation is controlled by the temperature inside the tank but also by an external component: *type 516, Hourly Forcing Function Scheduler, Weekdays, Saturday and Sundays Separate* (TESS libraries). This element forces the auxiliary heater to be operated among a predefined time interval (5 AM to 10 PM), which corresponds to a "highly-expectable" using-hours range.

7.3.2.5 Water storage tank

Water storage tank is simulated through component *type 4e, Storage tank; user-designated inlets, uniform losses*.

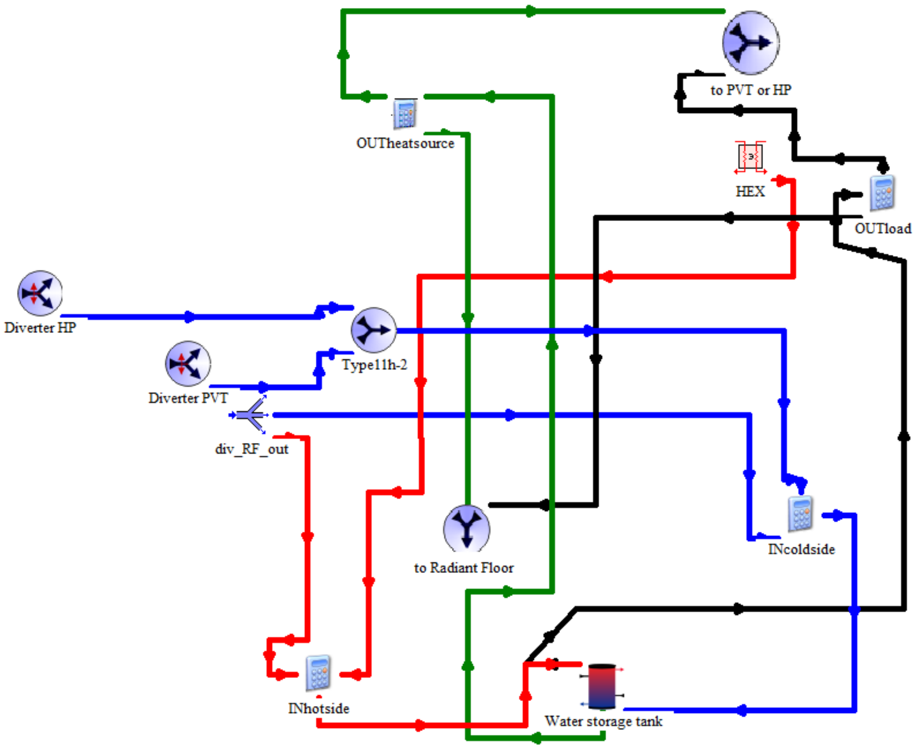


Figure 43: Visualization of water storage tank hydronic loop. Red arrows indicate "hot-side" inlets; blue arrows indicate "cold-side" inlets; green arrows indicate "to heat-source" outlet; black arrows indicate "to load" outlet. Source: TRNSYS Simulation Studio

Tank volume is defined as 800 litres (0.8 m^3); being water the contained liquid, fluid density and specific heat are 1000 kg/m^3 and 4190 J/kg·K respectively. Regarding tank-loss coefficient, default value of 0.83 $W/m^2 \cdot K$ is used. 50 nodes are defined, each of which is 2.5 cm (0.025 m) high, meaning that the overall height of the tank is 1.25 m. No auxiliary heating elements are included. Hot-source flow entering node is number 5 while cold-source flow entering node is 20th. Technical room temperature is considered as environment temperature of the surroundings. Inputs and outputs can be summarized as follows:

- Inlets:
  - Hot-side inlet: in winter-time, when PumpHEXhot is operating in summer-time the hot side inlet flux derives from the heat exchanger load-side´s outlet. In summer season it consists of the return flux from the embedded system.
  - Cold-side inlet: in winter season it consists of the return flow from the radiant floors, while in summer-time it is the cooled flux coming from the heat pump (during daytime) or the PV/Ts (during night-time).
- Outlets:
  - Outlet to heat-source: coming from the water storage tank, this flux is directed towards the PV/Ts or the heat pump in winter-time to be heated; in summer-time it is connected to the radiant system, where it absorbs heat while keeping the house cooled.
  - Outlet to load: the flux is directed towards the embedded system in winter season, where it is supposed to release heat; in summer-time it is connected to the PV/Ts or the heat pump to be cooled.

The 800 liters-water storage tank is meant to be a buffer component, allowing “heat sources” and “heat loads” to be decoupled. This idea is assumed to be beneficial for the whole system because it permits the control of their temperatures and flow rates separately.

7.3.2.6 Radiant floor system

Following figure represents the hydronic loop of the embedded system.

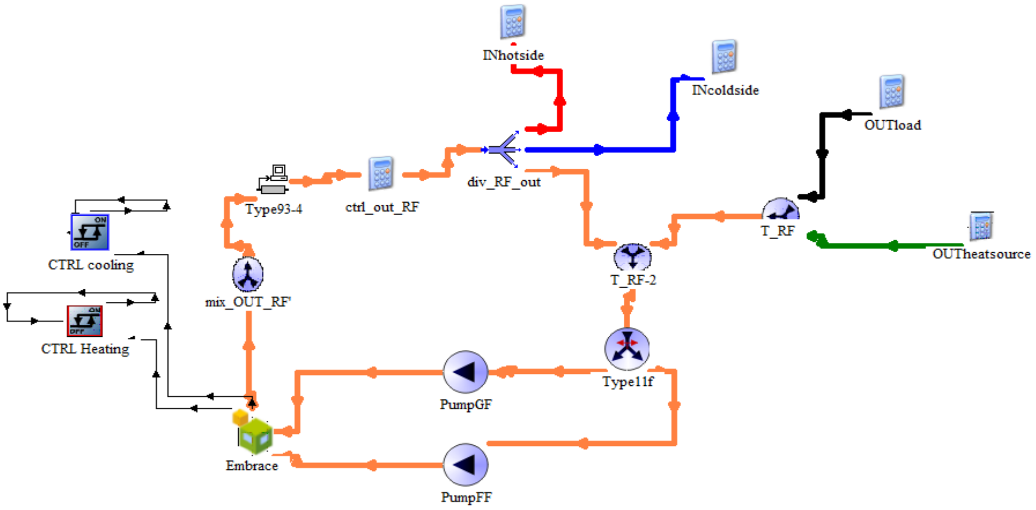


Figure 44: Visualization of the radiant floor hydronic loop. Orange arrows indicate the embedded system; for other arrows see Figure 43 for reference. Source: TRNSYS Simulation studio

Radiant system is embedded in both floors of *Embrace* and it is fed with water coming from the storage tank. A T-piece named *T_RF* collects the outlet flow of the tank, either “to heat-source” or “to load” depending on the season, and directs the water to a pump called *Pump_RF* (modelled as *type 3d, single speed pump* with maximum flow rate of 320 kg/h; default values are used for the other parameters). The pump is controlled by an external equation named *ctrl_RF_pump*, whose main output is the control signal that drives pump´s operation. Since pump´s activation is strictly connected to the indoor temperature, two Aquastat controllers (*type 2, Aquastat: heating/cooling mode*) have been added. Their features are shown below:

- Aquastat: heating control
  - *Temperature to watch*: *Embrace*´s ground floor operative temperature (which is supposed to be slightly lower than first floor operative temperature in the heating season).
  - *Set-point temperature:* 22.5°C, in order to satisfy EN 15251 Category I requirements for residential buildings (table A.3 – residential buildings, sedentary activity of 1.2 met) in heating conditions.
  - *Turn-on/off temperature difference:* a dead-band of ±0.5°C is considered.
  - Default values have been considered for the other parameters.

- Aquastat: cooling control
    - *Temperature to watch*: *Embrace*´s first floor operative temperature (which is supposed to be slightly higher than ground floor operative temperature in cooling season).
    - *Set-point temperature*: 24.5°C in order to meet EN 15251 Category I limitations in cooling season (table A.3 – residential buildings, sedentary activity of 1.2 met).
    - *Turn-on/off temperature difference*: a dead-band of ±0.5°C is considered.
    - Default values have been considered for remaining parameters.

As a result, the component *Pump_RF* is indoor temperature-driven according to the following strategy:

- Winter season:
    - Pump is activated when indoor air temperature falls below the given set-point Aquastat heating control is activated (which means that indoor operative temperature has fallen below controller´s set-point, taking into account the dead-band too). The flow rate in this case is 220 kg/h.

- Summer season:
    - During day-time, pump is operating when Aquastat cooling control is activated (which means that the indoor operative temperature has raised above cooling controller´s set-point, taking into account the dead-band too). Flow rate is set as 320 kg/h.
    - During night-time, pump is operating when Aquastat cooling control is activated. Flow rate in this case is 260 kg/h, the same circulating in the PV/Ts loop as mentioned at subchapter 7.3.2.1.

The flow rate is split in two fluxes, one feeding ground floor´s system and the other one feeding first floor´s system. Although it is a rough approximation, the subdivision is directly proportional to the covered area, which doesn´t include the first floor area above the technical room, as stated in subchapter 7.2.1. The pump feeding the ground floor has a maximum power of 32 W, while the one feeding the first floor has a maximum power of 25 W (Grundfos UPS2 brochure, 2013).
After circulating through *Embrace*´s floors, flows coming from both of the floors are mixed in a T-piece element called *mix_OUT_RF*, resulting in an unique flow.
Since it could occur during the cooling season that the radiant floor supply temperature is too low (that could result in surface condensation but also conditions that do not meet the ISO 7730 requirements), since surface temperature is limited down to 19°C according to (ISO 7730, Table A.3, category A), radiant floors have been equipped with a recirculation connection between system´s return and supply fluxes.

Outlet flux recirculated percentage is calculated as shown below:

$$\%_{recirculated} = \frac{T_{setpoint,summer} - T_{inlet,RadiantFloor}}{T_{outlet,RadiantFloor} - T_{setpoint,summer}}$$

$T_{setpoint,summer}$ for the radiant floors is 14°C. Recirculated fraction is limited to 50%; maximum fraction is assumed based on design assumptions. The non-recirculated percentage flows towards the water storage tank, in particular either to the "hot-side" inlet or the "cold-side" inlet depending on the season, as mentioned in subchapter 7.3.2.5.

7.3.2.7 Ventilation system

Ventilation system is composed of a heating and humidifier coil (*type 754, Simple Heating and Humidifying System; Temperature Controlled – TESS libraries*), a cooling coil (*type 752, Simple Cooling Coil Using Bypass Fraction Approach; Temperature Controlled*), an exhaust air heat recovery system (*type 760, Sensible Air to Air Heat Recovery with Controlled Outlet Conditions*), controller equations called *Ventilation Seasonal* and *Ventilation In*. The sketch is shown is the following figure.

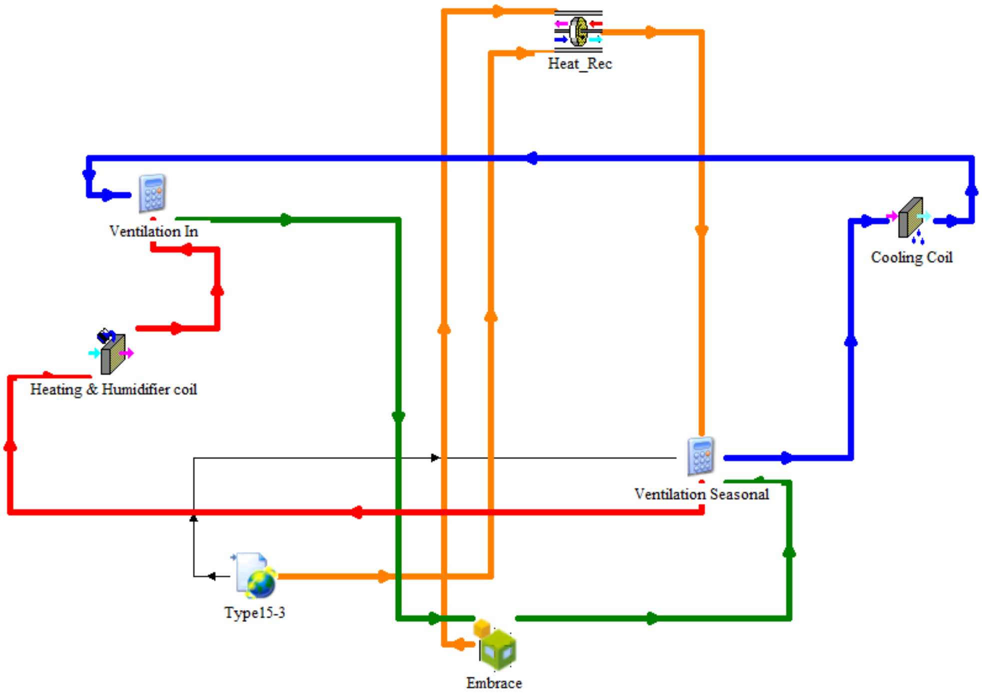


Figure 45: Visualization of the ventilation system. Red arrows indicate winter operation mode; blue arrows indicate summer operation mode; green arrows are common for both of the season; orange arrows indicate the heat recovery flows. Source: TRNSYS Simulation Studio.

In *Ventilation Seasonal* equation set, seasonal temperature and relative humidity are calculated in order to be input values of the heating and humidifier coil or the cooling coil. The control strategy is the following:

- Winter season: heat recovery system's outlet temperature and relative humidity are used as input for the heating and humidifying coil.
- Summer season: temperature and relative humidity to feed the cooling coil are:
    - Heat recovery system's outlet values if its temperature is lower than outside temperature
    - if not Outdoor values.

The *Heating and Humidifier coil* is operated in winter season, when supply clean-air needs to be heated and humidified before entering the conditioned space. Inlet temperature and relative humidity are given from *Ventilation Seasonal* component, while air mass flow rate is set as 195 kg/h, being the design hourly air-change value (ACH) equal to 1.25. Outlet air set-point temperature is set equal to 17°C, thus allowing a 4÷5°C temperature gap to let fresh air mix with indoor air (mixing ventilation). Default values are used for the other parameters.

The *Cooling coil* is operated in summer season. Its temperature and relative humidity input values come from *Ventilation Seasonal* element as well. Air flow rate is set to be 195 kg/h while the outlet air set-point temperature is 22°C, leaving a 2÷3°C temperature difference between supply and indoor air temperature, thus allowing mixing process. Default values are used for the other parameters.

Outlet values of both of the coils pass through the equation component *Ventilation in*. The purpose of this element is to feed the ventilation system of the dwelling with air coming from the heating and humidifying coil in winter-time and with air coming from the cooling coil in summer-time.

Embrace's exhaust air temperature and relative humidity become input values for the heat recovery device. Heat is transferred from exhaust air flux to supply clean-air flux (or vice versa, depending on the season). Flow rates of both of the fluxes are 195 kg/h. Sensible heat recovery effectiveness is assumed to be 0.85. Default values are used for the remaining parameters.

7.4 TRNSYS Simulation without storage tank

The system described in chapter 7.3 contains two different sources of thermal storage capacity; the water storage tank and the house itself can be considered as heat sinks which can act as buffers for the system. Therefore, simulating the system explained so far with respect to different buildings of increased thermal mass could lead to ambiguous or partially incorrect conclusions, since the dwelling and the water tank could partially shadow the effect of thermal mass of each other. Starting from this observation the TRNSYS model has been modified, avoiding the presence of the water storage tank and relying on the only thermal mass provided by the building. The basic idea is therefore to isolate the effect of the structure with respect to the system's performances and in particular to the night radiative cooling effect.

Most of the previous model has been left unchanged, however the PV/Ts and the heat pump need to be connected to the house either directly or through the DHW tank. The control strategy is required to be different and it is modeled according to the following basic concepts:

- In winter season:
    - If the PV/Ts outlet temperature is within the range 30÷37°C, the PV/Ts outlet flow is sent directly through the radiant system of the house, while the DHW tank is fed, if necessary, by the heat pump, which is operating in heating mode. The lowest limit of the range derives from an assumption based on the design supply temperature for the embedded system and the upper limit is set as 37°C to avoid surface temperatures higher than 29°C (thus satisfying Category A requirements for surface temperatures – ISO 7730, 2006, Table A.3);
    - In case the PV/Ts outlet temperature is higher than 55°C, the flow feeds the DHW first. The DHW tank outlet flux ("to heat-source") is then sent to the radiant system assuming that the water temperature from the DHW is appropriate to be fed into the radiant system. Finally, the return flow from the embedded pipes is sent either to the PV/Ts or the heat pump operating in heating mode);
    - In any other case the heat pump is activated, provided the necessary hot water stream to feed the DHW first and then the radiant system.

- In summer season:
    - During day-time, the heat pump (operating in cooling mode) provides the necessary cooling energy to the house in case the indoor temperature exceeds the set-point settings. At the same time PV/Ts are requested to feed to DHW tank with hot water. If the minimum requirements in terms of minimum deliverable temperature are not fulfilled, the DHW tank can rely on its auxiliary heater to maintain a satisfying temperature "to load" (i.e. the temperature of the hot water, mixed with cold water, which feeds the tapping appliances).
    - During night-time, PV/Ts are obviously not able to provide any hot water stream and the heat pump is deactivated. Since it is assumed the absence of any tappings, the DHW tank is able to maintain temperatures high enough. On the other hand, the house could present cooling needs: the necessary cold water flux is provided by the night-radiative cooling effect in the PV/Ts, achieved making the

embedded system´s return water circulate in the collectors during the night to be cooled down. The water then is sent to the building again, where it is able to extract heat. This strategy is activated as soon as the indoor temperature rises over the set-point settings.

Since it is a "free-source" of heating, priority is given to the PV/Ts but obviously a heat pump is necessary to cover the periods when the PV/Ts minimum outlet temperature requirements are not satisfied.

Set-point temperatures for indoor temperature in heating and cooling seasons are kept the same, being 22.5°C for the heating season and 24.5°C for the cooling season, in order to meet the EN 15251 (2007) – Annex A, Category I requirements for residential buildings and sedentary activity (1.2 met). Moreover, control strategy requests the embedded systems´ supply temperature to be within the range 30°C÷37°C during the heating season and within the range 13°C÷18°C in the cooling season.

Pumps are activated with the same flow-rates explained in subchapter 7.3.2. PV/T´s pump is implemented as a single speed pump (*Type 3d, Single speed pump*) with a maximum flow-rate of 320kg/h (although the flow-rates are 220 kg/h in heating season, 320 kg/h and 260 kg/h in cooling season during day-time and night-time, respectively) and a corresponding maximum power of 35 W. Heat pump´s hydronic pump has identical characteristics. The total flow-rate is split between ground floor flow-rate and first floor flow-rate, according to the active floor surface. Ground floor embedded pipes are fed through a single speed pump, with maximum flow-rate of 275.2 kg/h (the actual flow-rates are 189.2 kg/h in heating season, 275.2 and 223.6 kg/h in cooling season, during day-time and night-time respectively) and a corresponding maximum power of 32 W. The flow through the intermediate floor radiant system is ensured by a single speed pump with maximum flow-rate of 44.8 kg/h (the actual flow-rates are 30.8 kg/h in heating season, 44.8 and 36.4 kg/h in cooling season, during day-time and night-time respectively) and a corresponding maximum power of 25 W. Grundfos UPS2 pump series has been chosen as the reference (Grundfos UPS2 brochure, 2013).

DHW tank and DHW hydronic system are identical to those described in subchapter 7.3.2.4. An hourly forcing function scheduler has been included as in the case with the water storage tank to avoid the activation of the auxiliary heater during night-time and allowing it to be activated from 5 AM (therefore two hours before the first supposed tapping draw) to 10 PM.

Ventilation is exactly identical to the one described in subchapter 7.3.2.7.

8. THERMAL MASS CALCULATION

8.1 Thermal mass – Embrace – Structure 0

Since the main purpose of this report is to evaluate how thermal mass affects system´s performances and to see the behavior of different systems with different thermal masses when coupled to PV/T system, *Embrace*´s thermal mass has been calculated in order to have a reference case which all further considerations will be referred to. The reference for calculation procedure is German Standard (VDI 2078 - Cooling load calculations of air-conditioned rooms, 1996) in order to permit the structure to be classified according to the subdivision presented in VDI 2078 – Section 5.5. Standard VDI 2078 (1996) refers to office buildings and is based mostly on empirical thermal storage/mass data of German buildings, however it represents an useful standard classification and thus it has been considered in this report.

Table 28: Building classification according to its effective thermal mass. *VDI 2078, Section 5.5*

Classification	Effective Heat Capacity$_{\text{reference value}}$ [$Wh/m^2 \cdot K$]	Classification range [$Wh/m^2 \cdot K$]
Very Light (XL)	5	Thermal mass ≤ 50
Light (L)	68	50 < Thermal mass ≤ 100
Medium (M)	123	100 < Thermal mass ≤ 200
Heavy (S)	248	Thermal mass > 200

"Effective" heat capacity calculation differs from the mere sum of the heat capacity of each wall, each of which calculated adding each layer´s thermal capacity contribution. "Effective" heat capacity calculation procedure takes into account the presence of insulation layer within the structures. Therefore, according to VDI 2078 Section 5.5, the effective thickness of the layer has to be calculated as follows:

- Insulating layers are characterized by thermal conductivity $\lambda < 0.1$ W/m·K and thermal resistance $R_{th} > 0.15$ $m^2 \cdot K/W$.
- Layers located behind thermal insulation has to be considered as follows:
  - In case the overall insulating thermal resistance is $0.15 < R_{th} \leq 0.30$ [$m^2 \cdot K/W$], they have to be considered with respect to half of their mass
  - In case the overall insulating thermal resistance is $R_{th} > 0.30$ they are excluded from the calculation.

- Regarding internal walls (which face the conditioned environment on both sides), only half of the mass has to be taken into account.

It has been decided to follow VDI 2078 instead of other references (i.e. ISO 13790) because it suits better for the purposes of the studies. In fact, preliminary calculations were performed according to ISO 13790 but, due to *Embrace´s* small floor area and other design characteristics, the results made the building fit between in the ISO 13790 heavy-weight category. Therefore it would have been difficult to highlight differences between the thermal capacities of the other structures involved. Thus, it was decided to follow VDI 2078 indications, despite it was published few years in advance than ISO 13790.
Following VDI 2078 directions Embrace´s thermal mass is evaluated. The result is 85.45 $Wh/m^2 \cdot K$ (detailed calculation can be found in Appendix). The result makes *Embrace* fit in the light-weight range. The overall result is referred to floor surface area.

Standard VDI 2078 (1996) provides an additional criterion to classify the building structure, based on the ratio between overall building mass and floor area. The subdivision is made according to the values presented in Table 29.

Table 29: Overall mass/Floor area ratio. Source: VDI 2078 (1996) – Section 5.5

Classification	$\Sigma m_{TOT}/A_F$ [kg/m^2]	Classification range [kg/m^2]
Very Light (XL)	27	Ratio ≤ 150
Light (L)	236	150 < Ratio ≤ 300
Medium (M)	463	300 < Ratio ≤ 800
Heavy (S)	859	Ratio > 800

The ratio for Embrace results 107.2 kg/m^2, making it fit in the light-weight category.
In order to evaluate the effect of increased thermal mass on system´s performances, three different structures have been hypothesized. A fully-identical geometry is considered, as well as the same floors structures, in order to fix geometrical parameters and active surfaces. Therefore only the passive surfaces are changed. The basic idea is to isolate the influence of the thermal mass, maintaining the same heat output and thus guaranteeing identical “boundary” conditions for every considered case.

Another German standard, VDI 2078-1 (2003), is connected to the one considered. Despite some similarities they present differences. Firstly, different range subdivisions for the thermal capacity are suggested. Secondly, the calculation method is different, since in VDI 2078-1 a maximum implementable thickness of

0.10 m is fixed. Finally, no classification based on the ratio overall mass/floor area is provided. The differences mentioned above could be due to the fact that VDI 2078-1 was published a few years later than the VDI 2078. All the calculations and the considerations presented in this report refer to VDI 2078.

Three different types of concrete are considered, which could be categorized as "light-weight", "medium-weight" and "heavy-weight". Table 30 shows their main features considered for calculations:

Table 30: Different concrete types´ main features

	Thermal conductivity λ [W/m·K]	Specific heat capacity (HC) [kJ/kg·K]	Density [kg/m^3]
"Light-weight" concrete	0.25	0.83	1280
"Medium-weight" concrete	0.7	0.88	1760
"Heavy-weight" concrete	1.6	0.92	2080

For the sake of ease, a simple concrete structure had been considered, consisting of (from the layer facing the conditioned room to the one facing outside – (Introduction to Building Physics, 2003)):

- Interior plaster finishing
- Concrete
- Glass fiber insulation layer
- Exterior plaster finishing

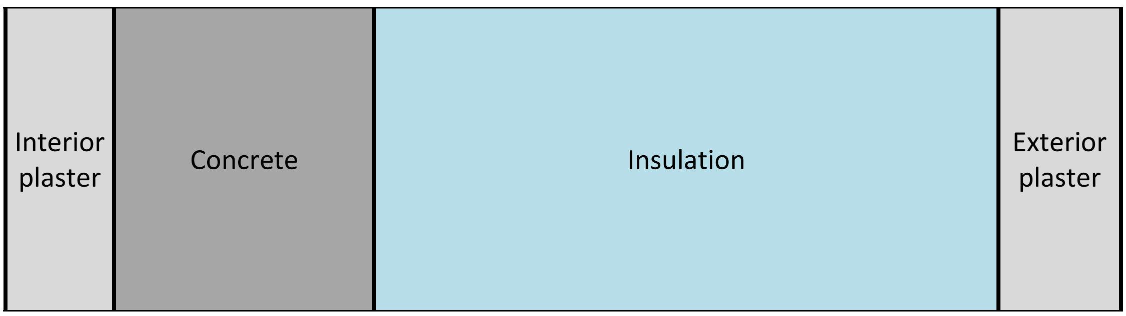


Figure 46: Concrete wall structure

Materials' general properties are presented in Table 31.

Table 31: Concrete structure materials´ properties

	Thermal conductivity λ [W/m·K]	Specific heat capacity [kJ/kg·K]	Density [kg/m^3]
Interior plaster	0.7	1.1	1500
Glass fiber insulation	0.032	35	35
Exterior plaster	0.7	1.1	1500

Concrete properties and thickness differ from case to case, as mentioned above.

8.2 Thermal mass – Structure 1

Structure 1 calculations involve "light-weight" concrete within the structure presented in Figure 46. Wooden materials have been substituted in external and internal walls but floor structures have not been modified. Therefore, *Embrace*´s walls result as shown in Table 32, Table 33 and Table 34.

Table 32: External wall – "Light-weight" concrete

		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	0.25	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.83	0.9	1.1
Density	[kg/m^3]	1500	1280	35	1500
Thickness	[m]	0.015	0.1	0.3	0.015

Table 33: Internal wall, non-insulated – "Light-weight" concrete

		INT PLASTER	LIGHT CONCRETE	EXT PLASTER
λ	[W/m·K]	0.7	0.3	0.7
Specific HC	[kJ/kg·K]	1.1	0.83	1.1
Density	[kg/m^3]	1500	1280	1500
Thickness	[m]	0.015	0.05	0.015

Table 34: Internal wall, insulated – "Light-weight" concrete

		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	0.3	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.83	0.9	1.1
Density	[kg/m^3]	1500	1280	35	1500
Thickness	[m]	0.015	0.05	0.2	0.015

Thermal mass has been calculated according to VDI 2078.
The result is 171 Wh/m^2·K, which makes structure 1 belong to the medium-weight category. The ratio of total-mass to floor-area is 562 kg/m^2, belonging to the medium-weight category for thermal mass.

8.3 Thermal mass – Structure 2

"Medium-weight" concrete is used in structure 2 to substitute the wooden materials among external and internal walls. Floor structures are fully-identical to the original case. Walls´ main features are shown in Table 35, Table 36 and Table 37.

Table 35: External wall - "Medium-weight" concrete

		INT PLASTER	MEDIUM CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	0.7	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.88	0.9	1.1
Density	[kg/m^3]	1500	1760	35	1500
Thickness	[m]	0.015	0.12	0.3	0.015

Table 36: External wall, non-insulated – "Medium-weight" concrete

		INT PLASTER	MEDIUM CONCRETE	EXT PLASTER
λ	[W/m·K]	0.7	0.7	0.7
Specific HC	[kJ/kg·K]	1.1	0.88	1.1
Density	[kg/m^3]	1500	1760	1500
Thickness	[m]	0.015	0.06	0.015

Table 37: Internal wall, insulated – "Medium-weight" concrete

		INT PLASTER	MEDIUM CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	0.7	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.88	0.9	1.1
Density	[kg/m^3]	1500	1760	35	1500
Thickness	[m]	0.015	0.06	0.2	0.015

Calculation has been performed according to VDI 2078. Thermal mass is calculated to be 238 Wh/m^2·K, fitting in VDI 2078 heavy-weight for thermal capacity. The ratio between overall mass and floor area results in 842 kg/m^2, which means that structure 2 meets the heavy-weight category requirements for the ratio mass/floor-area.

8.4 Thermal mass – structure 3

Structure 3 calculations involve "heavy-weight" concrete within the structures. Wooden materials have been substituted in external and internal walls but floor structures have not been modified as well as in structure 1 and structure 2. Therefore, *Embrace*´s walls result as shown in Table 38, Table 39 and Table 40.

Table 38: External wall – "Heavy-weight" concrete

		INT PLASTER	HEAVY CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	1.6	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.92	0.9	1.1
Density	[kg/m^3]	1500	2082	35	1500
Thickness	[m]	0.015	0.18	0.3	0.015

Table 39: Internal wall, non-insulated – "Heavy-weight" concrete

		INT PLASTER	HEAVY CONCRETE	EXT PLASTER
λ	[W/m·K]	0.7	1.6	0.7
Specific HC	[kJ/kg·K]	1.1	0.92	1.1
Density	[kg/m^3]	1500	2082	1500
Thickness	[m]	0.015	0.085	0.015

Table 40: Internal wall, insulated – "Heavy-weight" concrete

		INT PLASTER	HEAVY CONCRETE	GLASS FIBER	EXT PLASTER
λ	[W/m·K]	0.7	1.6	0.032	0.7
Specific HC	[kJ/kg·K]	1.1	0.92	0.9	1.1
Density	[kg/m^3]	1500	2082	35	1500
Thickness	[m]	0.015	0.085	0.2	0.015

Structure 3 thermal mass results 318 Wh/m^2·K, therefore it fits in VDI 2078 heavy-weight category for thermal capacity. The ratio between building mass and floor area is 1183 kg/m^2, thus it fits in the heavy-weight category for the ratio as well.

9. RESULTS

Once the TRNSYS models have been completed, simulations have been carried out, taking into account the four different structures defined in chapter 8 and its following subchapters, for both Copenhagen climate conditions and Paris climate conditions. In order to have more comparable results, the season-schedule settings presented in subchapter 7.3.1 (winter season: 1st January – 20th May and 11th September – 31st December; summer season: 20th May – 10th September) has been kept also for simulations involving Paris weather file. This assumption is based on the goal of having more comparable results, despite the season-schedule is supposed to be climate-dependent (therefore summer season should be extended in simulations involving Paris climate) and despite indications about the winter and summer season periods in papers, among which (Olesen, Sommer, & Duchting, 2002).

Set-points temperatures, flow-rates and general input values are identical in every case considered.

The comparison of the results will be based on the performances in terms of comfort classes, the operational hours of the devices included in the system, their energy consumption, efficiencies and productions. Particular attention will be given to the implementation of the nigh radiative cooling and its exploitation. Results of cases with different thermal capacity will be compared, as well as results of the cases including the water storage tank or not. Finally outcomes will be used to compare the energy performances in Copenhagen climate and Paris climate.

Due to the number of combinations involving two different climates, for the sake of ease and simplicity, the following abbreviations are introduced in the presentation of the results.

Table 41: Abbreviations used for the presentation of the results – Copenhagen climate

	Water storage tank present	Water storage tank not present
Structure 0	T_0_CPH	noT_0_CPH
Structure 1	T_1_CPH	noT_1_CPH
Structure 2	T_2_CPH	noT_2_CPH
Structure 3	T_3_CPH	noT_3_CPH

Table 42: Abbreviations used for the presentation of the results – Paris climate

	Water storage tank present	Water storage tank not present
Structure 0	T_0_PAR	noT_0_PAR
Structure 1	T_1_PAR	noT_1_PAR
Structure 2	T_2_PAR	noT_2_PAR
Structure 3	T_3_PAR	noT_3_PAR

9.1 Indoor temperature

9.1.1 EN 15251 Comfort categories

Indoor temperature evaluation is based on (EN15251, 2007). The temperature is classified according to this standard and its Category I and Category II for Residential buildings and sedentary activity (1.2 met). Figure 47, Figure 48, Figure 49and Figure 50 present the percentage (referred to the 8760 hours) of hours where the indoor operative temperature meets either Category I or Category II requirements. The considered ranges of temperatures are presented below.

- EN 15251 Category I
  - Winter season: 21°C ≤ Indoor temperature ≤ 25°C
  - Summer season: 23.5 ≤ Indoor Temperature ≤ 25.5 °C

- EN 15251 Category II
  - Winter season: 20°C ≤ Indoor Temperature ≤ 25°C
  - Summer season: 23°C ≤ Indoor Temperature ≤ 26°C

In the presentation of the results some abbreviations are used.

- GF stands for Ground Floor air-node
- FF stands for First Floor air-node

The “overall” percentages mentioned in the figures below are the result of the weighted average of ground floor´s contribution and first floor´s contribution. First, the yearly percentage of each air-node is calculated as result of the weighted average of winter and summer season (the weights are the numbers of hours each season contains). Then the overall value is calculated. The weights consist of the air volume of the air-node. The overall percentage within either EN 15251 Category I range or EN 15251 Category II range is taken as indicator for the further considerations about indoor conditions.

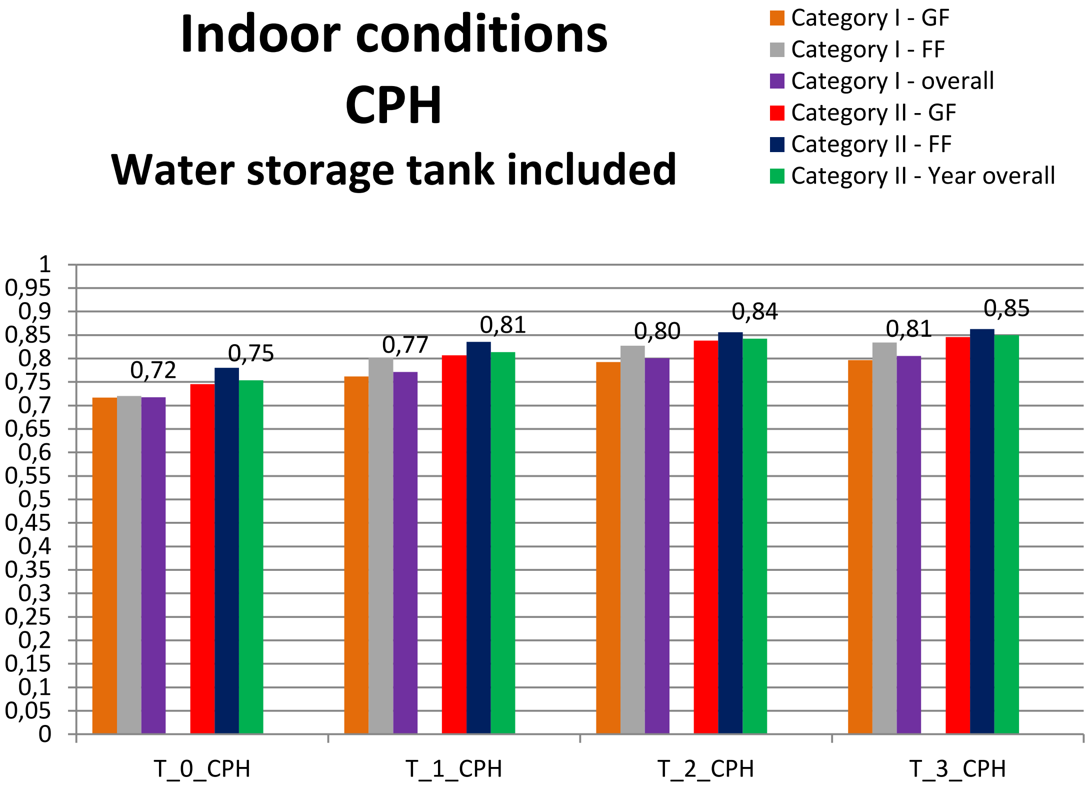


Figure 47: Indoor conditions – Copenhagen – Water storage tank included

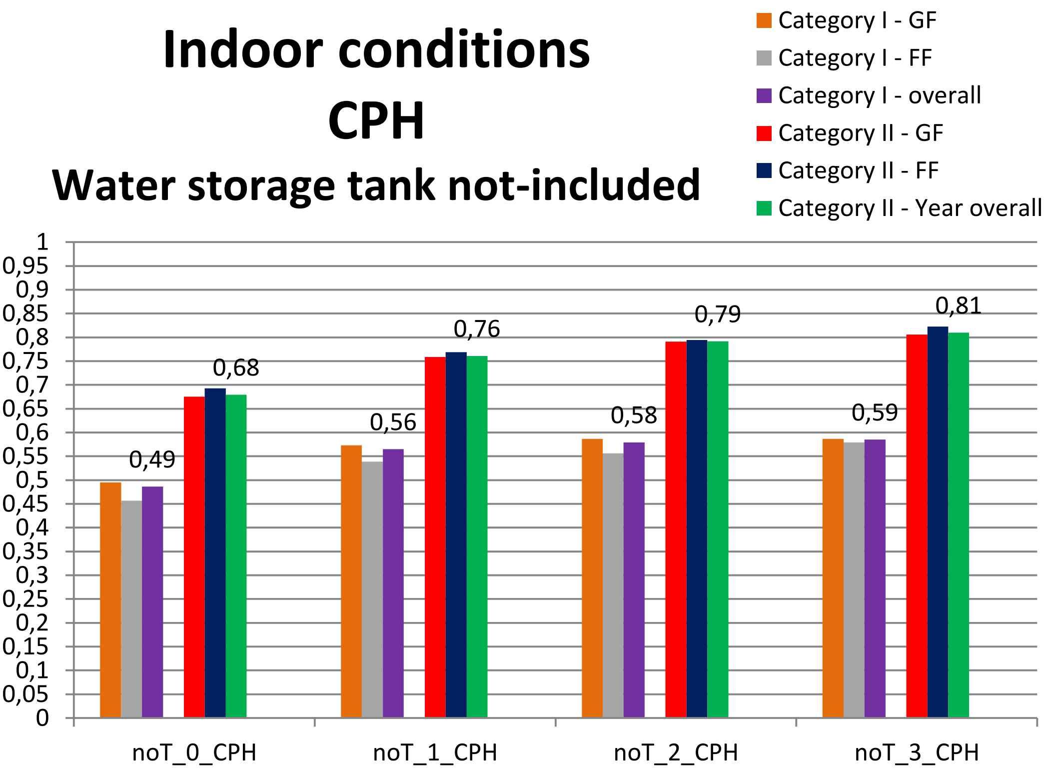


Figure 48: Indoor conditions – Copenhagen – Water storage tank not-included

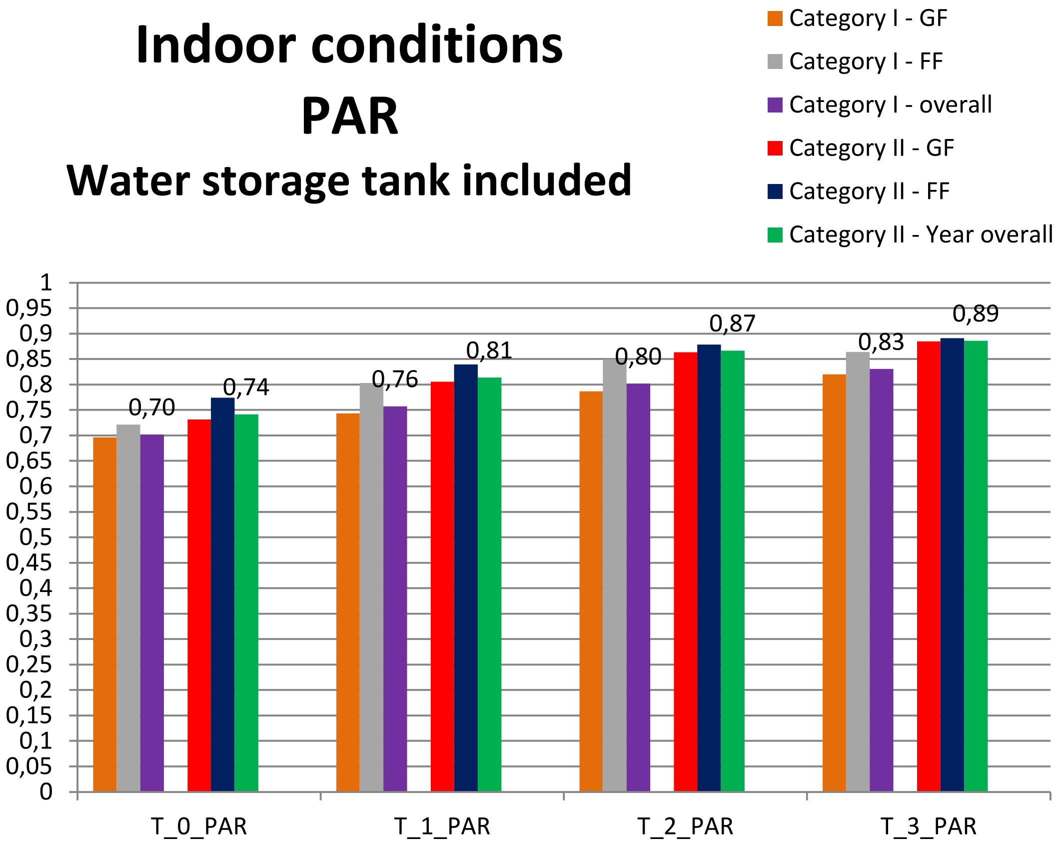


Figure 49: Indoor conditions – Paris – Water storage tank included

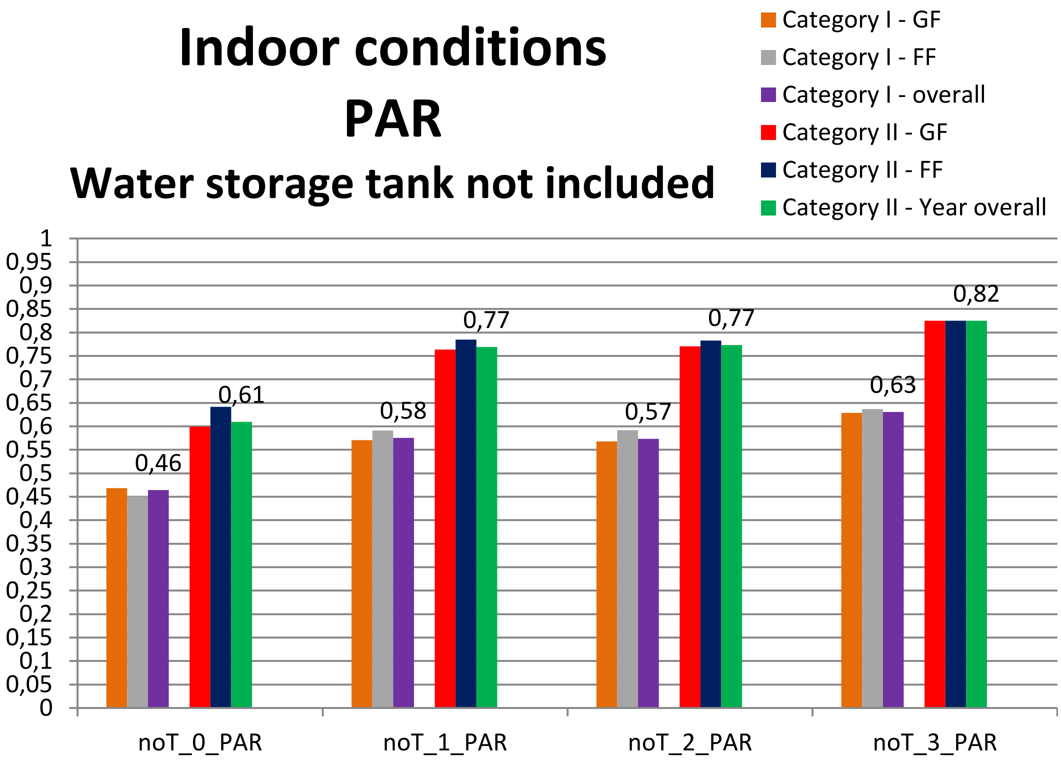


Figure 50: Indoor conditions –Paris – Water storage tank not-included

Data labels are referred to overall indoor temperature, being classified within Category I range (violet) and Category II range (green).
It´s observed that increased thermal mass results in higher percentages within the ranges. The presence of the water storage buffer tank is beneficial for the comfort conditions, since the decrease of the percentages between the cases including the tank and the cases not including the tank is not negligible.
Paris comfort conditions result to be higher than the Copenhagen´s case, but it should be kept in mind that the season schedule has been adjusted for Copenhagen and extended also to the simulations involving Paris climate.

9.1.2 *Over-heating*

Another parameter to highlight in indoor temperature evaluation is the number of hours in which the indoor temperature exceeds 26°C and 27°C. According to (DS 469, 2013), indoor temperature is not allowed to exceed 26°C for 100 hours and 27°C for 25 hours. Therefore, this parameter has been calculated for every simulation.

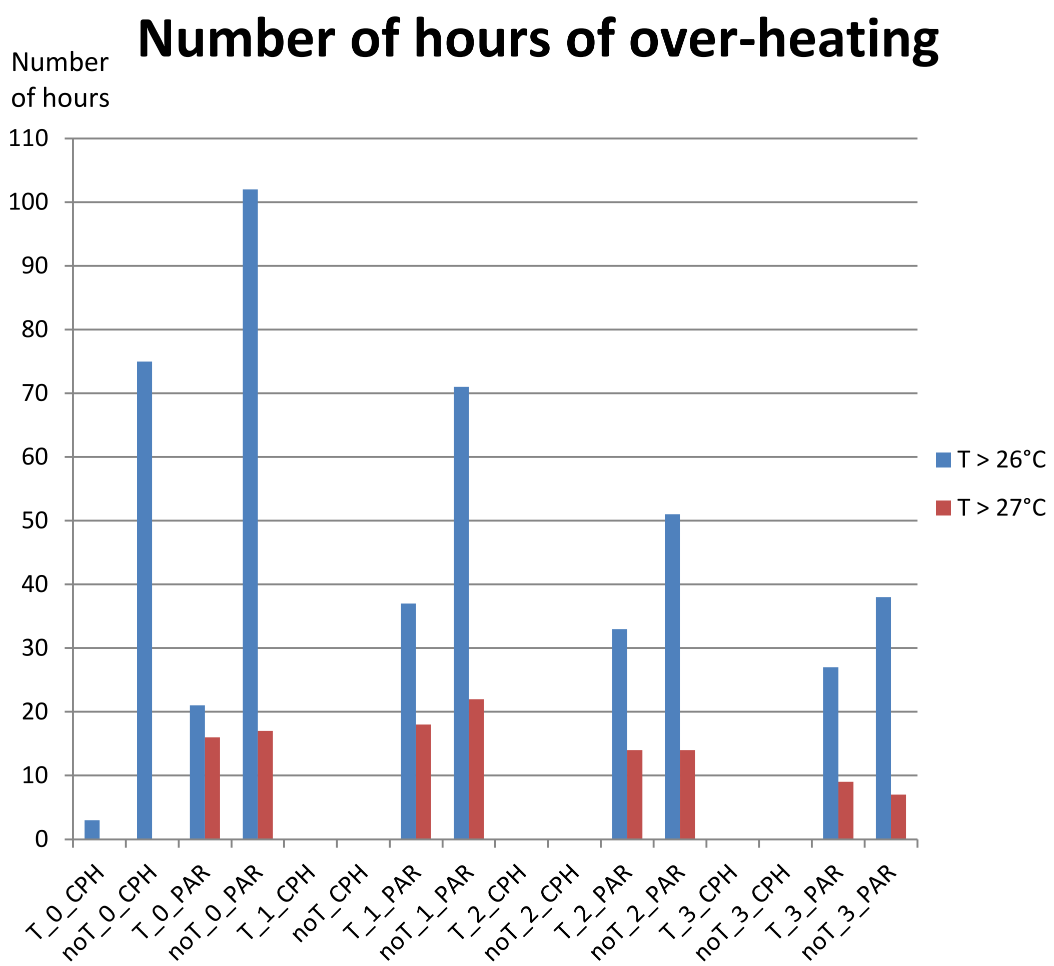


Figure 51: Number of hours in which the indoor temperature exceeds 26°C or 27°C

The results show that the number of hours in which the indoor temperature exceeds 26°C or 27°C decreases by increasing the thermal mass. Moreover it´s observed that values are higher for Paris rather than for Copenhagen, due to warmer weather conditions. Furthermore, the presence of the water storage tank is favorable, resulting in shorter periods in which the indoor temperature exceeds the fixed limits. The requirement regarding the maximum allowed amount of hours of over-temperature is generally satisfied in any condition. In the case named "noT_0_PAR" the amount of hours in which the temperature exceeds 26°C is 102, however it is considered acceptable as well.

9.1.3 Indoor average temperatures

In this subchapter, ground floor and first floor air-nodes average operative temperatures are listed in Table 43. Operative temperature is taken into account, recorded in TRNSYS among the outputs of the component implementing the building.

Table 43: Average operative temperatures for ground floor and first floor air-nodes

	Winter – GF [°C]	Winter – FF [°C]	Summer - GF[°C]	Summer - FF[°C]
T_0_CPH	22.9	22.7	22.0	22.5
noT_0_CPH	21.7	21.6	22.2	22.7
T_0_PAR	23.3	23.3	22.7	23.4
noT_0_PAR	22.2	22.4	22.2	23.6
T_1_CPH	22.8	22.6	22.4	22.8
noT_CPH	21.7	21.6	22.5	22.9
T_1_PAR	23.2	23.2	23.2	23.7
noT_1_PAR	22.2	22.3	23.4	23.9
T_2_CPH	22.6	22.5	22.6	23.0
noT_2_CPH	21.7	21.6	22.6	22.9
T_2_PAR	23.1	23.1	23.4	23.9
noT_2_PAR	22.1	22.1	23.6	24.0
T_3_CPH	22.6	22.4	22.7	23.1
noT_3_CPH	21.7	21.6	22.7	23.0
T_3_PAR	23.0	23.0	23.5	24.0
noT_3_PAR	22.1	22.1	23.5	24.0

It´s observed that with higher thermal mass indoor temperature generally slightly decreases in winter season and increases in summer season. This behavior can be explained considering the bigger thermal mass the building can rely on. Therefore, during winter-time structures with higher thermal capacity have the capability to absorb and store heat (due to internal gains and solar radiation if present), resulting in slightly lower operative temperature.
The presence of the tank influences significantly the average indoor temperature.
Obviously, temperatures are generally higher in Paris rather than in Copenhagen, due to a warmer climate.

Figures below show the influence of thermal mass has on the indoor operative temperature (average season value).

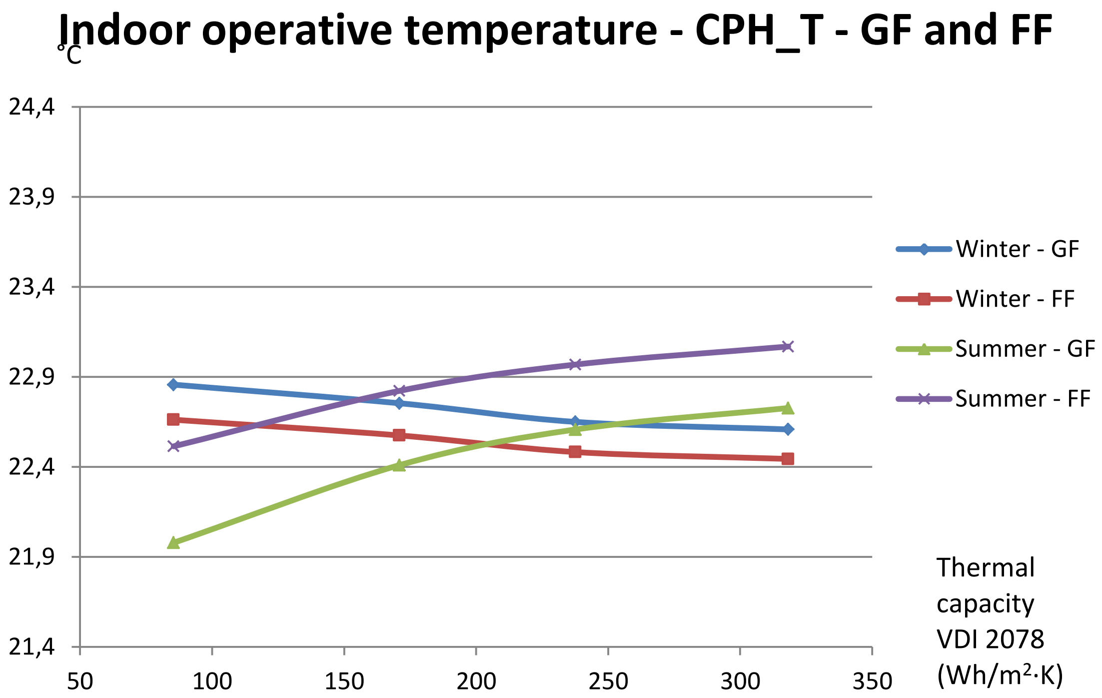


Figure 52: Thermal mass´ influence on indoor operative temperature (average season value) – Copenhagen – Water storage tank included

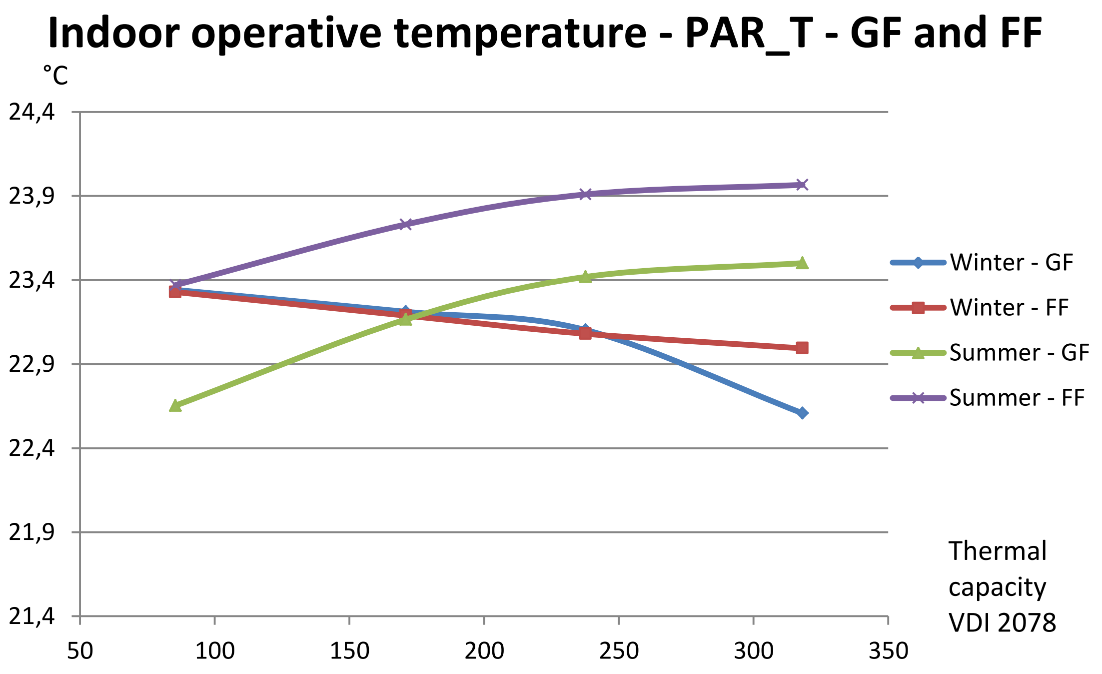


Figure 53: Thermal mass´ influence on indoor operative temperature (average season value) – Paris – Water tank included

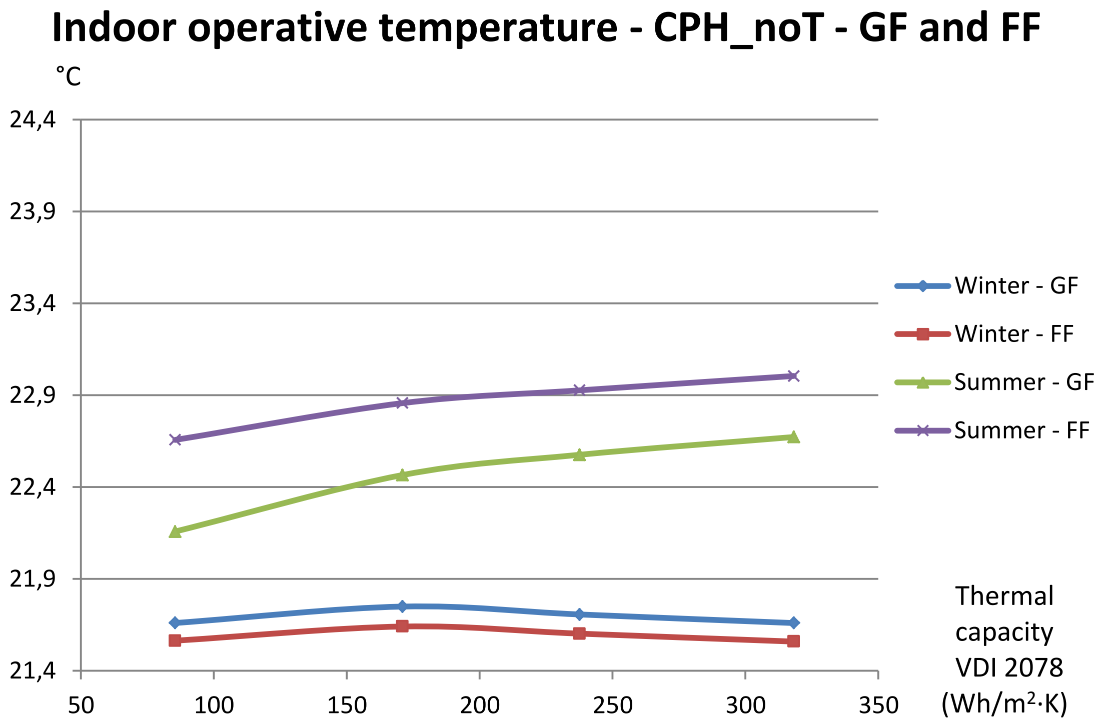


Figure 54: Thermal mass´ influence on indoor operative temperature (average season value) – Copenhagen – Water storage tank not included

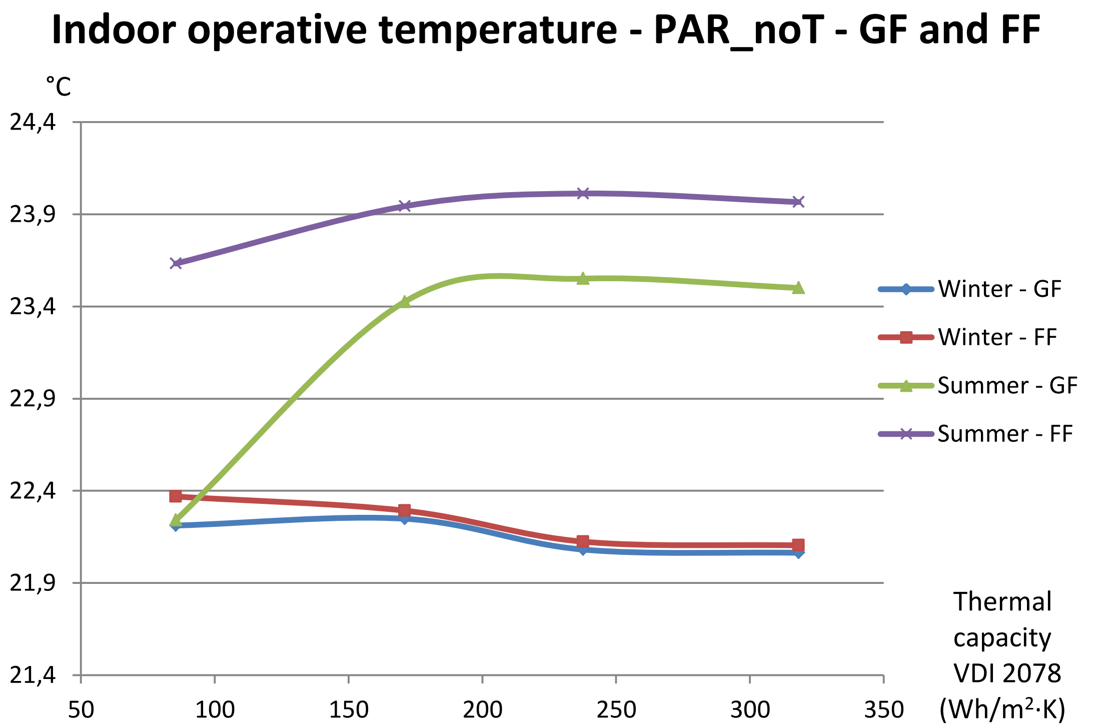


Figure 55: Thermal mass´ influence on indoor operative temperature (average season value) – Paris – Water storage tank not included

Figure 52, Figure 53, Figure 54 and Figure 55 show the influence thermal mass has with respect to the indoor temperature. A similar trend is observed in every case. Regarding indoor temperatures in winter time, it is noticed that temperature decreases in buildings with increased thermal capacity, while in summer time indoor temperature tends to increase. This is due to the capability of the structure to absorb heat, slightly lowering the indoor temperature in winter conditions. The summer behavior can be explained considering the release of the absorbed heat, which contributes to increase the indoor temperature. However, this behavior would deserve further studies.

Figures below highlight the differences in the indoor operative temperatures in winter season. The purpose is to underline the reduced amplitude of the indoor temperatures and activity time of the pumps. Cases “T_0_CPH” and “T_3_CPH” have been taken as examples. It is evident that temperature´s fluctuation is smoothed and delayed in case the building has higher thermal mass.

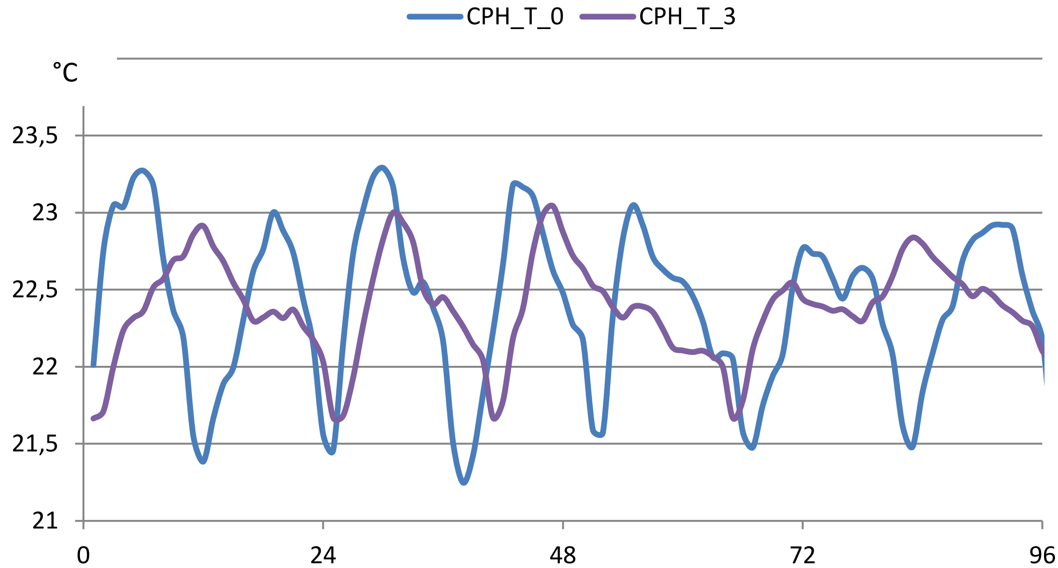


Figure 56: Indoor operative temperature for two structures, detail – winter-time (the first 100 hours of February have been taken into account)

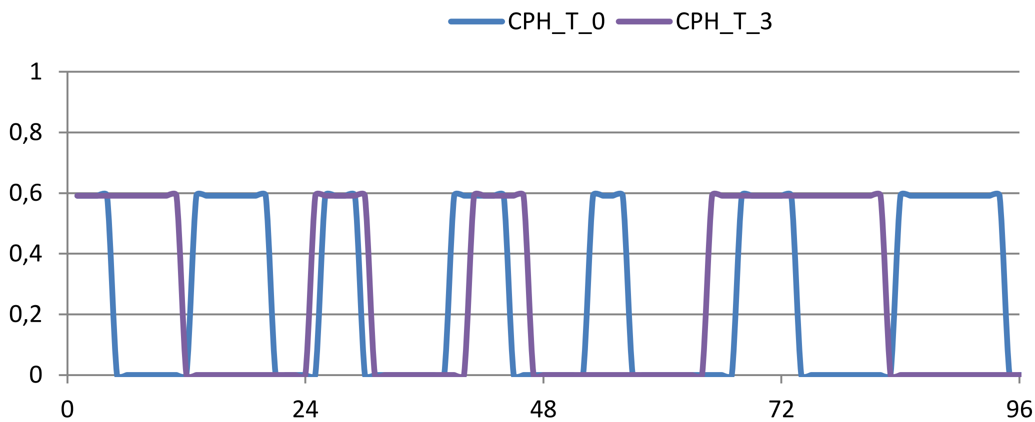


Figure 57: Periods of activation of the embedded system´s pump – winter time (the first 100 hours of February have been taken into account)

It is also observed that in case of higher thermal capacity the pump feeding the embedded system is activated less frequently. The decrease of the number of ON/OFF switches is thus beneficial from the consumptions point of view and to prevent damages in the pump.

Figures below allow extending these observations to the summer case. Simulations called “T_0_CPH” and “T_3_CPH” have been considered as examples.

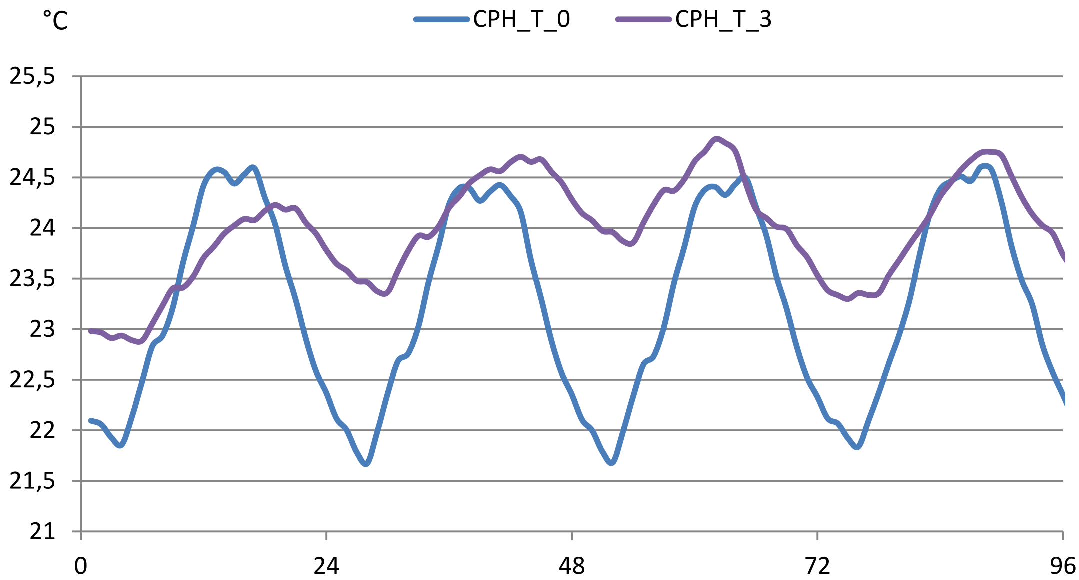


Figure 58: Indoor operative temperature for two structures, detail – Summer-time (the first 100 hours of July have been taken into account)

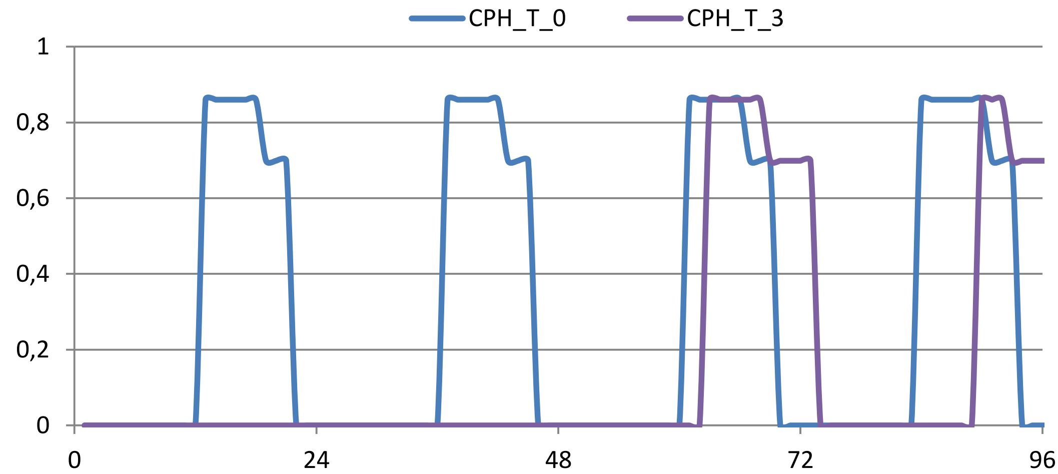


Figure 59: Periods of activation of the embedded system´s pump – Summer time (the first 100 hours of July have been taken into account)

Y-axis in figures above is control signal the pump operation is based on. It can be noticed that in winter-time it has a unique value, while in summer-time pump operation is divided between day-time operation and night-time operation.

9.2 Radiant floor supply temperature

Embedded system supply temperature has been evaluated for every considered case. Average supply temperature is presented, for both winter and summer seasons. Table 44 shows the results for Copenhagen climate, while Table 45 shows the results for Paris climate. Embedded system supply temperature derives from one of the output of the TRNSYS component implementing *Embrace*.

Table 44: Radiant floor supply temperatures for Copenhagen

		Tank included	Tank not included
Winter	0_CPH	38.2	35.7
	1_CPH	38.3	36.1
	2_CPH	38.6	36.3
	3_CPH	38.6	36.2
Summer	0_CPH	13.8	15.0
	1_CPH	14.1	13.7
	2_CPH	14.3	13.2
	3_CPH	14.2	13.3

Table 45: Radiant floor supply temperature for Paris

		Tank included	Tank not included
Winter	0_CPH	36.9	35.2
	1_CPH	36.8	35.0
	2_CPH	36.9	35.1
	3_CPH	36.9	35.2
Summer	0_CPH	14.3	15.9
	1_CPH	13.8	14.9
	2_CPH	14.3	14.4
	3_CPH	14.1	14.2

In the cases where the water storage tank is included, it is not always possible to notice a clear tendency in the supply temperatures, since it is generally fluctuating. This observation (among others) justifies the addition of the second model where the water storage tank is not included, in order to let the system rely only on the thermal mass provided by the building and to exploit the passive heating/cooling strategies more.

Results of the model not including the water storage tank are listed on the right columns of Table 44 and Table 45. In heating season, supply temperatures for Copenhagen increase with higher thermal mass, while in cooling season supply temperatures decrease with higher thermal mass. In fact, structures with higher thermal capacity are able to absorb and store higher quantities of heat (due to internal gains, since the effect of external weather on the thermal mass is almost completely negligible due to the presence of the insulation layer of the external side of the massive layer). Therefore structures are able to absorb more and in winter-time this results in "warmer" structures and subsequently in higher supply and return temperatures. Regarding cooling season, the decrease of supply temperatures with higher thermal mass is due to the same reason: supply temperatures are lower in order to remove and extract the heat stored, which amount increases with higher thermal capacities. Moreover, in the cases where the water storage tank is avoided, supply temperatures are higher for Paris climate than Copenhagen climate, due to longer activity periods of the pump feeding the embedded system.

"T" cases are now taken into account. During the heating season, the presence of the tank results in higher supply temperatures for both of the climates. In fact, the tank allows storing water which has higher temperatures, resulting in higher indoor temperatures and therefore higher percentages of hours within the comfort categories. The difference is more evident in Copenhagen climate. It is also observed that in cooling season, supply temperature tends to slightly increase (Copenhagen) or to stay constant (Paris) with higher thermal mass in the cases where the water storage tank is included. In "T" cases, supply temperature is approximately 14°C for both of the climates. The raise in terms of embedded system supply temperature is due to the fact that increased thermal mass results in increased possibility to store the heat which otherwise is supposed to be removed by the radiant system; therefore, since the heat is stored, the embedded system is in charge of extracting a lower amount of heat.

Table 46 and
Table 47 present the return temperatures for Copenhagen and Paris.

As previously mentioned, it is observed that during the heating season return temperatures are higher in case the water storage tank is included; moreover, structure with higher values of thermal mass correspond to slightly higher return temperatures (as well as supply temperatures). Regarding cooling season, the presence of the tank results in lower supply and return temperatures; since buildings with higher thermal capacity are able to absorb more heat, return temperature is observed to be higher in structure with increased thermal mass (coupled with lower supply temperatures).

Table 46 and Table 47 also present the calculated temperature drops (σ). Similar trends are observed in every case: the temperature drop is significantly higher in winter season and significantly lower in cooling season in the cases in which the water storage tank is not included. Furthermore, σ tends to increase with higher values of thermal mass. This can be explained referring to Table 43, Figure 52, Figure 53, Figure 54, Figure 55, Table 44 and Table 45. With respect to the values presented, it is observed that in both of seasons the radiant system is able to rely on temperature differences between the supply temperature and the indoor operative temperature that are higher in the heavier cases, meaning that the building corresponds to higher values of thermal mass. For this reason, the heat exchange relies on higher temperatures differences corresponding to higher temperature drops between supply and return water.

Table 46: Embedded system return temperatures for Copenhagen

		T_{return} GF	T_{return} FF	σ GF	σ FF
Winter	T_0_CPH	30.8	31.9	7.4	6.3
	T_1_CPH	30.8	32.0	7.5	6.3
	T_2_CPH	30.9	32.2	7.7	6.4
	T_3_CPH	30.9	32.2	7.6	6.3
	noT_0_CPH	26.9	28.5	8.8	7.2
	noT_1_CPH	27.2	28.8	8.9	7.3
	noT_2_CPH	27.3	29.0	9.0	7.3
	noT_3_CPH	27.3	29.0	8.9	7.3
Summer	T_0_CPH	21.0	22.4	7.2	8.5
	T_1_CPH	21.7	22.9	7.6	8.8
	T_2_CPH	22.0	23.1	7.7	8.8
	T_3_CPH	22.2	23.3	8.0	9.1
	noT_0_CPH	21.5	22.8	6.5	7.8
	noT_1_CPH	22.2	23.4	8.5	9.7
	noT_2_CPH	22.5	23.6	9.3	10.5
	noT_3_CPH	22.6	23.6	9.3	10.3

Table 47: Embedded system return temperature for Paris

		T_{return} GF	T_{return} FF	σ GF	σ FF
Winter	T_0_PAR	30.1	31.3	6.8	5.6
	T_1_PAR	29.9	31.2	6.9	5.6
	T_2_PAR	30.0	31.3	6.8	5.6
	T_3_PAR	30.0	31.3	6.9	5.6
	noT_0_PAR	26.5	28.2	8.7	7.0
	noT_1_PAR	26.5	28.2	8.5	6.8
	noT_2_PAR	26.5	28.2	8.6	6.9
	noT_3_PAR	26.5	28.2	8.7	7.0
Summer	T_0_PAR	20.5	22.0	6.3	7.8
	T_1_PAR	21.4	22.7	7.6	9.0
	T_2_PAR	21.8	23.0	7.5	8.7
	T_3_PAR	21.9	23.1	7.8	9.0
	noT_0_PAR	21.3	22.6	5.5	6.7
	noT_1_PAR	21.6	22.8	6.7	7.9
	noT_2_PAR	21.9	23.0	7.5	8.6
	noT_3_PAR	22.0	23.1	7.8	8.9

It has to be mentioned that results would be different in case the “active” surface were concrete-based instead of wooden.

It is interesting to highlight the trend of embedded system supply temperature with respect to the thermal capacity. It is presented in Figure 60 and Figure 61

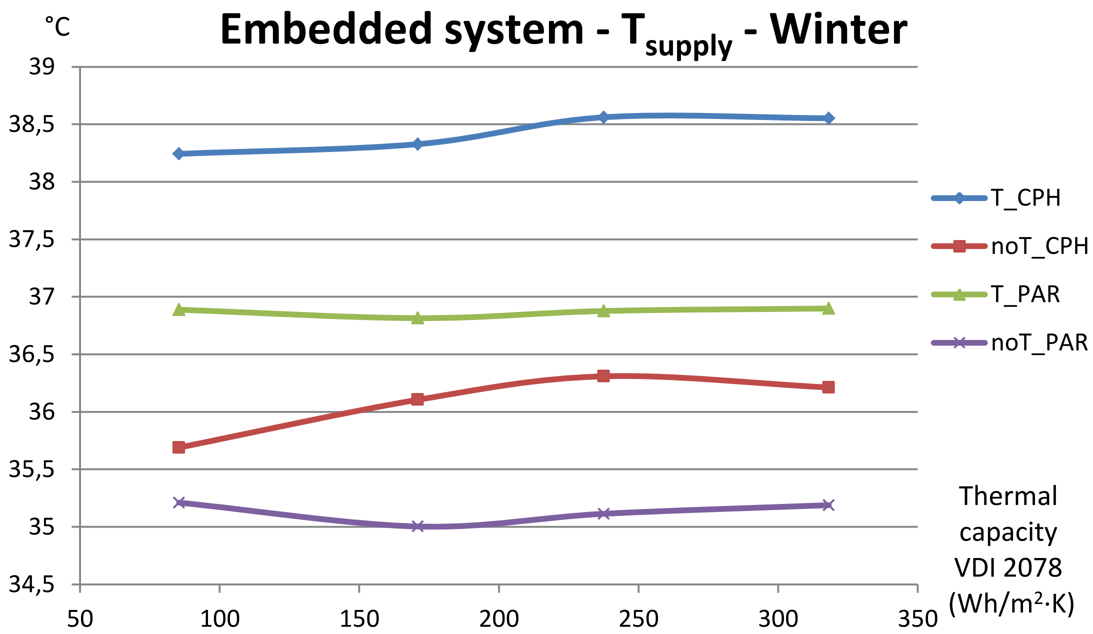


Figure 60: Embedded system – Supply water temperature for winter season

As previously mentioned, the presence of the tank affects the performances of the system and results in higher supply water temperatures, in both of the climates considered. Since Paris cases are characterized by warmer weather conditions, T_{supply} is lower than in Copenhagen cases. However, the trend generally consists of higher supply water temperature for higher building thermal capacity values.

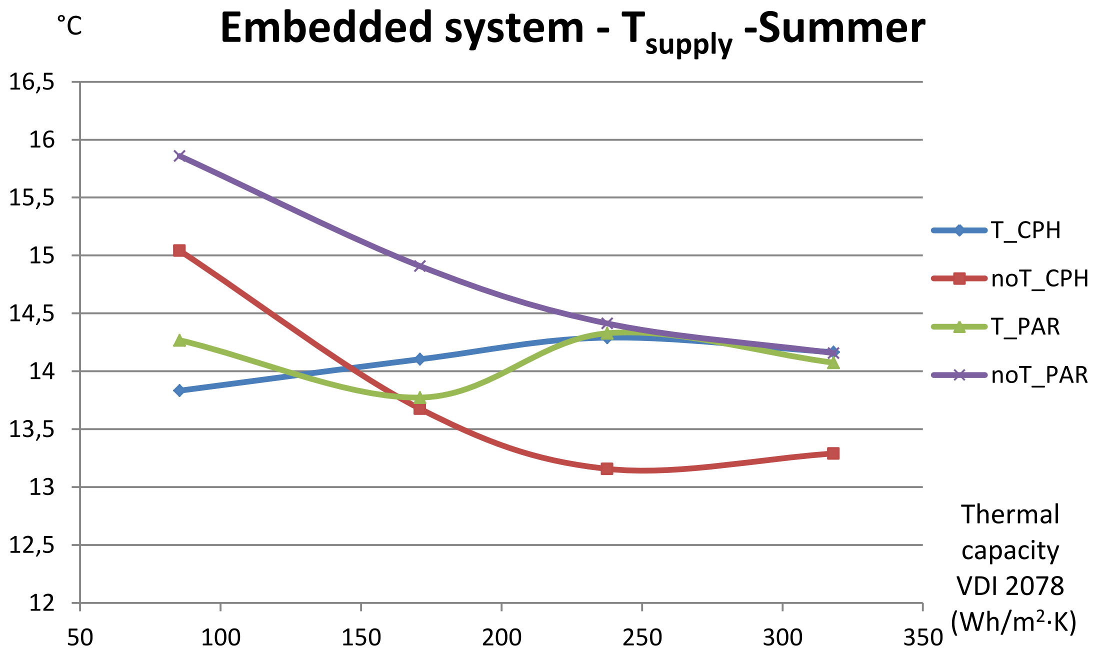


Figure 61: Embedded pipes – supply water temperature for summer season

As mentioned before, T_{supply} for Copenhagen and Paris are slightly different in the cases "T", i.e. the cases where the storage tank is included in the model; in the cases where the tank is avoided T_{supply} tends to be higher in Paris rather than in Copenhagen. Furthermore, it´s observed that the general tendency is to have lower supply temperatures for higher thermal capacities in both of the climates in the cases in which the tank is avoided, while in case the tank is present supply temperature tends to be higher in buildings with higher thermal capacity. This difference could be explained considering that systems not-including the tank are characterized by a faster-response (in fact, the tank, which has a buffer role, is excluded) but this contradiction would deserve further studies.

9.3 Operational hours - Energy consumption/production - Efficiency

Important parameters when evaluating the energy performance of the building are the number of hours where the auxiliary devices and main components are operative and their corresponding energy consumption. Obviously, these parameters are strongly related. In the following subchapters results are presented.

9.3.1 Photovoltaic/thermal solar collectors (PV/Ts)

9.3.1.1 Electrical energy production

Photovoltaic/thermal solar collectors are characterized by the features described in subchapter 7.3.2.1. The results are presented in the following Figure 62 and Figure 63.

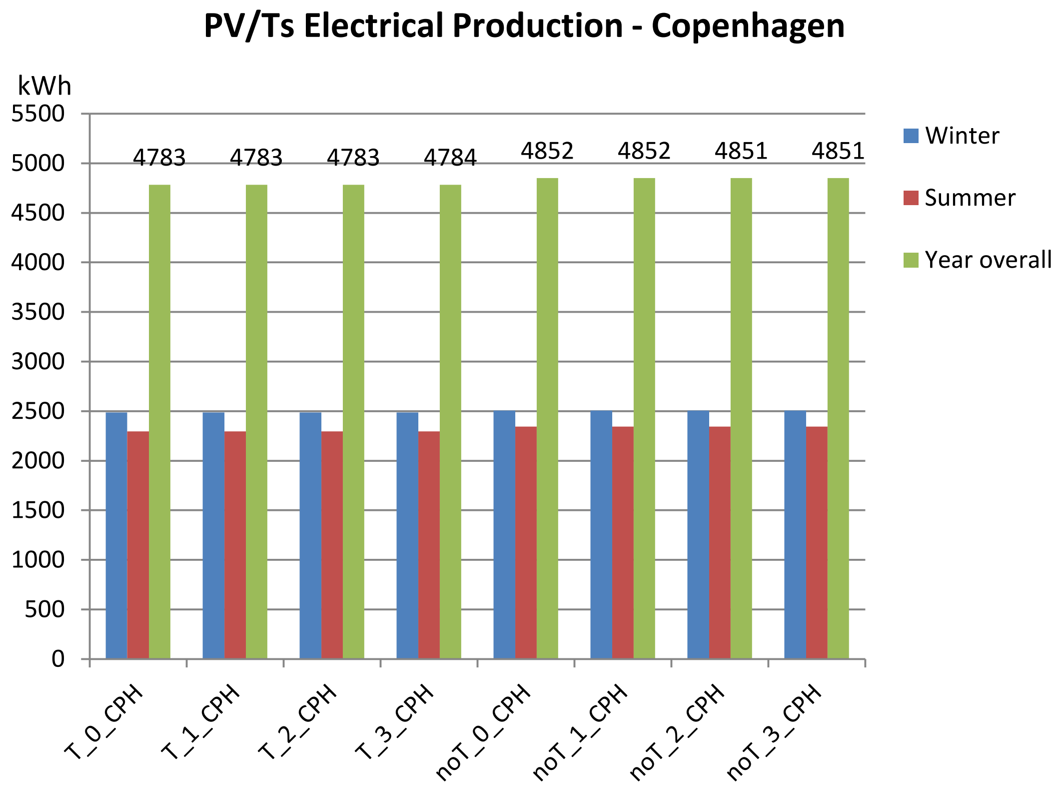


Figure 62: PV/Ts electrical production - Copenhagen

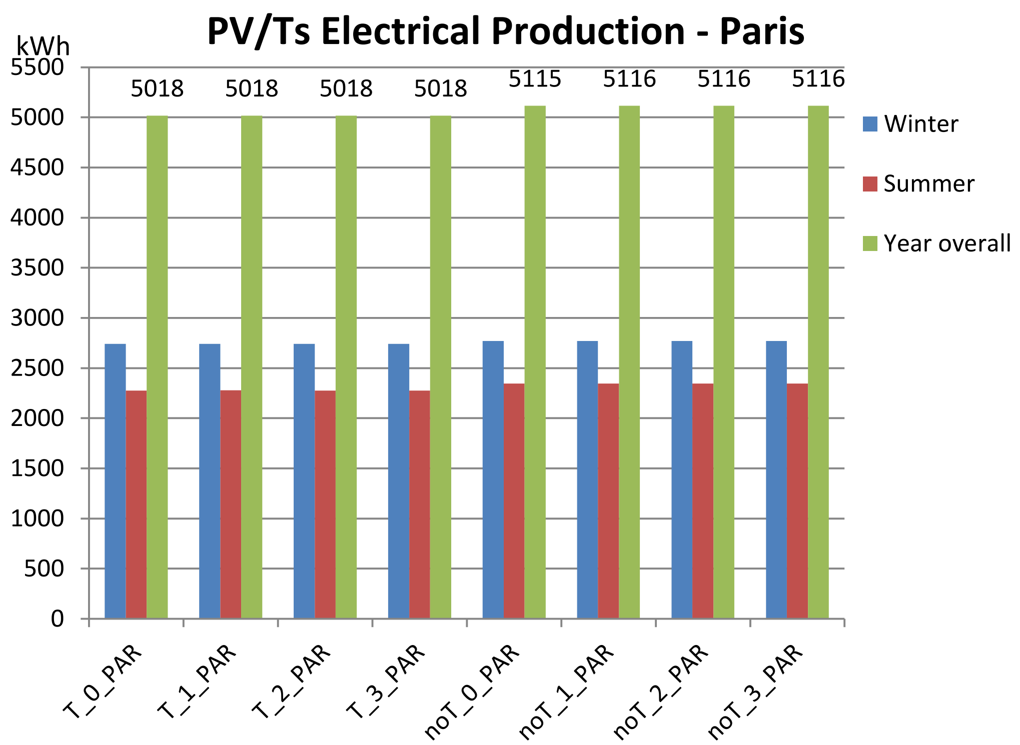


Figure 63: PV/Ts electrical production - Paris

It is observed that the thermal mass does not influence almost at all the PV/Ts electrical production. On the other hand the presence of the tank affects the electrical production (+1.42% for Copenhagen and +1.95% for Paris). This can be explained considering the different control strategies used whether the tank is included or not: if the system does not contain a water storage tank, the control strategy allows PV/Ts to feed directly the radiant floor if the temperature is within a certain range and therefore this results in an higher number of operations hours in which the water circulates through the collectors, maintaining them cooled and thus increasing their electrical efficiency.
As expected, electrical production is Paris is higher than in Copenhagen (+4.68% in case the tank is included, +5.44% in case the tank is not included).
Winter production results higher than summer production: this is due to the modification of the season-schedule settings, which takes into account longer periods for winter-time.

9.3.1.2 PV/Ts electrical and thermal efficiency

In this subchapter, electrical and thermal efficiency results are presented.
Figure 64 shows the electrical efficiency of the PV/Ts for the different combinations.

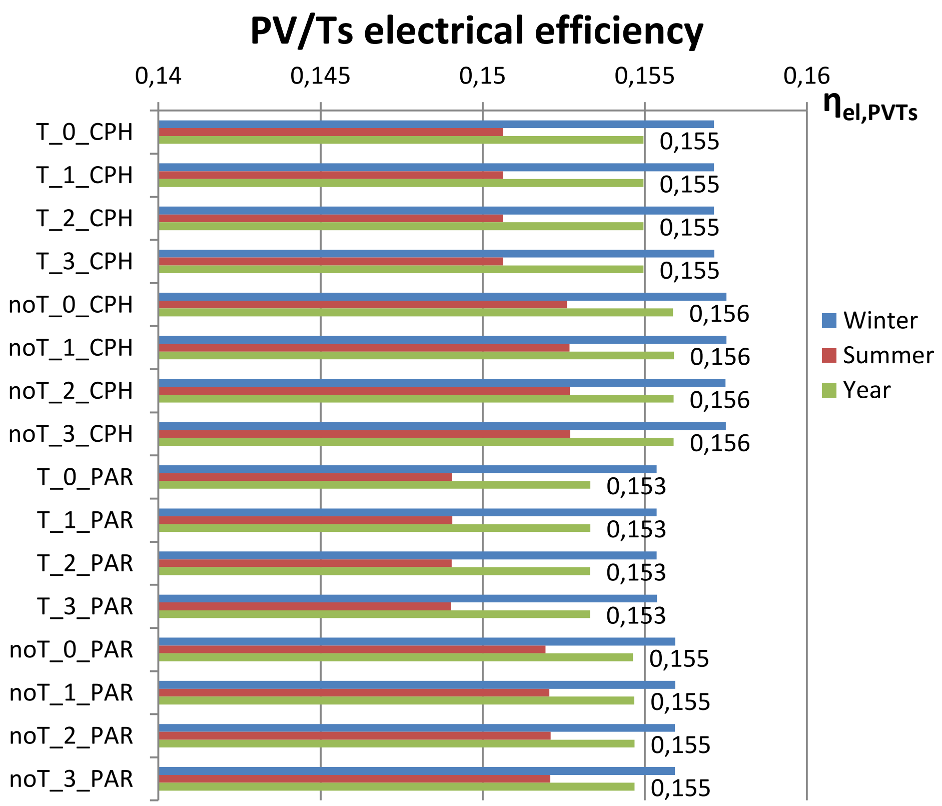


Figure 64: PV/Ts electrical efficiency – data labels are referred to the yearly average values

Figure 64 shows that the electrical efficiency is constant, despite the differences in thermal mass. Electrical efficiency, as mentioned in subchapter 9.3.1.1, is slightly higher when the water tank is not included in the system: in fact, due to the control strategy the water circulates through the solar collectors for longer periods, keeping their temperature lower and therefore increasing their electrical efficiency; in fact, the electrical efficiency is a function of the cell temperature and the incident radiation (TESSLibs 17 - Electrical Library, mathematical reference, 2012).

Figure 65 shows the thermal efficiency of the PV/Ts, calculated as the fraction of incident solar radiation converted to delivered fluid energy (TESSLibs 17 - Electrical Library, mathematical reference, 2012).

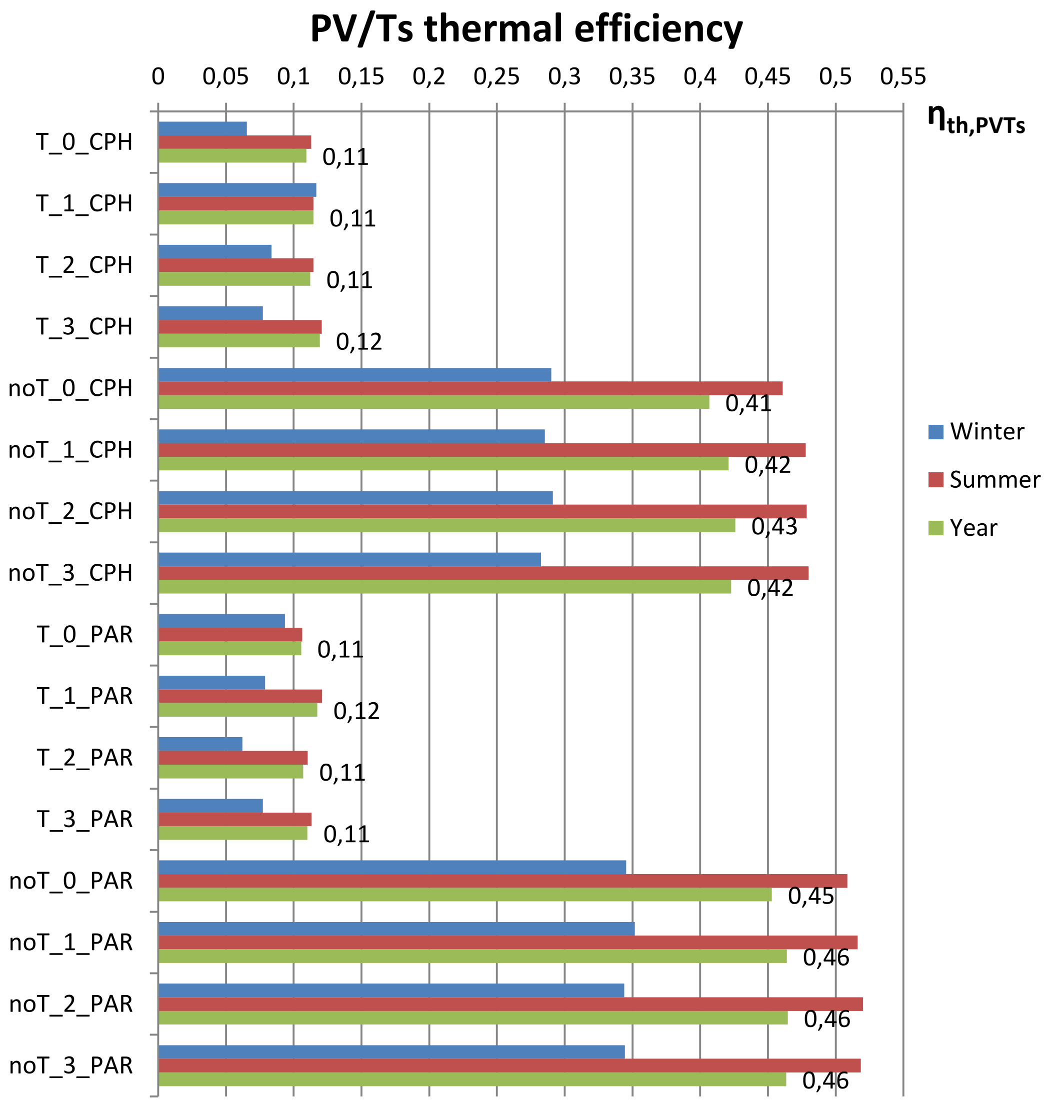


Figure 65: PV/Ts thermal efficiency – data labels are referred to the yearly average values

Thermal mass appears to have a slightly positive effect in terms of thermal efficiency of the solar collectors. Furthermore, the presence of the tank affects in a significant way the performances, probably due to the number of operational hours

in which the pump feeds the PV/Ts: since it is higher in the cases without the storage tank, this results in higher thermal efficiencies.
The most significant contribution is provided by the summer-season, as expected.

9.3.1.3 Pump operational hours and energy consumption

This chapter presents the results regarding the operational hours of the pump driving the PV/Ts.

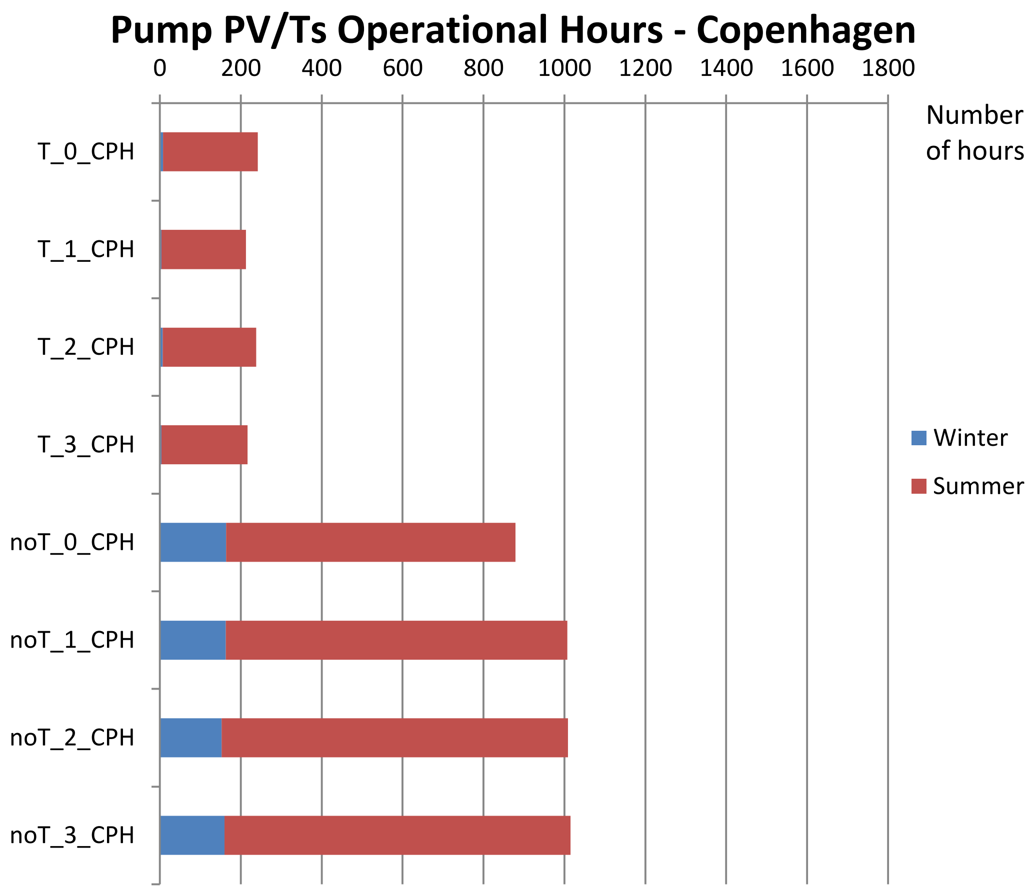


Figure 66: Pump PV/Ts operational hours – Copenhagen

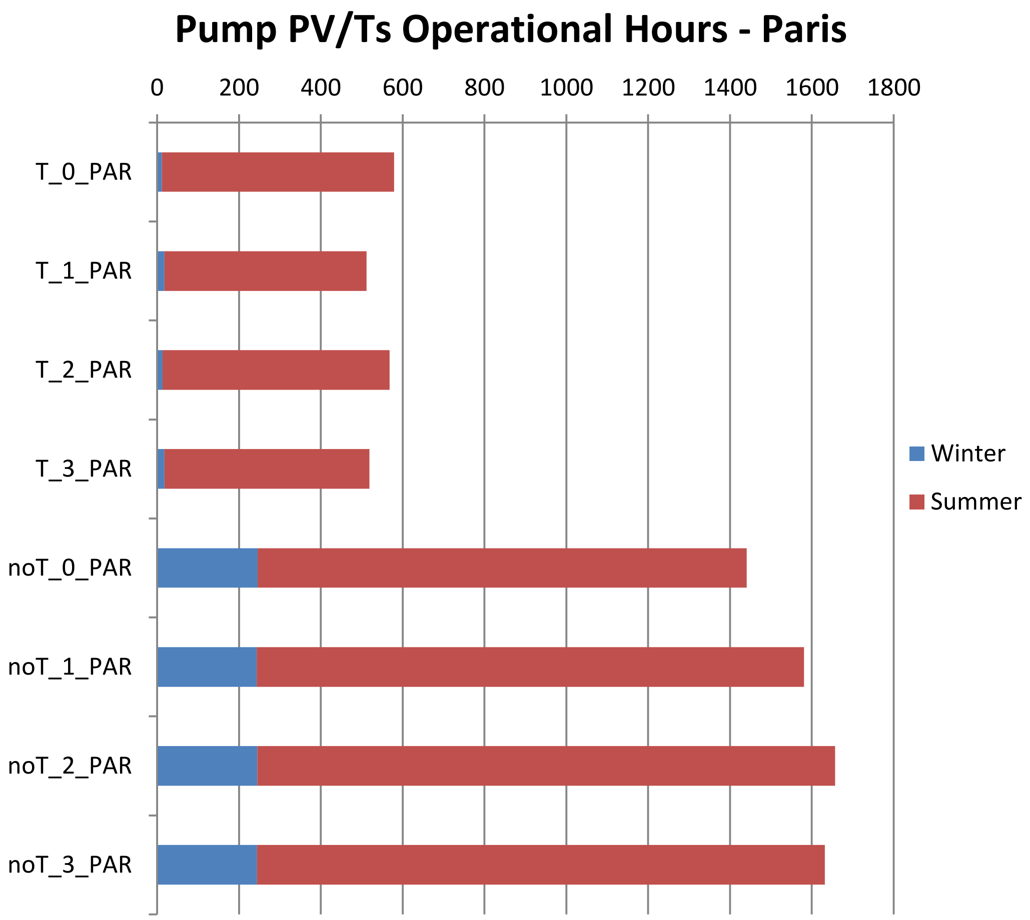


Figure 67: Pump PV/Ts operational hours - Paris

Results are listed in Table 48.

Table 48: Pump PV/Ts operational hours

	CPH - Winter	CPH - Summer	CPH - Year	PAR – Winter	PAR - Summer	PAR - Year
T_0	8	234	242	12	567	579
T_1	4	209	213	18	494	512
T_2	7	231	238	13	555	568
T_3	4	213	217	18	501	519
noT_0	164	715	879	246	1195	1441
noT_1	163	844	1007	243	1338	1581
noT_2	153	856	1009	245	1412	1657
noT_3	160	855	1015	244	1388	1632

It is observed from the figures and the tables above that the presence of the water storage tank significantly affects the number of hours in which the pump driving the PV/Ts is operated. This was expected and it is due to the control strategy. In fact, as mentioned before, in the cases without the water storage tank the PV/Ts are able to feed the embedded system directly if their outlet temperature is within a certain range, which is not possible in the cases with the tank because the hot water stream has to be directed towards the DHW tank first. Since this range is within 30°C÷37°C, this can explain the greater number of hours when the pump is activated.

Despite some fluctuations in the results in case the system is equipped with the tank, it is possible to state that generally the amount of operational hours decreases in buildings with higher thermal mass in case the tank is present, otherwise the behavior is opposite.

Differences are lower for Paris cases because the weather is generally more favorable to the exploitation of solar radiation even in winter-season.

Figure 68 presents the amount of nocturnal operational hours in summer.

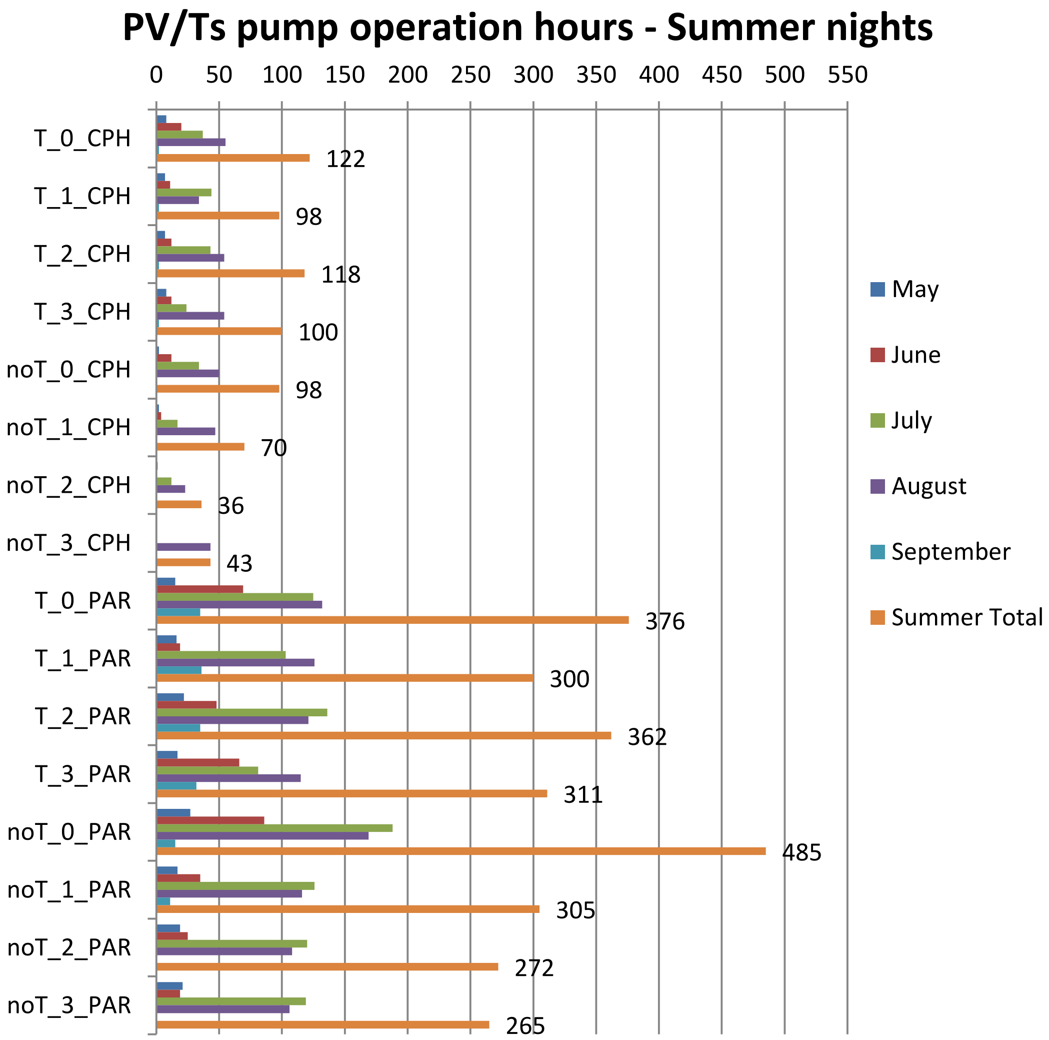


Figure 68: Pump PV/Ts operational hours – Summer nights

From Figure 68 it is possible to state that, with regards to the cases in which the water storage tank is included, the behavior is oscillating and therefore it is not possible to observe a clear trend (even though the number of operational hours seems to decrease in buildings with higher thermal capacity). In case the system is not equipped with the storage tank, it can rely only on the thermal mass of structure: the trend indicates that the activity hours decrease in buildings with higher thermal mass. This is due to the capability of the system to store heat and, since the heat stored does not have to be removed, this results in fewer hours of activity of the pump in "heavier" buildings.

9.3.1.4 Pump energy consumption

Results regarding pump´s energy consumption are presented hereinafter.

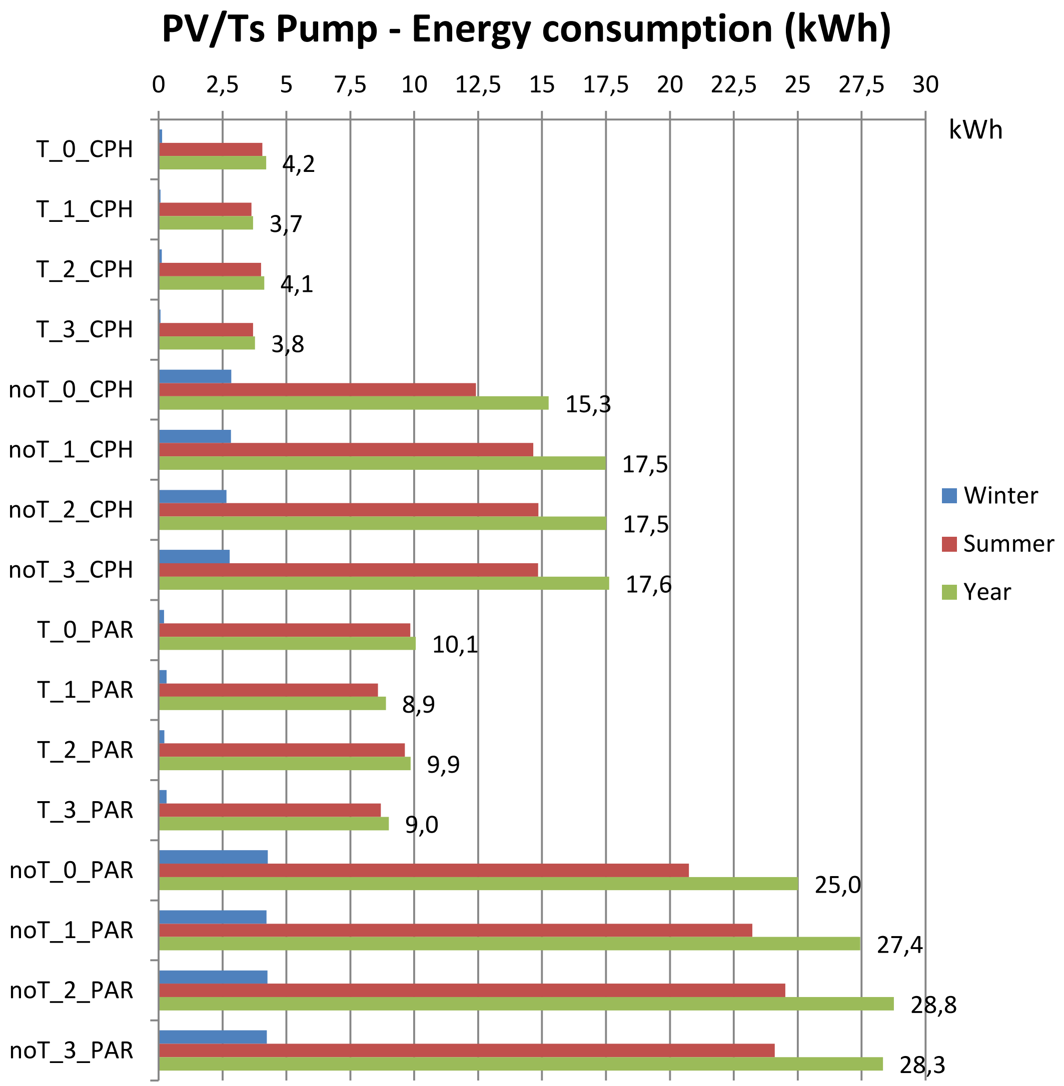


Figure 69: PV/Ts pump energy consumption [kWh/year]. ***Data labels are referred to the yearly overall energy consumption***

Figure 69 shows that the choice of including the water storage tank or not affects in a remarkable way the energy consumption deriving from the pump. Higher consumptions are observed for both of the climates in case the tank is not included. This is obviously directly dependent to the number of operational hours.
Regarding the influence thermal mass has on the energy consumption of the pump, similar trends are observed with respect to those presented in subchapter 9.3.1.3. Thus, the tendency shows fluctuations in case the water storage tank is included, even though it is possible to state that structure 1, structure 2 and structure 3 present lower energy consumption than structure 0. In case the system is not equipped with the water storage tank, higher energy consumptions are recorded in buildings with higher thermal mass. It is also noticed that in case the water storage tank is included, the operational hours in winter-time are not significant. This though is attributable to the different control strategy, which in this case implements the case in which the PV/Ts are able to feed the embedded system directly with outlet temperatures lower than the ones required for the DHW tank.
Results are higher in simulations involving Paris climate, due to more favorable conditions for the exploitation of the PV/T panels.

9.3.1.5 Night-time radiative cooling

In this subchapter results regarding the night-time radiative cooling effect are presented.
Based on the following figure it is also possible to state that the benefit deriving from the night radiative cooling is strongly climate-dependent.

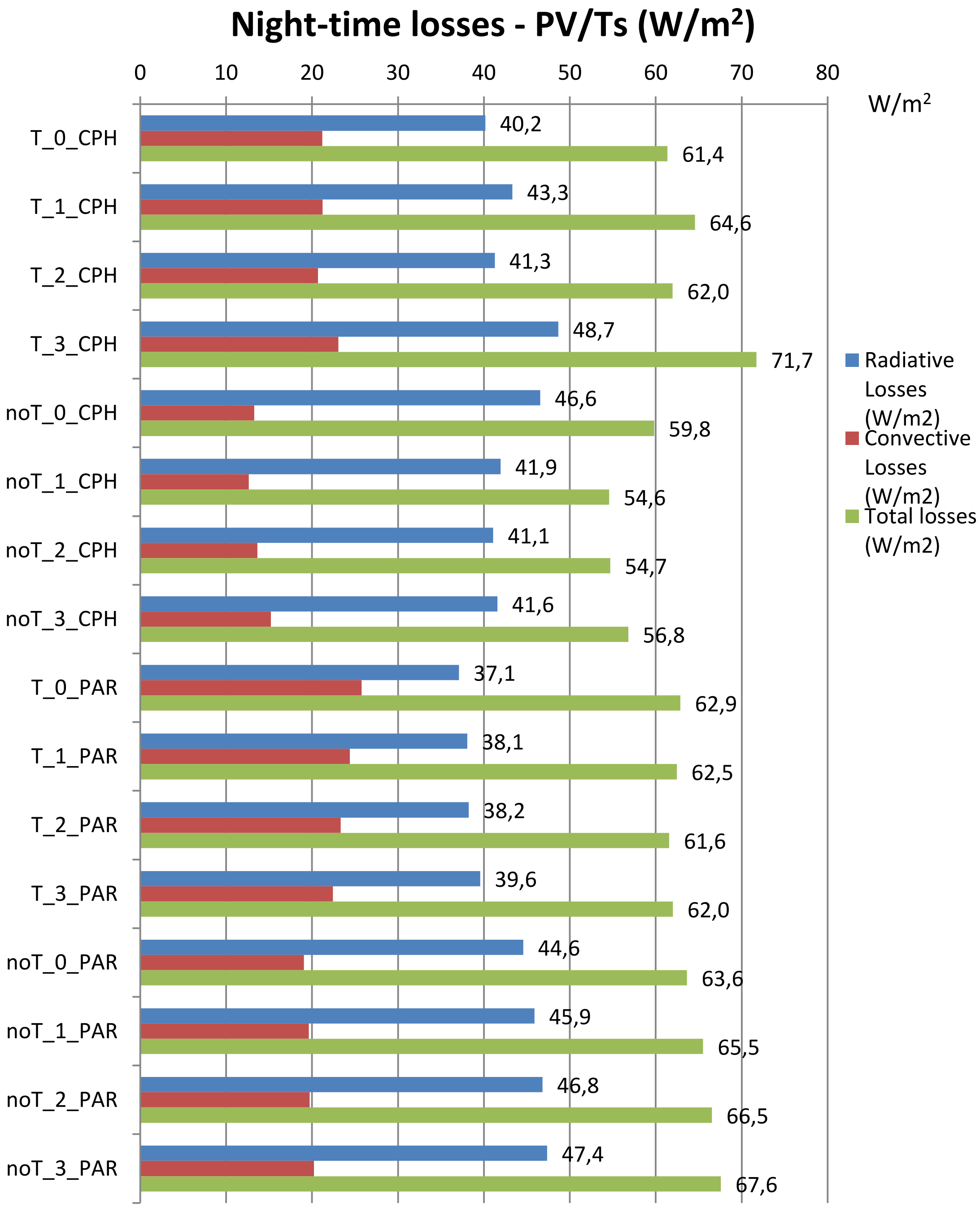


Figure 70: PV/Ts night-time radiative and convective losses. Data labels are referred to the summer average radiative losses and to summer total average losses

Radiative heat exchange relies on the temperature difference between the PV/Ts surface and the sky, while convective heat exchange relies on the difference between PV/Ts surface temperature and outdoor air temperature.

Figure 70 highlights the main results of the night-time radiative cooling effect. Generally, it is possible to state that in “T” cases the radiative losses increase with higher thermal capacity of the structure, since the structure is able to absorb and store higher amounts of heat, which is (partially) eliminated through the night

radiative cooling. In “noT” cases results show opposite results for Copenhagen and Paris. Furthermore, the presence of the tank reduces the free cooling benefits. Values obtained are in good agreement with the ones found in literature (Photovoltaic thermal collectors for night radiative cooling of buildings, 2011). Radiative losses are shown to be relevantly higher than convective losses, both in Copenhagen and in Paris climate.
Regarding “T” cases, losses are higher in Copenhagen than in Paris because of lower nocturnal sky temperature (1.9°C for Copenhagen and 5.2°C for Paris). The differences regarding the cases in which the tank is not included would deserve further investigations.
Figure 71 shows the share of the overall cooling energy demand that can be covered by the night radiative cooling (combining convective and radiative losses). It is possible to state that generally the share tends to increase with higher thermal mass in “T” cases, while the opposite happens for “noT” cases. Results show generally higher benefit for the application in Copenhagen climate.

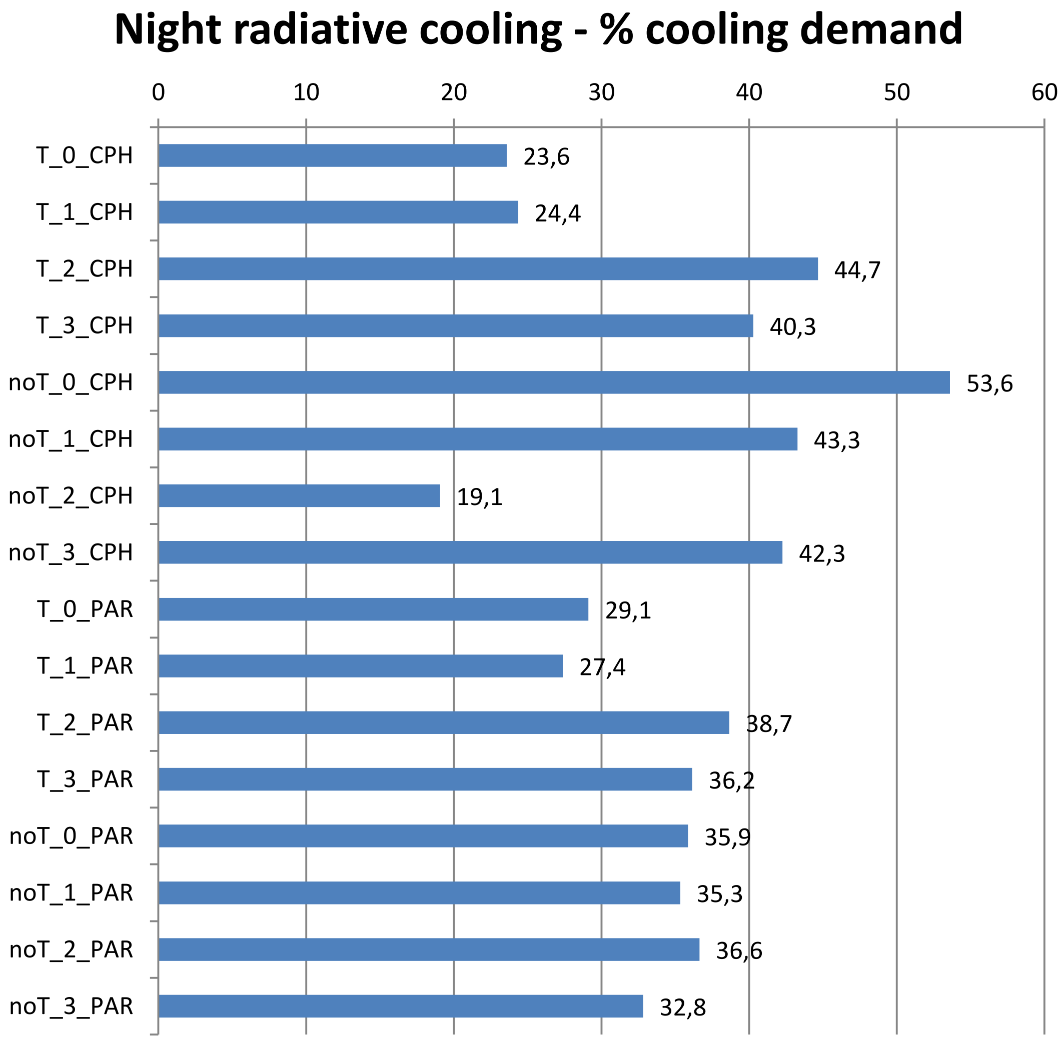


Figure 71: Night radiative cooling - % of the total cooling demand

Figure 72 presents the PV/Ts inlet temperature in case the summer nocturnal operation is activated.

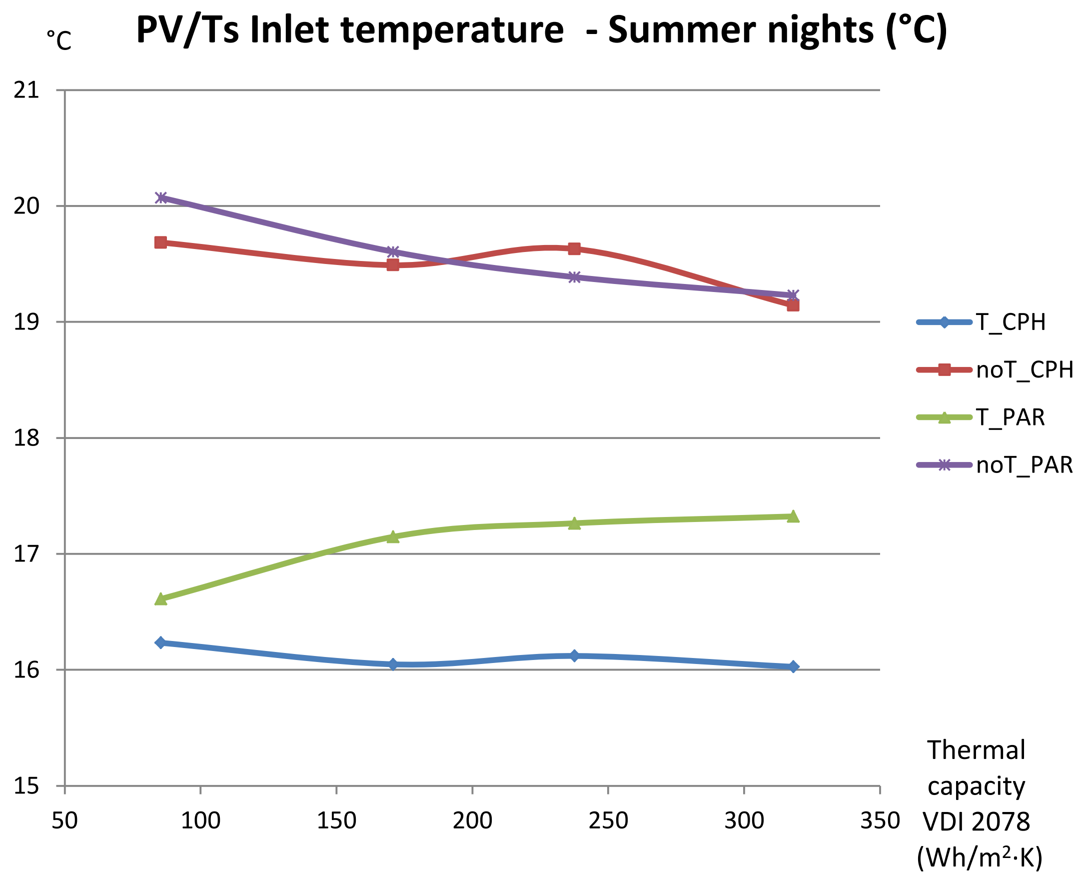


Figure 72: PV/Ts average inlet temperature – summer nights – night radiative cooling

Figure 73 shows the amplitude of the temperature difference between supply and return flows in case the summer nocturnal operation is activated.

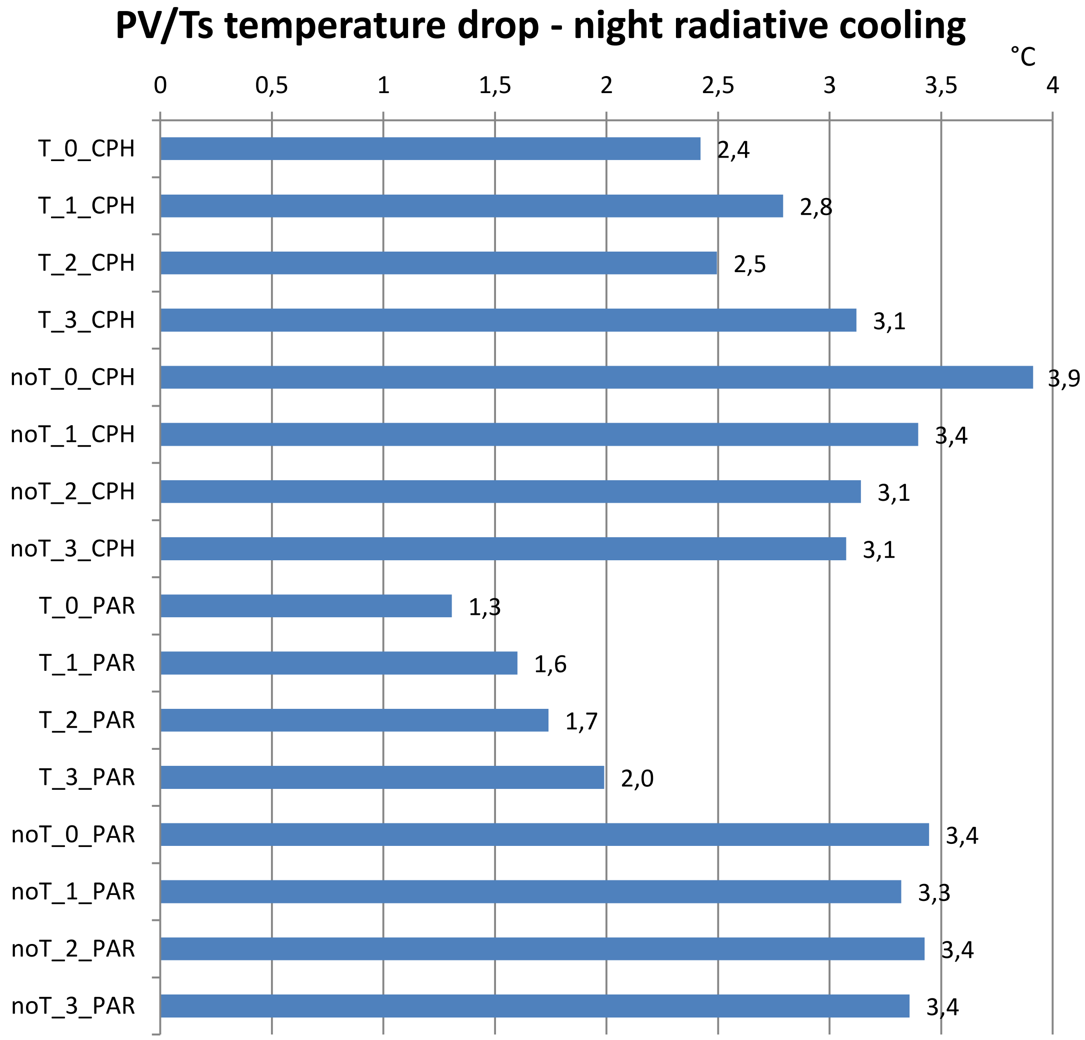


Figure 73: PV/Ts average temperature difference between supply and return – summer nocturnal operation.

It is possible to state that the trend of the temperature drop in PV/Ts when the night radiative cooling is exploited presents some similarities to the one of the PV/Ts losses (Figure 70). In particular, in "T" cases for Copenhagen the trend corresponds to the one of the overall losses in Figure 70, while in "noT" cases the tendency resemble the one of the radiative losses in Figure 70. This is expectable, because radiative heat exchange is based on the temperature´s 4th power, while convective heat exchange is based on the 1st power; thus, radiative heat exchange is predominant. The same observation can be extended to "T" cases for Paris. Regarding "noT" cases in Paris it is difficult to highlight either a clear trend or a correspondence with Figure 70; however it should be kept in mind that under these conditions PV/Ts are operated for longer time, resulting in higher surface temperatures. Thus, the correspondence between the water temperature drop and the losses (associated to the difference between the PV/Ts surface temperature and either the sky temperature or the outdoor air temperature, depending on the case) is probably concealed by the initial cooling of the surface.

9.3.2 Heat pump

9.3.2.1 Pump operation hours

This subchapter highlights the results in terms of number of hours in which the pump driving the heat pump is activated, that corresponds to the number of hours in which the heat pump is operative.

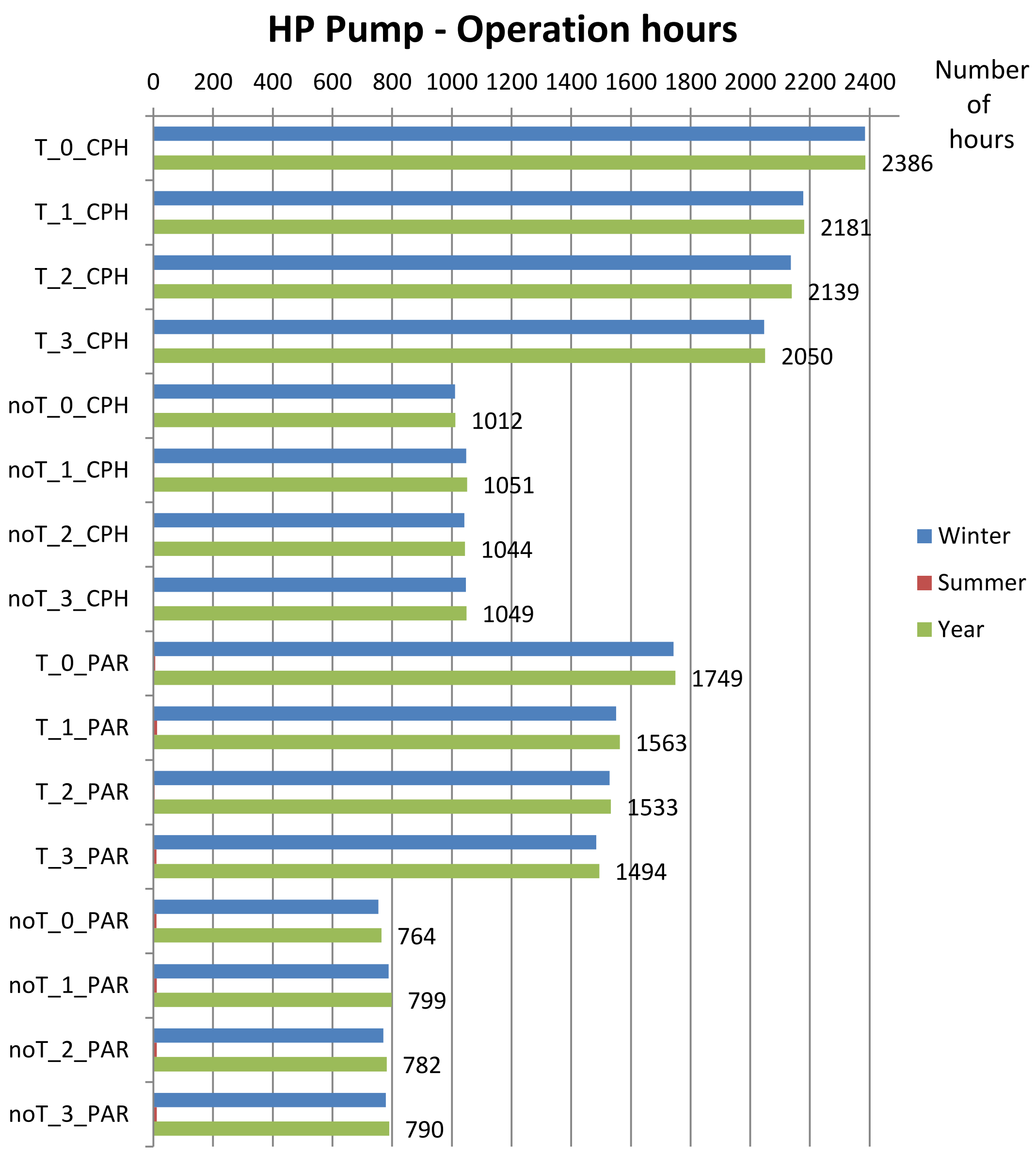


Figure 74: Heat pump´s operational hours – Data labels are referred to the yearly overall number of operative hours

Figure 74 highlights the influence the water storage tank has to the system, resulting in higher number of hours in which the pump driving the HP is activated. In fact, the tank represents a large amount of water that needs to be either heated up or cooled down. This though could lead to unnecessary periods of activity of the heat pump, because regardless of the heating/cooling of the house it is in charge to maintain the temperature inside the water storage tank within certain ranges. Therefore, even though the house does not require any water flow from the house, the tank needs to be heated or cooled. Differences between the cases with or without the tank (with fixed structure) are relevant: from +95% to +136% for Copenhagen and from +89% to +129% for Paris. Despite the differences in terms of absolute values, relative differences are therefore almost identical for both of the climates.

Furthermore, it is observed that in "T" cases the number of hours in which the heat pump is activated decreases with higher thermal capacity of the building. This behavior is observed in winter-time, while in summer-time the amount of operative hours is almost negligible. Therefore, the water storage tank is averagely warmer and therefore it is possible to decrease the amount of hours in which the heat pump is activated.

"noT" cases are more unstable and it is difficult to highlight a clear tendency of operational hours of the heat pump depending on the thermal capacity of the building.

Pump is almost never activated in cooling-season, because the system can rely on the thermal capacity of the building (and in case the one of the water storage tank) and on the night-time radiative cooling.

Since the operation strategies of the pump driving the HP and the HP itself are closely related, HP´s operative hours result identical to those described in this subchapter.

9.3.2.2 Pump energy consumption

Regarding the energy consumption of the pump driving the heat pump, results are presented below.

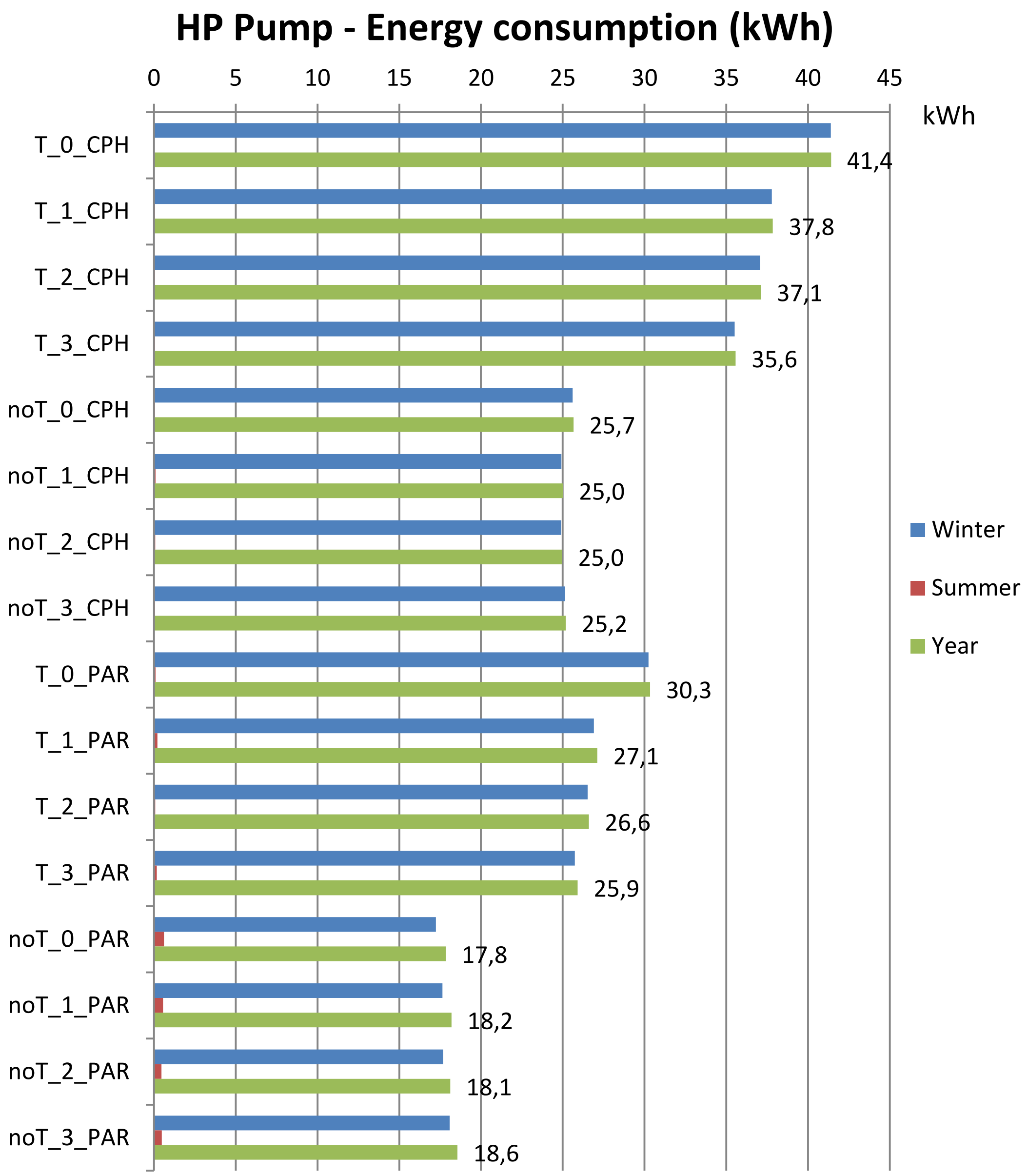


Figure 75: Heat pump´s pump energy consumption (kWh) – data labels are referred to the yearly overall energy consumptions

General tendencies are identical to those explained in subchapter 9.3.2.1 regarding operative hours of the pump (energy consumption is clearly correlated to the number of operative hours), even though relative differences are different to the previous ones: from +41% to +61% for Copenhagen, from +40% up to +70% for Paris. The pump power is recorded among the output of the TRNSYS component implementing the pump and it is calculated from either a linear relationship with flow rate or by a polynomial expression relating pump power to control signal, depending on the parameters supplied by the user.

9.3.2.3 Heat pump energy consumption

Heat pump energy consumption is recorded through the output named "Heat pump power" of the TRNSYS component. According to TRNSYS software manual (TRNSOLAR Energietechnik GmbH, 2010), it consists of the total power (compressor, blower, controllers) consumed by the device while operating, also including the contribution of any auxiliary heater, if present. The heat pump is not provided with any auxiliary heater.
Accordingly, values recorded are electrical energy.
The results are presented in the following Figure 76.

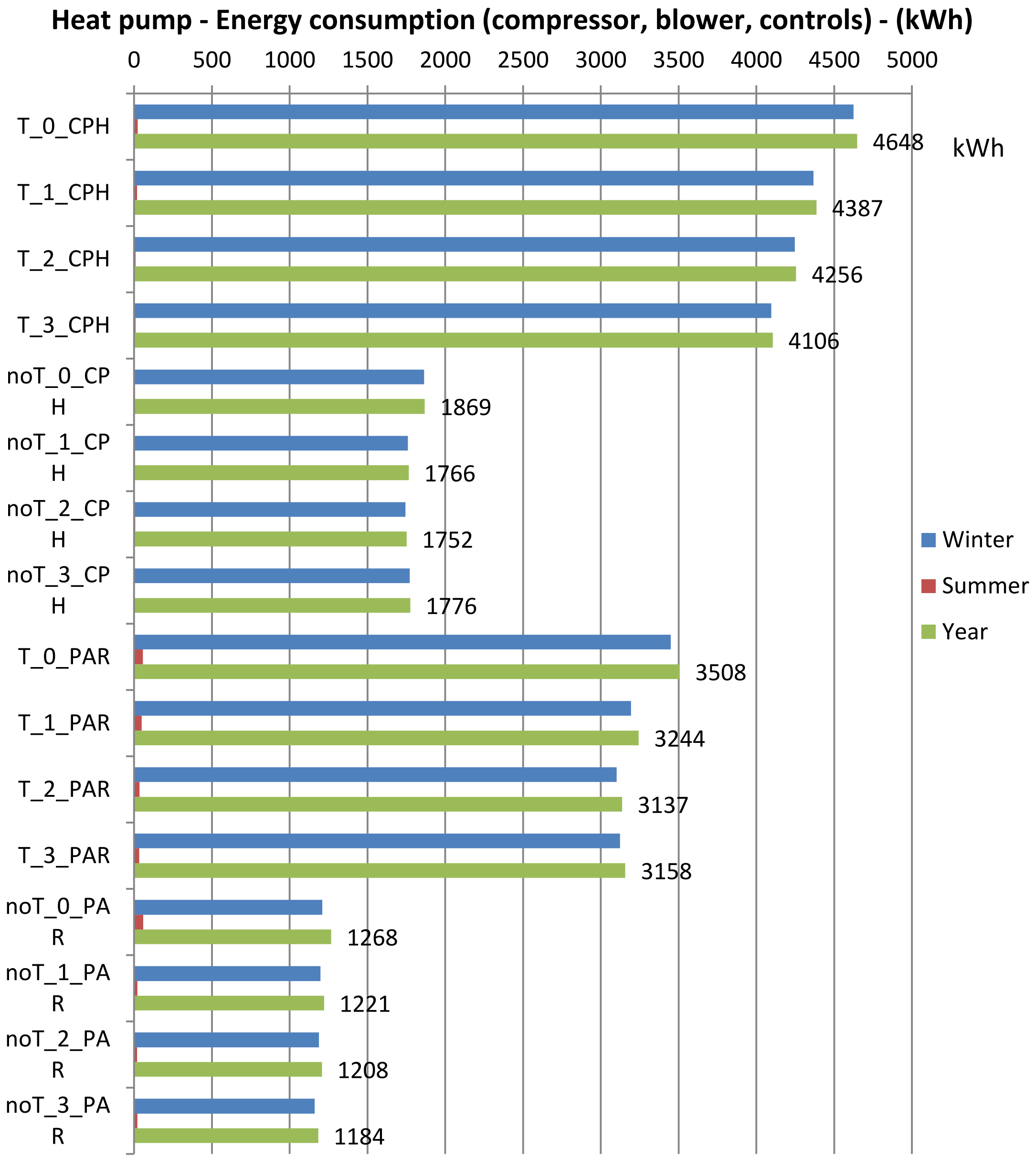


Figure 76: Heat pump energy consumption (kWh) – data labels are referred to the yearly overall energy consumption

General tendency reflects the one observed regarding operative hours and energy consumption of the pump driving the device. Therefore, the presence of the water storage tank affects deeply the energy consumption of the heat pump because, when present, it has to heat or cool a large amount of water (even when the house does not require any flow), process which is energy-consuming. Including the water storage tank in the system or not makes a relevant difference, resulting in higher energy consumption: from +131% to +159% for Copenhagen, from +159% to +177% for Paris. Since the main contribution to the yearly overall consumption derives from the energy consumed during the heating-season, Paris climate is more favorable and results in lower energy consumptions (with respect to Copenhagen climate).

9.3.2.4 Heat pump´s Coefficient of Performance

In this subchapter the results regarding the evaluation of Coefficient of Performance (COP) of the heat pump are highlighted for each considered combination. COP is calculated as the ratio of the rate at which heat is released by the cooling medium in the condenser and the electrical power consumed by compressor and fans (TESSLibs 17 - HVAC Library, Mathematical reference, 2012). Figure 77 shows the COP value for every case taken into account.

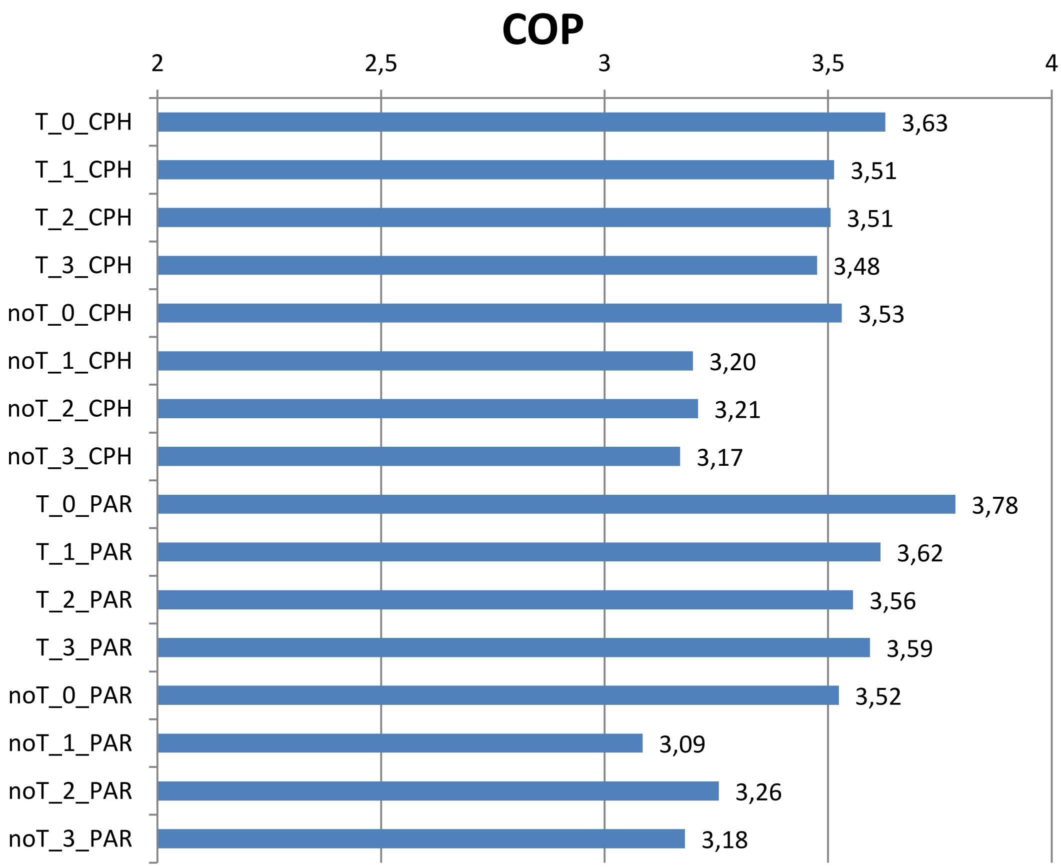


Figure 77: COP value for the heat pump

Figure 77 underlines that COP value decreases with higher thermal capacity structures. This is due to the fact that, in winter conditions, the condenser-side of the heat pump is associated with warmer temperatures (being the water tank temperature in case it is present, the embedded pipes water temperature otherwise according to Figure 16 and Figure 17). The presence of the tank is beneficial for the COP values of the device, allowing the heat pump to exchange more regularly and avoiding excessive frequency on ON/OFF, which is harmful for the heat pump.

COP values for Paris are slightly higher because the heat pump is able to absorb heat on the evaporator-side from warmer outside air, thus reducing the difference between the condensation temperature and the evaporation temperature and therefore increasing its thermodynamic performance.

9.3.3 Domestic Hot Water tank

9.3.3.1 DHW tank auxiliary heater

DHW tank is provided with an auxiliary heater in charge of heating up the water to its set-point temperature in case the system is not able to satisfy this requirement. Its energy consumption is recorded through the output named “Auxiliary heating rate” of the TRNSYS component. Energy provided by the auxiliary heater is accounted as electrical energy.

Figure 78 shows the auxiliary heater consumptions for each of the considered combinations.

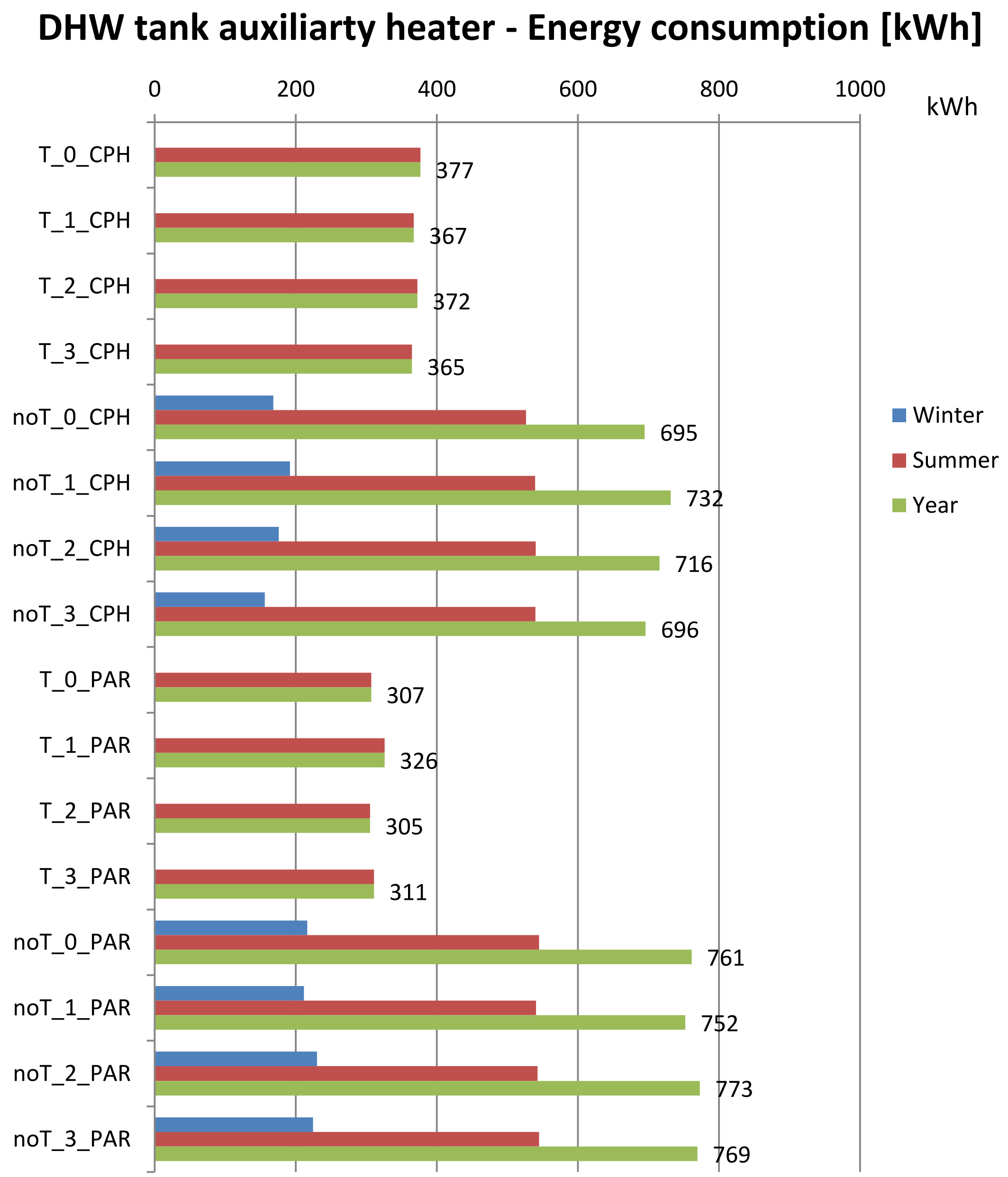


Figure 78: DHW tank auxiliary heater – energy consumption (kWh) – data labels are referred to the yearly overall energy consumption

It is observed that the most relevant contribution to the overall yearly energy consumption derives from the cooling season, when the only heating source is represented by the PV/Ts and it is likely that it is not enough to maintain an appropriate temperature at the top of the DHW tank; therefore the auxiliary heater needs to be activated.

In the cases in which the water storage tank is included in the system, the auxiliary heater is activated for a non-significant amount of hours in winter season. On the other hand, in the "noT" cases, the auxiliary heater is activated in winter season: this is probably due to a more unstable system, where the PV/Ts are in charge of

feeding either the embedded system or the DHW tank, and the tank itself conveys often a flow rate directed to the radiant floor, therefore with higher mixing tendency (or less favorable stratification conditions) resulting in higher consumption by the auxiliary heater to maintain the necessary temperature at the top of the tank. In reality though, the system are decoupled by the presence of a heat exchanger and therefore the effect could differ. In the TRNSYSY model decoupling heat exchangers are avoided in order to simplify its structure. The observation mentioned above could also explain the difference between the “T” cases and the “noT” cases, where the cases in which the water storage tank is avoided present higher consumptions: +84÷99% for Copenhagen and +131÷153% for Paris.
The influence thermal mass has on DHW auxiliary heater’s energy consumption is not clear.
Energy consumption in Paris is observed to be lower in case the tank is included, while it is higher in case the tank is excluded.

9.3.3.2 Temperature to load

Table 49 shows the “temperature to load”, which is the temperature sent to the tapping appliances (mixed with cold water). Following temperatures in Table 49 derive from the output of the DHW tank TRNSYS component and do not include mixing with cold water.
Table 49 highlights that the temperatures to load are among the acceptability range, thus proving that the control strategy satisfies the requirements for DHW.
During the heating season, temperatures delivered to the “load” are significantly higher in case the system is equipped with a water storage tank rather than in case the tank is absent. This is due to the fact that the tank allows storing warm water, resulting in higher temperatures of the PV/Ts inlet flow and therefore higher temperatures of outlet flow, which is then directed to the DHW tank.
Thermal mass does not appear to have a relevant effect on the average temperature delivered “to load”; it is though observed that in “T” cases it affects the maximum temperature delivered, resulting in higher temperature “to load” in buildings with higher thermal mass.
Temperatures delivered “to load” are higher in winter season rather than in summer season, due to the use of three alternative sources during the heating season (PV/Ts, the heat pump and the auxiliary heater) while during the cooling the cooling season the DHW tank relies on two source (the PV/Ts and the auxiliary heater). This consideration is correlated to Figure 78, in which it is shown that the biggest contribution to the DHW tank auxiliary heater´s consumption derives from summer-season.
Temperatures “to load” are generally slightly higher in Paris, due to more favorable weather conditions to the exploitation of the combination of the PV/Ts and the heat pump, which can guarantee higher inlet temperatures in the DHW tank and more constantly.

Table 49: DHW tank – Temperature to load

	Winter			Summer		
	T_{min}	T_{max}	$T_{average}$	T_{min}	T_{max}	$T_{average}$
T_0_CPH	47.4	69.6	59.1	44.0	58.9	53.1
T_1_CPH	49.7	69.4	59.4	44.0	61.7	53.1
T_2_CPH	49.7	70.0	59.5	44.0	59.0	53.1
T_3_CPH	49.5	70.3	59.4	44.2	64.8	53.1
noT_0_CPH	45.1	63.3	54.9	47.5	55.8	53.8
noT_1_CPH	44.8	63.1	54.8	47.5	55.8	53.8
noT_2_CPH	43.9	62.3	54.9	48.2	55.8	53.8
noT_3_CPH	44.9	63.0	55.0	48.1	55.8	53.8
T_0_PAR	49.0	70.9	60.4	44.1	59.3	53.1
T_1_PAR	49.0	71.6	60.4	44.1	59.6	53.2
T_2_PAR	49.9	73.6	60.5	44.1	64.8	53.1
T_3_PAR	49.9	70.8	60.5	44.2	65.3	53.1
noT_0_PAR	45.6	65.6	54.5	48.4	55.9	53.9
noT_1_PAR	45.4	64.6	54.8	48.4	55.9	53.9
noT_2_PAR	45.5	63.7	54.8	48.4	55.9	53.9
noT_3_PAR	45.5	63.9	54.8	48.4	55.9	53.9

9.3.4 Ventilation

9.3.4.1 Ventilation energy consumption

Ventilation system energy consumption is recorded through the output called “Air side heat transfer” of the TRNSYS components for the heating and humidifying coil (winter) and the cooling coil (summer). The results are presented in the figures below.

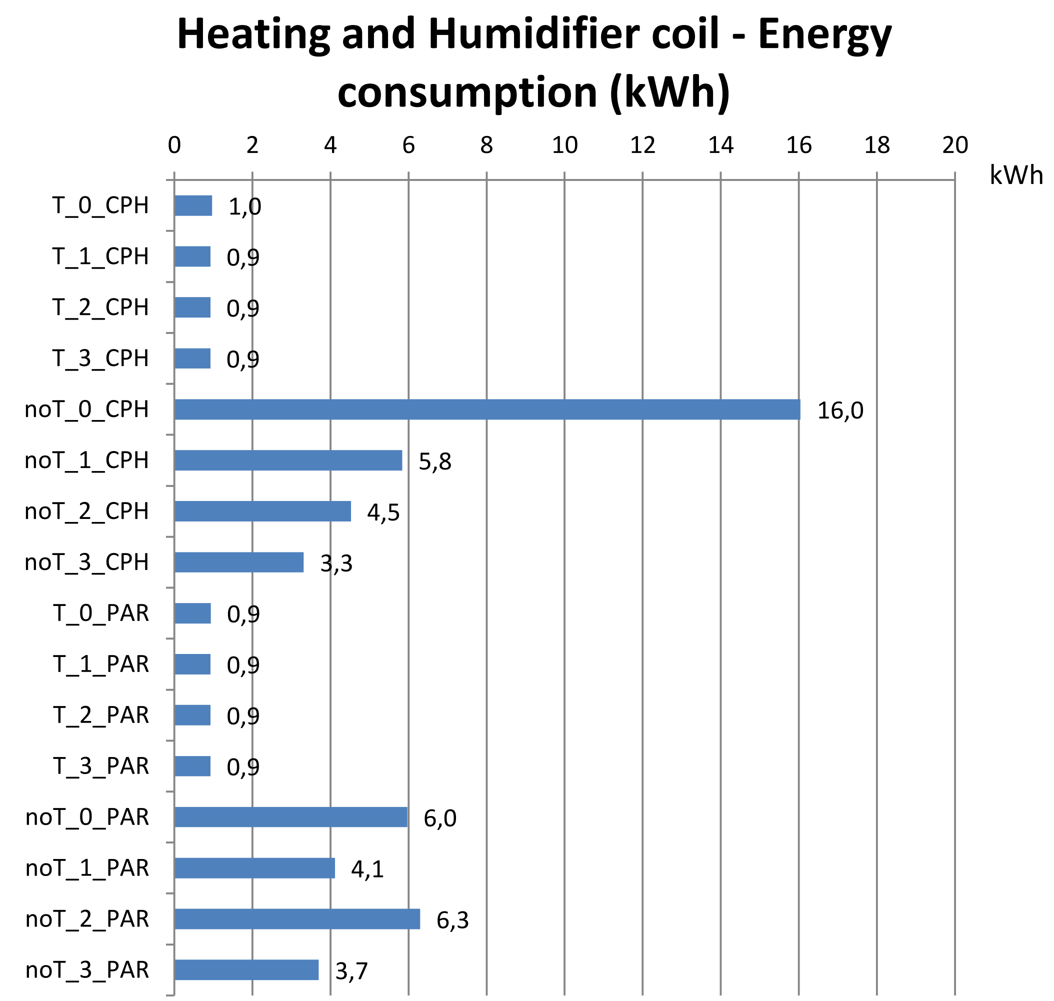


Figure 79: Heating and Humidifying coil – Energy consumption (Winter season – kWh)

Figure 79 highlights the energy consumption of the ventilation system during the heating season. It shows that consumption is higher in the cases where the water storage tank is not included: indoor temperature is lower in those cases and therefore it is possible to recover a smaller amount of energy to preheat the "fresh" air in the heat recovery system. Thus, the system has to provide greater quantities of heat.

Thermal mass appears to have a positive influence in the reduction of the energy consumption, except the case the system is not provided with the tank in Paris conditions, where the behavior is less clear.

Consumptions are generally lower in Paris climate because of higher temperatures of "fresh" supply air.

Heating and humidifier energy consumption is accounted as electrical energy, assuming the presence of a simple electrical resistance.

In the following figure the energy consumption of the cooling coil is presented. It is recorded among the output of the TRNSYS component as "Air side heat transfer".

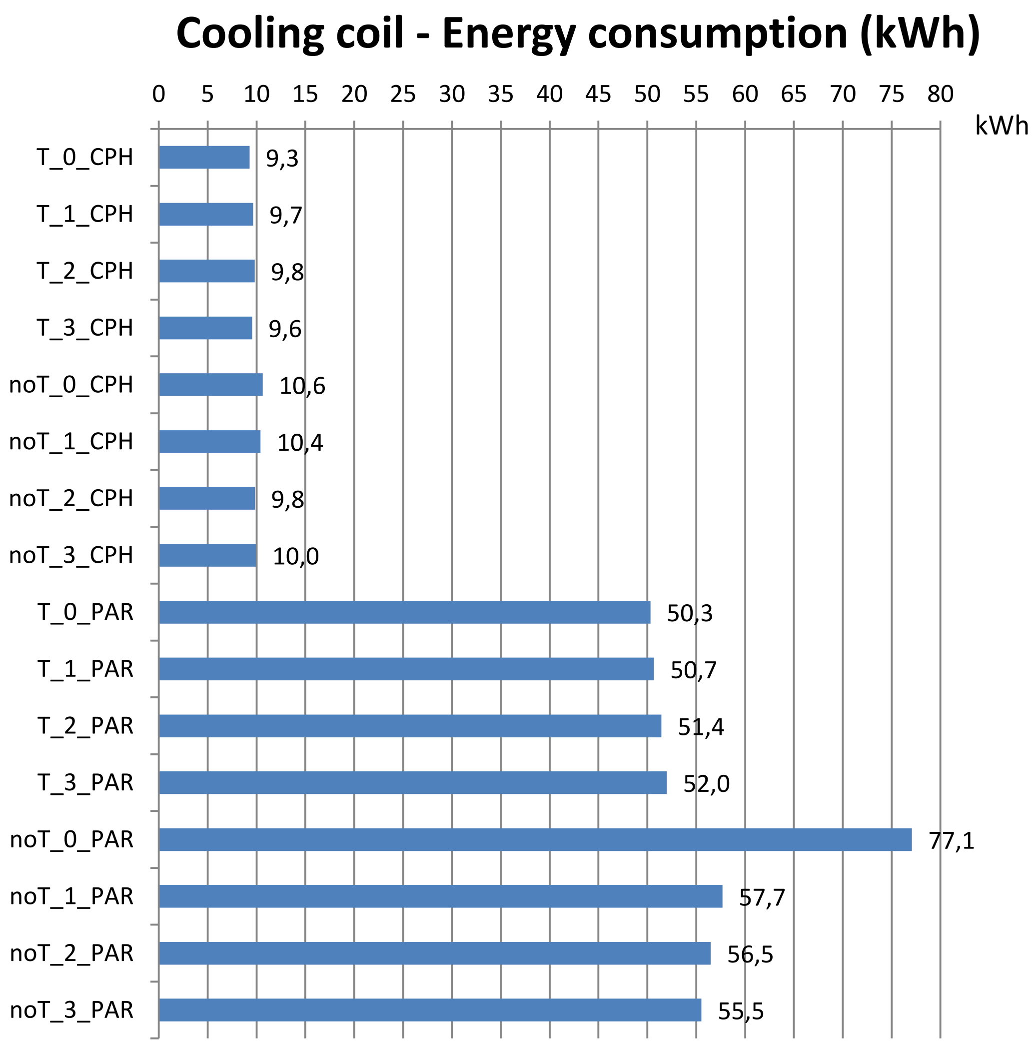


Figure 80: Cooling coil – Energy consumption (summer season – kWh)

Results show that in "T" cases the consumption is lower than in "noT" cases, despite the difference is narrow (except for the case "noT_0_PAR"). This is due to slightly higher indoor temperatures in case the system is not equipped with the tank, thus reducing the use of the heat recovery system and increasing the direct use of outdoor "fresh" air, not pre-cooled before entering the cooling coil.

The same consideration can be extended to explain the influence thermal mass has on the cooling coil energy consumption in "T" cases: in fact, it is observed that in buildings with higher thermal mass, the indoor temperature in summer-time is higher.

9.3.5 Embedded system

9.3.5.1 Embedded system pumps operational hours

This subchapter presents the results for the pumps feeding the radiant system. Figure 81 shows the results recorded, valid for both the pump feeding ground floor loops and the one feeding the first floor loop, which are activated simultaneously.

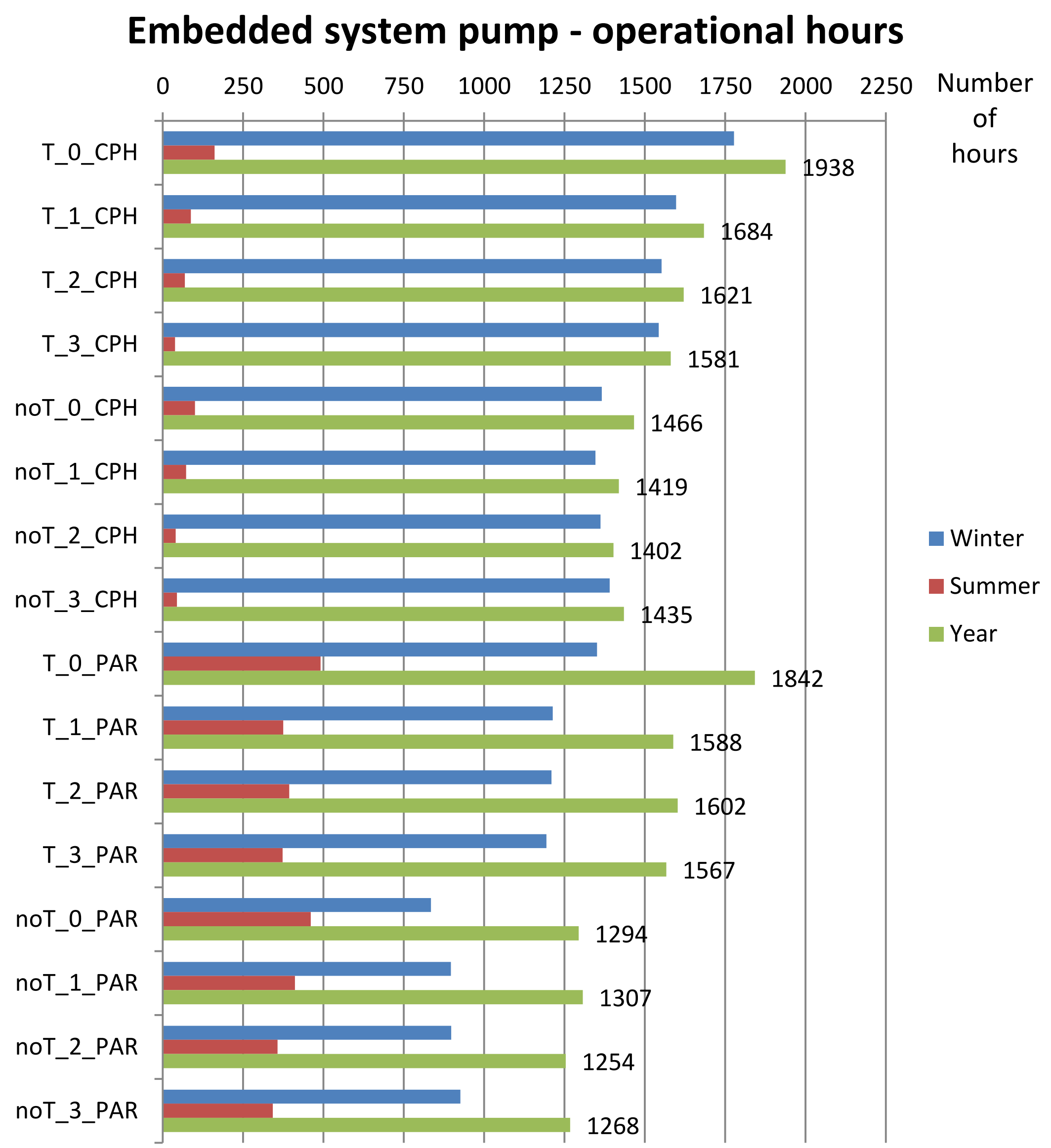


Figure 81: Embedded system pump – operational hours – data labels are referred to the yearly overall operational hours

Figure 81 highlights differences between the cases including the water storage tank and the ones without it. Despite the same control strategy, the number of hours where the pump is activated is higher in case the system is equipped with the water

storage tank. This behavior can be explained (for the cases involving Copenhagen climate) considering the results presented in Table 46: the exclusion of the tank from the model reduces the amount of operational hours because the radiant system can rely on higher temperature differences between supply flow and return flow. Moreover, the system can count on higher thermal mass values, which means that the capability of the system to accumulate heat is higher. The combination of these aspects results in lower activity periods for Copenhagen cases. Regarding Paris results, the difference between the temperature drop (σ) in “T” cases and “noT” cases is smaller but despite this a similar effect is observed probably due only to the effect of higher thermal mass.
The addition of thermal mass results in smaller operative periods for the radiant floor pump for the reasons mentioned above. Results allow highlighting the trend more clearly with respect to the cooling season. Despite higher indoor temperatures, the system is affected by the higher amount of heat that can be stored inside the structures, resulting in fewer operational hours (Figure 58).
Copenhagen cases present higher amounts of operational hours in winter-time and fewer operative hours in summer-time, due to the climate´s influence.

9.3.5.2 Embedded system pump operational hours – Summer night

Since one of the most innovative concept introduced in this report is the night-time radiative cooling, the number of hours in which the embedded system pump is active during summer nights is evaluated, in each considered case.
Results are shown in Figure 82 for every summer month (it should be kept in mind that winter season has been extended and it includes day within 1st January – 20th May and within 11th September – 31st December). Results indicate with the labels “May” or “September” include the remaining days of the month.

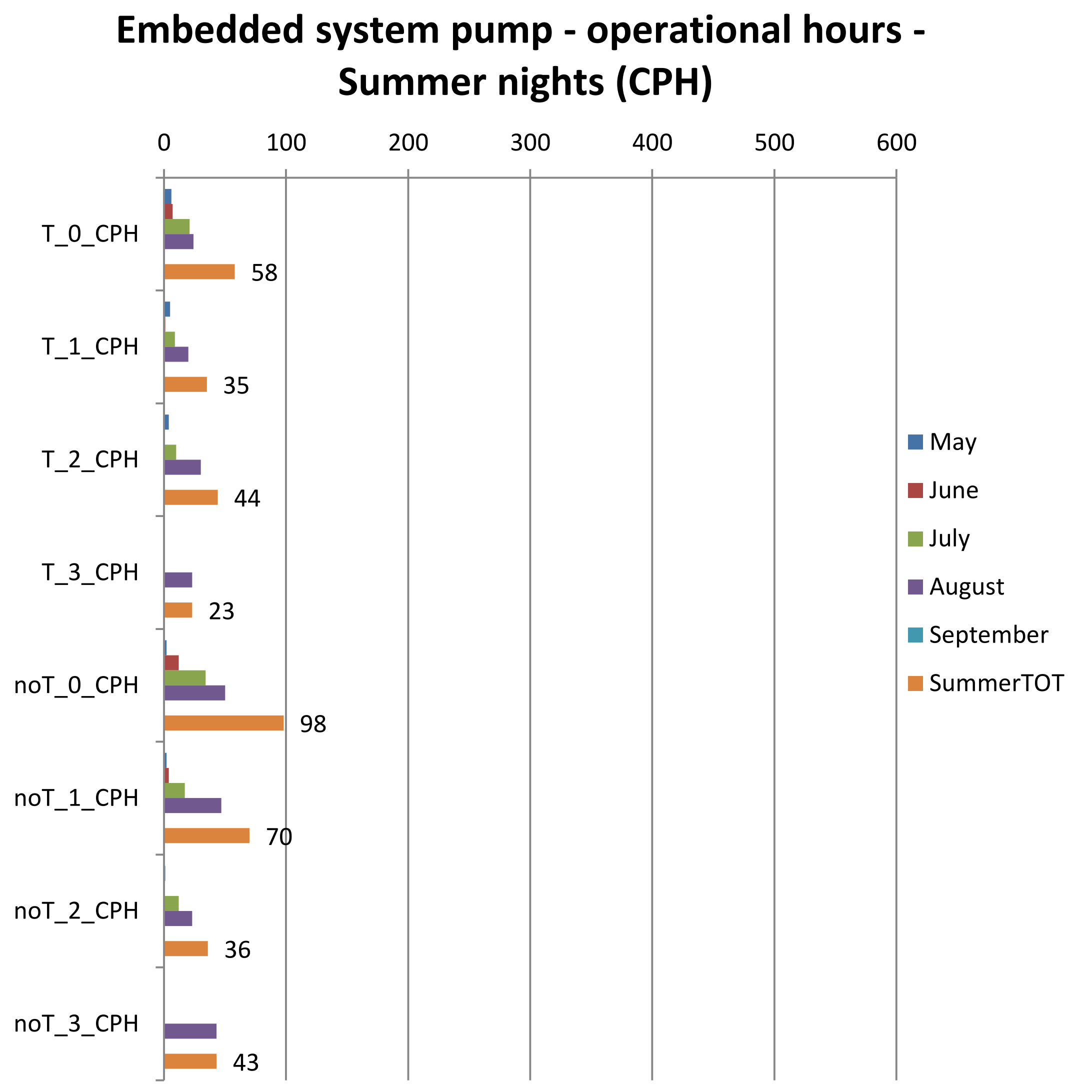


Figure 82: Operation hours during summer nights of the embedded system pump – Copenhagen. Data labels are referred to the overall summer values.

Regardless of the presence of the water tank, it is observed that the number of activity hours decreases with higher thermal mass (despite unexpected results for “T_2_CPH” and “noT_3_CPH”). This behavior reflects the one observed in subchapter 9.3.5.1 for the overall operational hours and it is attributable to the fact that increasing the thermal capacity of the structure corresponds to increase the possibility to store heat that does not have to be fully removed.
The absence of the water storage tank is beneficial in order to exploit in the best way the sky as a free source of cooling.

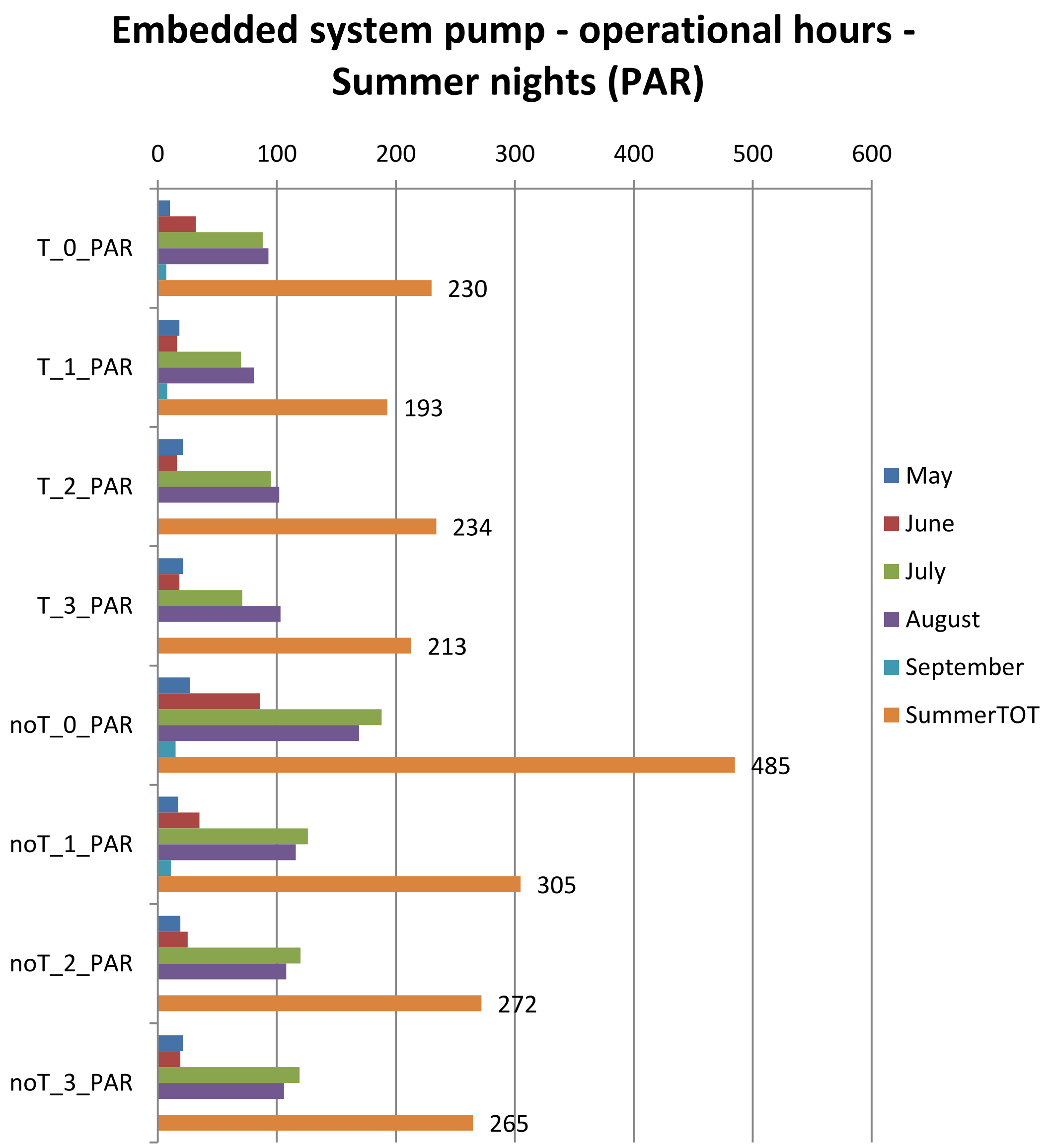


Figure 83: Operation hours during summer nights of the embedded system pump – Paris. Data labels are referred to the yearly to the overall summer values

Regarding Paris climate, a similar trend is recorded for "noT" cases, while oscillating results are recorded for "T" cases, probably due to the interaction between the water storage tank and the thermal mass.
The overall amount of operational hours is higher than the Copenhagen cases.

9.3.5.3 Embedded system pumps energy consumption

Results for the ground floor pump and for the first floor pump are presented in the following figures.

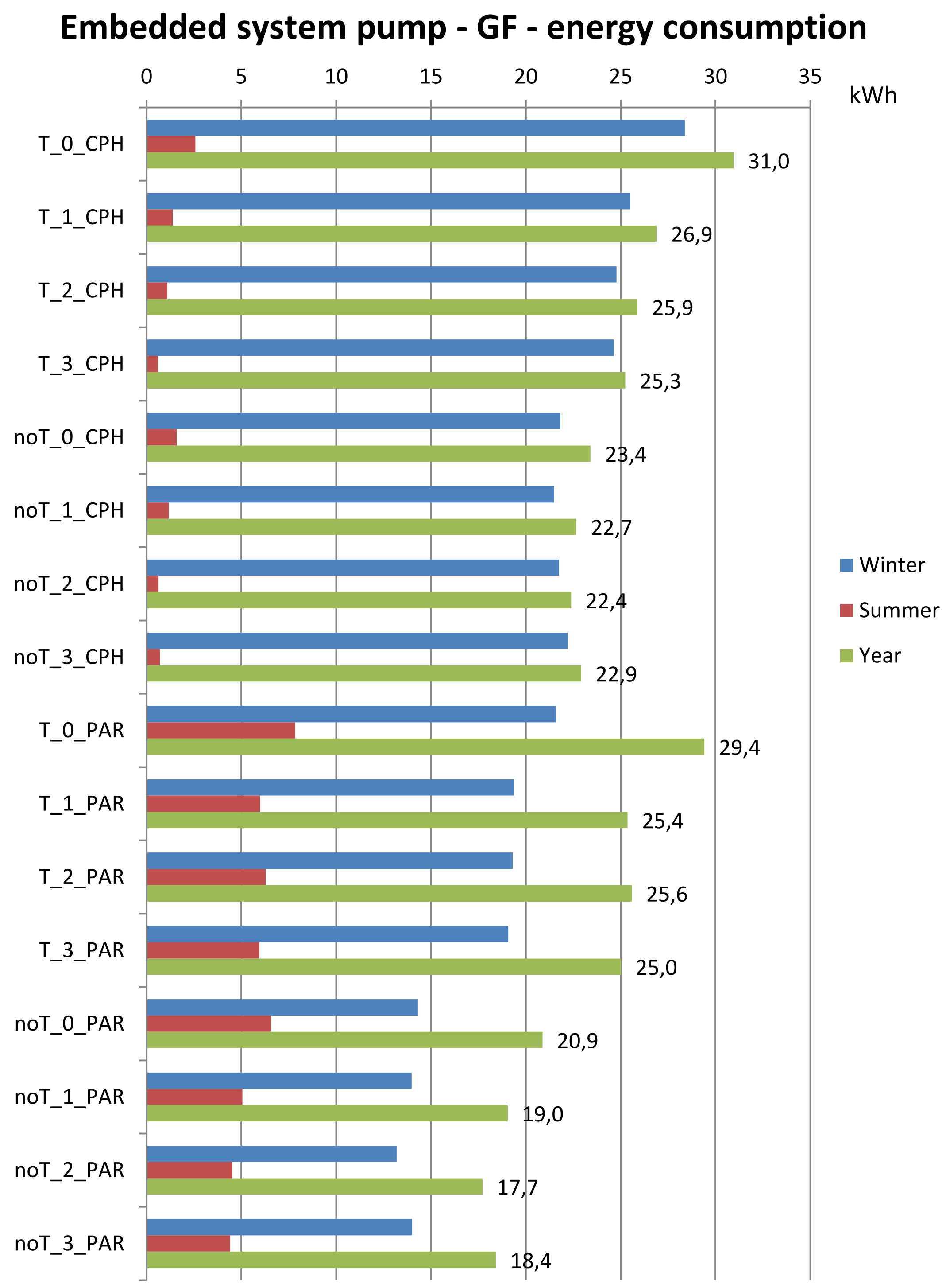


Figure 84: Embedded system pump – Ground floor – energy consumption (kWh). Data labels are referred to the yearly overall energy consumption

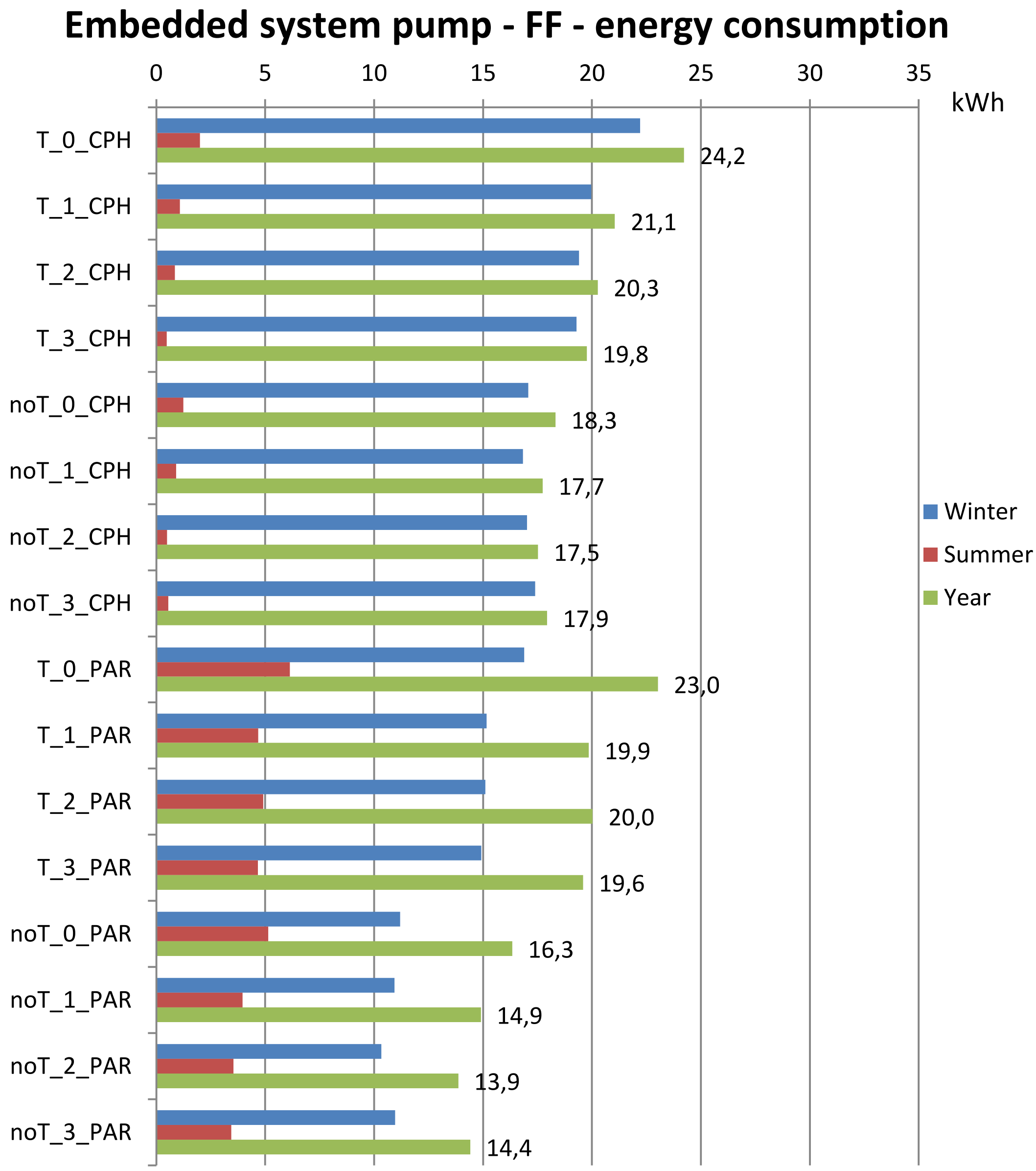


Figure 85: Embedded system pump – First floor – Energy consumption (kWh). Data labels are referred to the yearly overall energy consumption.

As shown in Figure 84 and Figure 85, the tendency is obviously identical to the one described in subchapter 9.3.5.1 regarding the activity hours of the pump.

9.3.6 Primary energy consumptions

Primary energy is calculated to compare overall energy consumption results.
Each device contributes in the final amount and each contribution needs to be converted (if necessary) in electrical energy in order to be multiplied with the Primary Energy Factor (PEF). Therefore, the final sum involves pumps´ energy consumptions (electrical energy), DHW tank auxiliary heater energy consumption (assumed to be a simple electrical resistance), heat pump energy consumption (including compressor, blower and controllers) and ventilation system energy consumption. Regarding ventilation system, the energy consumption of the heating and humidifier coil has been accounted as electrical (assuming the presence of a simple electrical resistance to heat the air flow), while for the cooling coil´s energy consumption has been divided by 2.5 in order to obtain the corresponding electrical energy consumption (11116 Sustainable Buildings, 2012). It should be kept in mind that assumptions (especially regarding ventilation system) might be too strong; further evaluations involving the actual devices of the house could lead to more accurate results.
Two different scenarios have been considered for the primary energy consumption evaluation, according the corresponding national regulations: Copenhagen 2013 and Paris 2013. PEFs for electrical energy are 2.5 for Copenhagen 2013 (Kurnitski, et al., 2011), 2.58 for Paris 2013 (Molenbroek, Stricker, & Boermans, 2011). Hereinafter are the results.

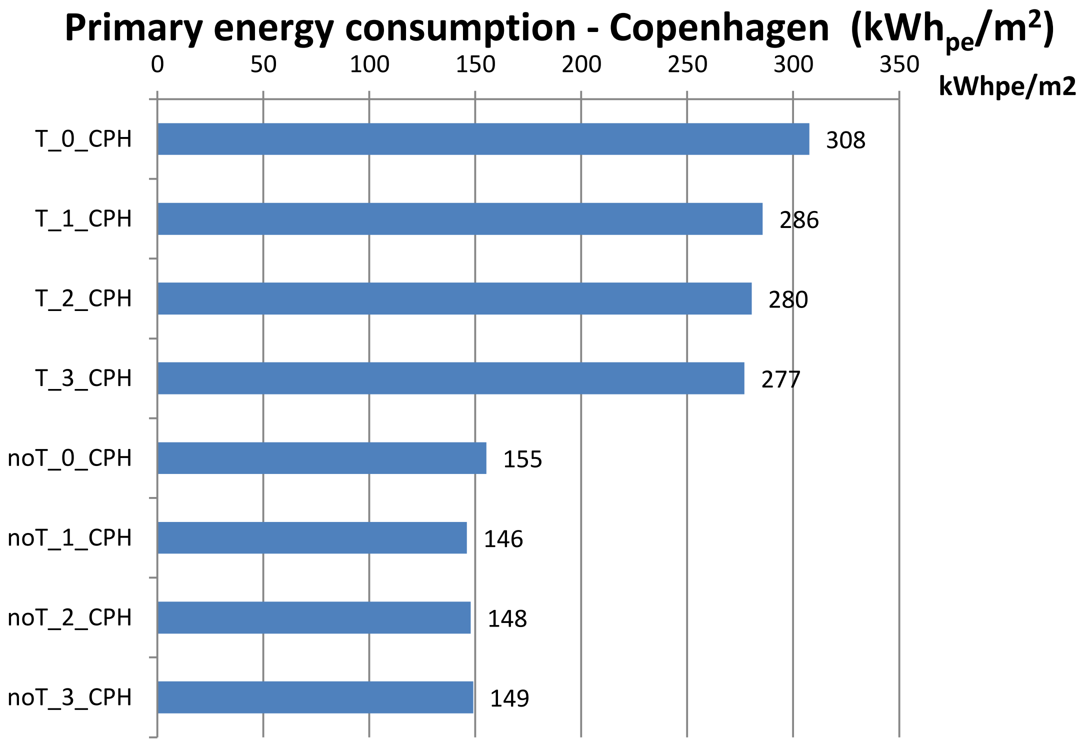


Figure 86: Primary energy consumption - Copenhagen (kWh$_{pe}$/m^2)

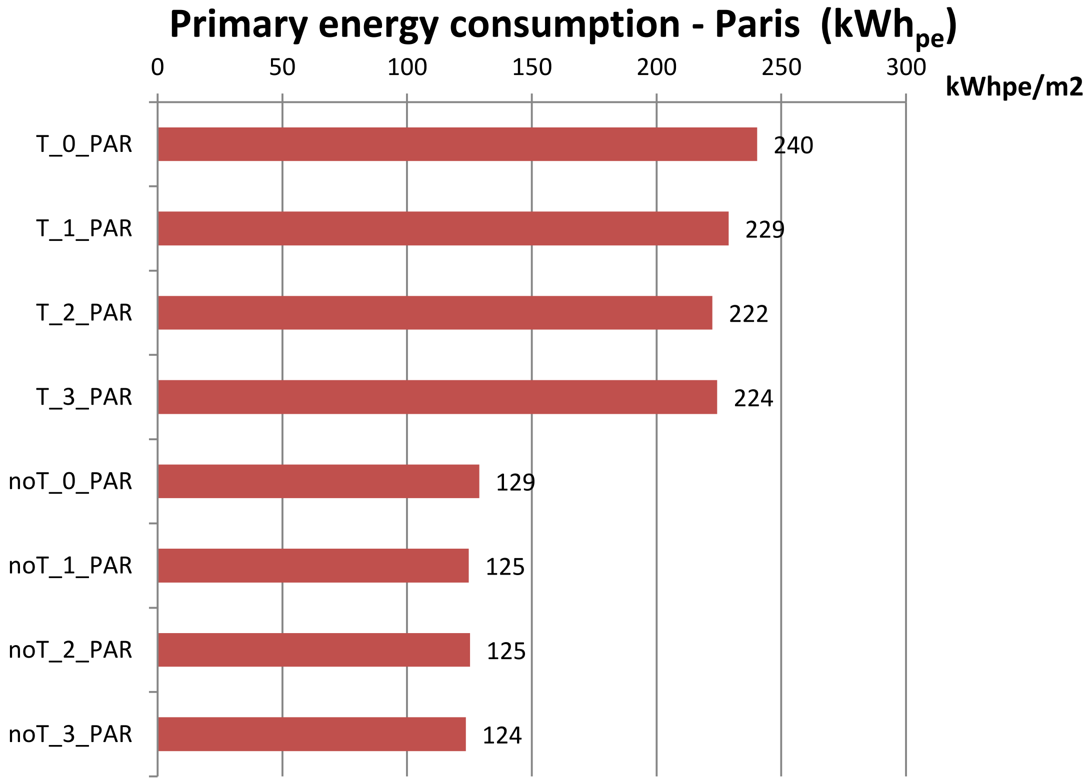


Figure 87: Primary energy consumption – Paris (kWh_{pe}/m^2)

Figure 86 involves PEF equal to 2.5 (Kurnitski, et al., 2011), while Figure 87 involves PEF equal to 2.58 (Molenbroek, Stricker, & Boermans, 2011).
It is shown that, since the presence of the water storage tank is associated with a more frequent use of the heat pump, it plays an important role in the calculation of the primary energy consumptions.
Results show that thermal mass is beneficial in order to decrease the amount of primary energy consumed.
Paris cases are less energy-consuming than Copenhagen cases.

It should be kept in mind that the values obtained are unexpectably high. This is mainly due to the heat pump and its operation. In case the tank is included in the simulations in fact, the device is in charge of conditioning a 800 liters water storage tank regardless of the real needs of the house, resulting in longer running periods and more frequent activation. Moreover, the heat pump is the device characterized by the highest grade of uncertainty and therefore just the capacity was adjusted, while TRNSYS default values were used for the other parameters. The heat pump´s capacity was adjusted with respect to the cases in which the system is equipped with the water storage tank and it´s highly likely that this leads to an oversized capacity. This explains why the results are still too high in case the tank is excluded.

The control strategy also plays a significant role: it had to be simplified in order to help TRNSYS in the calculation process, despite this leads to less realistic and unexpected results.
Therefore, it is possible to conclude that the simulations involve particular requirementes demanded by the SDE organization, thus leading to the high values of primary energy consumptions shown in the previous figures.
Further investigations should involve the realistic parameters of the heat pump and more accurate control strategy regarding the water storage tank.

10. DISCUSSION

Conclusions chapter improves and summarizes the discussions of the results presented in the previous subchapters. The main objective of this report is to evaluate the influence of the thermal mass when combined with PV/T collectors, with respect to two different systems, one including a water buffer tank while in the other one the tank is removed. The evaluation of thermal mass coupled with PV/Ts involves also the effect of the night-radiative cooling. The final evaluation is based on different parameters, belonging to three main categories: indoor conditions, number of operational hours and energy consumption. The comparison will highlight the influence of thermal mass on the mentioned parameters, the influence of the presence of the buffer tank, distinguishing between Copenhagen climate and Paris climate.

Nowadays, regarding the energy performance of a building most of the attention is focused on its energy consumption.

One of the purposes of the analysis is to highlight which component affects the energy performance of the dwelling in the most significant way. This was found to be the water storage tank. On one hand, it influences the system, resulting in better indoor comfort for the occupants, shorter periods of activity for the pump driving the PV/Ts (and therefore corresponding lower energy consumptions), generally higher COP values for the heat pump, lower DHW tank auxiliary heater energy consumption, lower consumptions for the ventilation system in both seasons; on the other hand though, systems including the water storage tank underline a lower exploitation of the sky as a "free-source" of cooling but, above all, higher utilization of the heat pump (and of the pump which is driven by). The frequent use of the heat pump results in much higher energy consumption and, since the energy consumed by the heat pump is accounted as electrical energy, this results in even higher primary energy demands, regardless of the primary energy factor considered for electricity. Though, it is necessary to keep in mind that at the time the TRNSYS model was completed, no detailed features about the heat pump were established. Therefore, only the heating and cooling capacities were adjusted in the TRNSYS model, thus maintaining the default values for most of the other required parameters. For this reason, previous considerations would deserve further investigations involving parameters as close as possible to the ones of actual heat pump that will be installed in the house.

Thermal mass (and its corresponding thermal capacity) is one of the most important and crucial points of this report. It is able to play a relevant role in the energy performance of the building and therefore it is important to stress its influence in a deeper way. Thermal mass, as expected, generally increases the quality of the indoor environment in terms of percentages within EN 15251 operative temperature´s comfort categories for residential buildings. It is observed though that the combination of building´s thermal mass and water tank´s thermal mass results in smoother trends, for both of the climates considered. The difference between Category I percentage and Category II percentage is higher in case the water storage tank is excluded. Therefore, it is possible to state that higher thermal capacity of the building results in improved indoor conditions in each of the considered cases, even though the best results derive from the combination of building´s thermal mass and the buffer tank.

Another effect of increased thermal capacity of the building consists of having warmer structures. In fact, higher thermal mass corresponds to higher capacity to absorb and store heat, principally due to internal gains and occupants activities. This affects the system from several points of view. Despite this trend would deserve further investigations, it is possible to hypothesize that for this reason embedded system supply temperatures are higher during the heating season and lower during the cooling season in building with higher thermal capacity. Having warmer structures in summer-time means that the system has to extract more heat and therefore this is achieved with the lower supply temperature. The presence of the tank also plays a relevant role: in fact, its presence permits to store warmer water, thus resulting in higher supply temperature during the heating season for both Copenhagen and Paris climates. It is also observed that during the cooling season, the presence of tank affects the system resulting in higher supply temperatures in higher thermal mass structures in case the tank is present, while the opposite is observed in case the tank is absent.

Building´s thermal mass affects the electrical and thermal performances of the PV/Ts in a negligible way while the buffer tank has a significant role. This is due to the control strategy: in fact, in case the tank is not included, the possibility during the heating season for the PV/Ts to feed directly the radiant floor is implemented, thus increasing the number of activity hours of the solar collectors for thermal purposes. This results in slightly higher electrical efficiencies (because the water circulating is able to decrease the temperature of the solar cells, thus increasing their electrical efficiency) and higher thermal efficiencies as well (due to the higher number of hours of utilization).

For the same reason it is observed that the presence of the storage tank influences the activation of the pump driving the PV/Ts, which is more frequent in case it is not included. Furthermore, “noT” cases present a certain amount of operational hours for the pump during winter, which is almost negligible in summer-season. Building´s thermal mass also affects the operation of the pump driving the PV/Ts. In case the storage tank is present, it is observed that generally the amount of operational hours decreases with higher thermal mass, while in case the tank is absent it increases with higher thermal mass. Regarding the “T” cases, the behaviour can be explained considering that the systems including higher thermal capacity are characterised by higher water temperatures; therefore the PV/Ts pump needs to be activated for shorter periods to provide the system with the necessary heat. The combinations in which the water storage tank is avoided would deserve further investigations.

Regarding the influence of the thermal mass (either building´s thermal mass or buffer tank´s thermal mass) has on the operation of the heat pump, first it is observed that the presence of the water storage tank adds a relevant amount of water that needs to be conditioned. Therefore, the activity of the heat pump is much more frequent in “T” case rather than “noT” cases, thus resulting in higher energy consumptions in case the water storage tank is included. With respect to the building´s thermal mass though, other trends are noticed: results show that the activity of the heat pump (and its corresponding energy consumption) decreases with increased thermal mass in “T” case, while the opposite is observed in “noT” cases. Regarding the cases where the tank is included, the explanation presented in the previous paragraph for the pump driving the PV/Ts could be extended to the heat pump (and the pump it is driven by). Regarding the cases in which the storage tank is absent, it is observed that the pump driving the heat pump is activated for longer periods with increased thermal mass, while heat pump energy consumption decreases, probably meaning that the heat pump is activated for a larger amount of hours but at partial load; this could be due to the faster response of the systems without the tank. However, cases where the storage tank is not included would deserve further investigations. Heat pump´s COP values are higher in case the water storage tank is included, due to the presence of a stable heat sink to which it can release the heat absorbed on the evaporator side from outside air. In case the water storage tank is considered, building´s thermal mass has a negative influence on the COP values because it corresponds to higher temperatures inside the buffer tank, higher temperature difference between the heat pump´s evaporation temperature and condensation temperature and therefore lower efficiency. This statement can be extended to the cases not-including the water storage tank.

DHW tank auxiliary heater behaves according to the considerations stated so far. In case the tank is considered, the system is characterised by higher temperatures and therefore the auxiliary heater needs to be activated for shorter periods. Winter contribution is almost negligible, meaning that in winter-time the combination of PV/Ts and heat pump is able to provide all the required energy for DHW purposes. The influence of building´s thermal mass on the auxiliary heater consumption is not clear and simulations show different trends for Copenhagen and Paris.

Regarding ventilation system consumption, it is higher in case the water storage tank is avoided because this results in lower and more variable indoor temperatures and therefore in lower possibility to recover heat in the heat recovery system. The influence building´s thermal mass has on the energy consumption of the ventilation system is not clear and seems to depend on the presence of the tank.

Regarding embedded system, it is possible to observe that the presence of the water storage tank results in higher amount of operational hours and therefore energy consumption. Furthermore, increased building´s thermal mass results in fewer activity hours for the pumps as expected, generally both during the heating season and the cooling season. Particular attention has been given to the operation of the radiant floor during summer-nights, where the sky is used as “free-source” of cooling through the night radiative cooling. Results highlight similar trends compared to the ones described above in case the system is not equipped with the tank, while in case the tank is present the behaviour is less clear and therefore it is more difficult to draw a conclusion. It is noticed that the amount of operational hours tend to decrease with higher thermal mass in “noT” cases: this is due to the capacity of the building to store a greater quantity of heat, thus reducing the amount of required activity hours in order to meet the control strategy requirements. The amount of operational hours is higher in Paris than in Copenhagen, which is probably due to higher outdoor temperatures and solar gains. Finally, it is observed that the nocturnal operation of the radiant floor is higher in the warmer months (July and August).

Thermal mass appears to be beneficial in order to reduce the primary energy consumption of the building, both in “T” cases (-11% for Copenhagen and -7% for Paris) and “noT” cases (-4% for both climates). Since on one hand increased thermal mass results in saving from the energy consumption point of view but on the other hand in higher investment costs, economic analysis should be carried out to weight the two effects.

Among the goals of this report regarding the evaluation of the performance of the building, it is possible to add the analysis focused on the night-time radiative cooling. This technology being relatively new, only a few references can be found in literature: (Eicker & Dalibard, 2011) and (Meir, Rekstad, & Lovvik, 2003) for example. The studies presented in those articles prove that the night radiative cooling involving PV/Ts is able to provide a cooling power approximately in the range 40÷65 W/m^2. Subchapter 9.3.1.5 presented the results for the nocturnal radiative losses in the PV/T collectors. Despite the trend is not completely evident in “noT” cases for Copenhagen, it is possible to state that generally the radiative losses increase in the buildings with higher thermal capacity. Since the amount of heat stored is higher, the structure has to be (at least partially) discharged involving higher radiative losses towards the sky. In the cases in which the system does not include a water storage tank, results underline higher radiative losses. This is due to the control strategy, which in case the system is equipped with the water storage tank aims to cool down a 800 litres tank, while in “noT” cases, it is indoor operative temperature-dependent. Thus, since the system has to cool the structure first in order to meet the indoor temperature requirements, the resulting cooling power needed is higher. Values are, on average, higher for Copenhagen cases, due to lower sky temperatures.

Outcomes regarding the nocturnal cooling highlight a correspondence between the number of operational hours of the pump driving the PV/Ts and the PV/Ts losses. In structures where the possibility of exploiting the sky “free heat-sink” is higher (which means higher nocturnal losses towards the sky), the pump feeding the PV/Ts is activated for fewer hours. This benefit is observed in structures with increased thermal mass. The share of the overall cooling energy demand that can be covered by the night radiative cooling increases in buildings with higher thermal mass in case the tank is present, while the opposite is noticed in case the tank is not present.

All the results discussed above are outcomes of TRNSYS simulation software, therefore corresponding to a certain grade of precision depending on the calculation capability of the software. Moreover, it should be kept in mind that simulations and calculations in general involve a certain number of assumptions due to lacks of information or simplifications based on the program needs. It has to be kept in mind that *Embrace* is still a “work-in-progress” project, therefore many components have not been defined yet, as well as other design features. Once every feature is defined in detail, minor corrections to the system´s parameters and to the design implementation of the house will lead to a more precise evaluation for the SDE 14 purposes.

11. CONCLUSION

The combination of building´s thermal mass and PV/Ts appears to be beneficial from different points of view. Firstly, the increase in the thermal capacity of the building results in better indoor conditions for the occupants. Secondly, this corresponds also to relevantly lower energy consumptions. Finally, the utilization of buildings with high thermal capacity allows exploiting more productively the night radiative cooling; in the considered climates this "free-cooling" strategy is able to provide most of cooling demand of the house, limiting the periods in which the heat pump is activated. The implementation of the system presented appears to be more beneficial (in terms of contribution in the total cooling demand) in mid-European climates rather than Nordic climates in case of heavy structures, while the opposite can be stated in case of lighter structures.

Further conclusions can be drawn regarding the presence of the water storage tank. The main advantage of equipping the system with a buffer tank consists of better indoor comfort conditions with respect to the correspondent case in which the tank is absent. On the other hand though, the amount of water contained in the water requires the heat pump to operate for much longer periods, resulting in much higher energy consumptions. The effect is slightly smoothed in buildings with higher thermal mass. However, it is possible to observe that the tank is replaceable with increased thermal mass, guaranteeing satisfying indoor conditions. Therefore, building´s thermal mass can be increased in order to avoid the presence of the water storage, thus resulting in much lower energy consumptions and good indoor conditions (even though the presence of the tank can ensure better indoor conditions for the occupants). Therefore, in case slightly poorer indoor comfort conditions are considered acceptable, the water storage tank can be avoided, with benefits from the energy consumption point of view.

12. FUTURE OUTLOOK

The evaluation of the combination between photovoltaic/thermal collectors and building´s thermal mass could be improved by further simulations or investigations. Hereinafter is presented a list of possible improvements/modifications that could be added in order to obtain deeper results.

- Reduction of the size of the water storage tank instead of its complete elimination and corresponding economic constraints and subsequent applicability of thermal mass
- Control strategy of the embedded system based on the surface temperature instead of the indoor air temperature
- Reduction of the size of the heat pump and the consequent behavior of the system
- Combination of PV/Ts and water-to-water heat pump
- Implementation of concrete-base radiant floors (wet or dry systems) or TABS and subsequent evaluation of the use of the same (or slightly different) supply temperature in heating and cooling seasons
- Investigation of the effective activated thickness in the concrete layer and the dynamic behavior of the structure in the process of absorption/release heat.
- Implementation of PCM layers inside the structure in case the concrete is not implementable due to structural constraints.
- Analysis of a different climate (southern-European)
- Weighting/optimization of indoor-climate/energy-consumption can be considered.

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14. APPENDIX

14.1 Load Calculations

14.1.1 General data

PARAMETERS

Type	2 people family house	
Location	Copenhagen/Paris	

DIMENSIONS

Orientation	Area (m^2)	U-value (W/m^2K)
South (vertical)	15.9	0.1
South (26° slope)	27.8	0.1
West (facing outside)	41.2	0.1
North (vertical)	20.2	0.1
North (62° slope)	13.5	0.1
East (facing outside)	7.6	0.1
East (facing inside)	28.0	0.1
Doors	13.2	0.7
Roof	18.0	0.1

Floor area	42.8	m^2
Room height	3	m

INFILTRATION

Passive house	0.1	1/h

14.1.2 Lighting

Ceiling (3 W/m^2)	128.5	W
Desk	0	W
K_1 (use factor)	1	
K_2 (allowance factor)	1	

14.1.3 Sensory and chemical loads

SENSORY POLLUTION LOADS

Floor	0.05	olf/m^2_{floor}
Walls & Ceiling	0.05	olf/m^2_{floor}
Tables & Chairs	0.5	olf
PC	0.25	olf/PC
Total sensory pollution load	14.1	olf
Sensory pollution loads	0.1	m^2/olf

CHEMICAL POLLUTION LOADS

Floor	0.001	$\mu g/s \cdot m^2_{floor}$
Wall & Ceiling	0.001	$\mu g/s \cdot m^2_{floor}$
Tables & Chairs	0.015	µg/s
PC	0.5	µg/s·PC
Total Chemical Pollution loads	2.23	µg/s
TVOC concentration outdoor	0	$\mu g/m^3$

POLLUTION LOADS CAUSED BY OCCUPANTS

Sensory	1	olf/px
CO_2	19	l/h·px
Water vapour	50	g/h·px

14.1.4 Heat loss to the ground

Copenhagen - Winter

U_f	0.3	W/m^2K	
A_f	42.8	m^2	
Perimeter (inner)	27.2	m	
B'	3.15		
$\theta_{int.i}$	22	°C	
$\theta_{m.e}$	9.3	°C	
θ_e	-12	°C	
f_{g2}	0.355		
f_{g1}	1.45		predefined value D.4.3
G_w	1		distance water-floor >1 m
$U_{equival.k}$	0.2	W/m^2K	

$H_{T.ig}$	4.403	W/K
Heat loss to the ground	0.145	kW

Paris - Winter

U_f	0.3	W/m^2K	
A_f	42.8269	m^2	
Perimeter (inner)	27.2	m	
B'	3.15		
$\theta_{int.i}$	21	°C	
$\theta_{m.e}$	12.7	°C	
θ_e	-5.9	°C	
f_{g2}	0.308550186		
f_{g1}	1.45		predefined value D.4.3
G_w	1		distance water-floor >1 m
$U_{equival.k}$	0.2	W/m^2K	
$H_{T.ig}$	3.83	W/K	
Heat loss to the ground	0.103	kW	

Copenhagen - Summer

U_f	0.3	W/m^2K	
A_f	42.8269	m^2	
Perimeter (inner)	27.2	m	
B'	3.15		
$\theta_{int.i}$	25.5	°C	
$\theta_{m.e}$	9.3	°C	
θ_e	30	°C	
f_{g2}	-3.6		
f_{g1}	1.45		predefined value D.4.3
G_w	1		distance water-floor >1 m
$U_{equival.k}$	0.2	W/m^2K	
$H_{T.ig}$	-44.71	W/K	
Heat loss to the ground	0.201	kW	

Paris – Summer

U_f	0.3	W/m^2K	
A_f	42.8269	m^2	
Perimeter (inner)	27.2	m	
B'	3.15		
$\theta_{int.i}$	25.5	°C	
$\theta_{m.e}$	12.7	°C	
θ_e	30.9	°C	
f_{g2}	-2.37037037		
f_{g1}	1.45		predefined value D.4.3
G_w	1		distance water-floor >1 m
$U_{equival.k}$	0.2	W/m^2K	
$H_{T.ig}$	-29.43	W/K	
Heat loss to the ground	0.16	kW	

14.1.5 Load calculations – weather shield not considered

Heating load – "Maximum" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	-0.15	kW
1	Window (no sun radiation considered)	-0.18	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.81	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.15	kW
8	Infiltration	-0.15	kW
	Balance	-1.42	kW
	I.r - I.in	-23.38	kJ/kg
	HEATING LOAD	**-33.14**	**W/m^2**

Heating load – "Average" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	-0.09	kW
1	Window (no sun radiation considered)	-0.11	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.48	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.15	kW
8	Infiltration	-0.09	kW
	Balance	-0.90	kW
	I.r - I.in	-21.91	kJ/kg
	HEATING LOAD	**-21.06**	**W/m^2**

Heating load – "Maximum" conditions – Paris

	HEAT LOAD		
0	Ventilation	-0.12	kW
1	Window (no sun radiation considered)	-0.15	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.66	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.10	kW
8	Infiltration	-0.12	kW
	Balance	-1.19	kW
	I.r - I.in	-28.11	kJ/kg
	HEATING LOAD	**-27.80**	**W/m^2**

Heating load – "Average" conditions – Paris

	HEAT LOAD		
0	Ventilation	-0.09	kW
1	Window (no sun radiation considered)	-0.11	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.50	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.10	kW
8	Infiltration	-0.09	kW
	Balance	-0.94	kW
	I.r - I.in	-22.65	kJ/kg
	HEATING LOAD	**-21.85**	**W/m^2**

Cooling load – "Maximum" conditions –Copenhagen

	HEAT LOAD		
0	Ventilation	0.02	kW
1	Window (sun radiation considered)	0.59	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	0.13	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.20	kW
8	Infiltration	0.02	kW
	Balance	1.29	kW
	I.r - I.in	34.61	kJ/kg
	COOLING LOAD	**30.17**	**W/m^2**

Cooling load – "Average" conditions –Copenhagen

	HEAT LOAD		
0	Ventilation	-0.03	kW
1	Window (sun radiation considered)	0.24	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.15	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.20	kW
8	Infiltration	-0.03	kW
	Balance	0.41	kW
	I.r - I.in	10.44	kJ/kg
	COOLING LOAD	**9.48**	**W/m^2**

Cooling load – “Maximum” conditions –Paris

	HEAT LOAD		
0	Ventilation	0.03	kW
1	Window (sun radiation considered)	0.62	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	0.15	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.16	kW
8	Infiltration	0.03	kW
	Balance	1.40	kW
	I.r - I.in	37.51	kJ/kg
	COOLING LOAD	**32.61**	**W/m^2**

Cooling load – “Average” conditions –Paris

	HEAT LOAD		
0	Ventilation	-0.02	kW
1	Window (sun radiation considered)	0.25	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.09	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.16	kW
8	Infiltration	-0.02	kW
	Balance	0.55	kW
	I.r - I.in	14.15	kJ/kg
	COOLING LOAD	**12.73**	**W/m^2**

14.1.6 Load calculations – weather shield considered

Heating load – "Maximum" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	-0.146	kW
1	Window (no sun radiation considered)	-0.18	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.81	kW
6	Thermal dif. Roof	0	kW
7	Thermal dif. Ground	-0.15	kW
8	Infiltration	-0.15	kW
	Balance	-1.42	kW
	I.r - I.in	-23.4	kJ/kg
	HEATING LOAD	**-33.14**	**W/m^2**

Heating load – "Average" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	-0.09	kW
1	Window (no sun radiation considered)	-0.11	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.48	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.15	kW
8	Infiltration	-0.09	kW
	Balance	-0.90	kW
	I.r - I.in	-21.91	kJ/kg
	HEATING LOAD	**-21.06**	**W/m^2**

Heating load – "Maximum" conditions – Paris

	HEAT LOAD		
0	Ventilation	-0.12	kW
1	Window (no sun radiation considered)	-0.15	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.66	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.10	kW
8	Infiltration	-0.12	kW
	Balance	-1.19	kW
	I.r - I.in	-28.11	kJ/kg
	HEATING LOAD	**-27.80**	**W/m^2**

Heating load – "Average" conditions – Paris

	HEAT LOAD		
0	Ventilation	-0.09	kW
1	Window (no sun radiation considered)	-0.11	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.50	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.10	kW
8	Infiltration	-0.09	kW
	Balance	-0.94	kW
	I.r - I.in	-22.65	kJ/kg
	HEATING LOAD	**-21.85**	**W/m^2**

Cooling load – "Maximum" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	0.03	kW
1	Window (sun radiation considered)	0.43	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	0.17	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.20	kW
8	Infiltration	0.03	kW
	Balance	1.19	kW
	I.r - I.in	32.16	kJ/kg
	COOLING LOAD	**27.86**	**W/m^2**

Cooling load – "Average" conditions – Copenhagen

	HEAT LOAD		
0	Ventilation	-0.02	kW
1	Window (sun radiation considered)	0.16	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.13	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.20	kW
8	Infiltration	-0.02	kW
	Balance	0.35	kW
	I.r - I.in	9.06	kJ/kg
	COOLING LOAD	**8.20**	**W/m^2**

Cooling load – "Maximum" conditions – Paris

	HEAT LOAD		
0	Ventilation	0.04	kW
1	Window (sun radiation considered)	0.46	kW
2	Equipment	0.46	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	0.23	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.16	kW
8	Infiltration	0.04	kW
	Balance	1.35	kW
	I.r - I.in	36.86	kJ/kg
	COOLING LOAD	**31.62**	**W/m^2**

Cooling load – "Average" conditions – Paris

	HEAT LOAD		
0	Ventilation	-0.01	kW
1	Window (sun radiation considered)	0.17	kW
2	Equipment	0.30	kW
3	Lamps	0.13	kW
4	Occupants	0.15	kW
5	Thermal dif. Wall	-0.07	kW
6	Thermal dif. Roof	0.00	kW
7	Thermal dif. Ground	-0.16	kW
8	Infiltration	-0.01	kW
	Balance	0.50	kW
	I.r - I.in	13.04	kJ/kg
	COOLING LOAD	**11.68**	**W/m^2**

14.1.7 Ventilation

Ventilation rate for sensory pollution

Perceived indoor air quality	1.40	dp
Perceived outdoor air quality	0.00	dp
Ventilation sys effectiveness	1.00	
Q_c	44.88	l/s
	0.04	m^3/s
Taken value	0.04	m^3/s
Q_c taken	161.55	m^3/h
Room volume	128.48	m^3
ACH	1.26	1/h

Heat gain by infiltration (example taken from Cooling load – "average" conditions – Paris)

Heat removed	-0.01	kW
Air density	1.20	kg/m^3
Specific heat of air	1.000.00	J/kg·K
T_{out}-T_{in}	-2.5	K
ACH	0.10	1/h
Air flow rate	0.00	m^3/s

Heat recovery in mechanical ventilation (example taken from Cooling load – "average" conditions – Paris)

T_{out} air before HEX	22.00	°C
T_{in} air before HEX	24.50	°C
Efficiency of HEX	0.80	
Ventilation rate	0.50	1/h
	64.24	m^3/h
	0.02	m^3/s
T_{out} air after HEX	24.00	°C
Heating/Cooling Load	-0.01	kW

Ventilation rate of 0.5 ACH is the minimum requirement as stated in (Danish Building Regulation 2012. BR10., 2010).

14.2 Radiant system

14.2.1 Heat transfer coefficient for MIRAGE simulations

ρ	1000.00	kg/m^3	
D	0.02	m	
v	0.50	m/s	
μ	0.00	Pa·s	(water at 40°C)
cp	4190.00	J/kg·K	
λ	0.60	W/m·K	(water at 40°C)
Re	8000.00		(Dittus-Boelter eq. Re>10000)
Pr	6.98		
Nu	66.35		
f	0.03		
Nu	64.41		
h	2415.51	$W/m^2·K$	(Gnielinski correlation - $3000<Re<5·10^6$)

14.2.2 Radiant system dimensioning – EN1264-2 (2008) - Heating

UNIVERSAL POWER FUNCTION METHOD - TYPE B MODIFIED

Layer 1 thickness has been set as null

B	6.50	W/m^2K			
s_u	0.01	(thickness of the layer above the pipe - thickness of plywood)			
λ_E	0.15	(th. Cond. Of the layer above the pipe - therm cond of plywood)			
s_u/λ_E	0.08				
a_T	1.08	(table A.6 - EN 1264-2)			
T	0.20	m			
m_T	-1.67	(if 0.05<T<0.375 m)			
α	10.80	$W/m^2·K$			
$\lambda_{u.0}$	1.00	W/m·K			
$s_{u.0}$	0.05	m			
a_U	0.80				
b_U	0.50				
s_{WL}	0.00	m			
λ_{WL}	226.00	W/m·K			
k_{WL}	1.82				
$a_{WL.KwL=inf.L=T}$	1.05	(table A.8f)			

$a_{WL.kWL=0.L=T}$	0.44	(table A.8a)			
$a_{WL.L=T}$	1.04				
$k_{WL.L=0}$	0.01	(s_{WL}=0)			
$a_{WL.L=0}$	0.41				
L	0.18	m	(width of the heat conducting plate)		
a_{WL}	1.04				
a_K	0.92	(table A.9)			
$R_{\lambda.B}$	0.00	(because layer 1 has been avoided)			
f	1.20				
a_B	1.00				
k_H	4.39				
q	33.14	W/m^2			
σ	5.00	K			
θ_i	22.00	°C			
$\theta_{V.des}$	32.32	°C			
$\Delta\theta_{H.des}$	7.54	°C			
$\Delta\theta_{V.des}$	10.32	°C			
R_o	0.09				
$R_{u.GF}$	8.70	m^2·K/W	$R_{u.FF}$	3.15	m^2·K/W
$A_{f.GF}$	34.23	m^2	$A_{f.FF}$	5.60	m^2
T_u	9.00				
$mH_{.GF}$	0.06	kg/s	m_{HFF}	0.01	Kg/s
	198.19	kg/h		26.73	Kg/h
$m_{H.tot}$	224.92	kg/h			

14.2.3 Radiant system dimensioning – EN15733-1 (2008) – Heating

THERMAL RESISTANCE ABOVE THE HEAT CONDUCTING LAYER				
R_o	0.08	m^2·K/W		
$R_{con.i}$	0.10	m^2·K/W		
h_i	10.80	W/m^2·K	EN 1264-2	
R_{si}	0.09	m^2·K/W		
Ri	0.27	m^2·K/W		

THERMAL RESISTANCE ON THE BACK-SIDE OF THE HEAT CONDUCTING LAYER				
R_{se} has not been considered. as stated in UNI EN ISO 13370 - 6.2				
w	0.40	m		
λ	2.00	W/m·K		
R_{si}	0.09	m^2·K/W		
R_t	10.40	m^2·K/W	(the layer where the pipes are placed has been considered as made of oak wood)	
d_t	21.39	m		
B´	3.15			
U	0.09	W/m^2·K		
$R_{e.0}$	11.14	m^2·K/W		
d	0.02	m	(deepness of the channel)	
b	0.02	m	(width of the channel)	
T	0.20	m	(pipe spacing)	
ΔR_e	0.08	m^2·K/W		
R_e	10.99	m^2·K/W		
THERMAL RESISTANCE BETWEEN THE HEAT SOURCE AND THE HEAT CONDUCTING LAYER				
L_{WL}	0.18	m	(width of the heat conductive plate)	
s_{WL}	0.00	m	(thickness of the heat conducting layer)	
λ_{WL}	220.00	W/mK	(therm. Cond. Of the heat conducting layer)	
	0.22	W/K	(heat conducting performance)	
l	0.24	m	(characteristic lenght of the fin)	
L_{fin}	0.08	m		
k_{fin}	0.97			
L_G	0.03	m	(gap between the heat conducting plates)	
k_{fin}^{min}	0.95			
k_{CL}	0.85			
R_{CL}	0.05	m^2·K/W		
PIPE COILS				
R_R´	0.10	m^2·K/W		
THERMAL CONTACT RESISTANCE BETWEEN THE HEAT CONDUCTING LAYER AND THE HEATING PIPE				

$R'_{R.cond}$	0.50	$m^2 \cdot K/W$		
RESISTANCE OF THE U-PROFILE OF THE HEAT CONDUCTING DEVICE				
R'_U	0.40	$m^2 \cdot K/W$		
TOTAL THERMAL RESISTANCE BETWEEN THE HEAT SOURCE				
R_{HC}	0.21	$m^2 \cdot K/W$		
DIMENSIONING				
DESIGN TEMPERATURE				
θ_i	22.00	°C		
θ_m	31.03	°C	(Ti-Tm=Ri·qi)	
σ	5.00	K	(hypothesis)	
$\theta_{V.des}$	33.76	°C		
$\theta_{R.des}$	28.76	°C		
$\Delta\theta_{H.des}$	9.03	°C		
$\Delta\theta_{V.des}$	11.76	°C		
$\sigma/\Delta\theta_{H.des}$	0.55			

14.2.4 Radiant system dimensioning – EN1264-2 (2008) – Cooling

$k_{H.floor}$	$R_{\lambda.B}=0$		
B	6.5	W/m^2K	
s_u	0.012	(thickness of the layer above the pipe - thickness of plywood)	
λ_E	0.15	(th. Cond. Of the layer above the pipe - therm cond of plywood)	
s_u/λ_E	0.08		
a_T	1.0778		(table A.6 - EN 1264-2)
T	0.2	m	
m_T	-1.66667		(if 0.05<T<0.375 m)
α	10.8	$W/m^2 \cdot K$	
$\lambda_{u.0}$	1	W/m·K	
$s_{u.0}$	0.045	m	

a_U	0.79721		
b_U	0.5		
s_{WL}	0.001	m	
λ_{WL}	226	W/m·K	
k_{WL}	1.8152		
$a_{WL.KwL=inf.L=T}$	1.05		(table A.8f)
$a_{WL.kWL=0.L=T}$	0.435		(table A.8a)
$a_{WL.L=T}$	1.043536		
$k_{WL.L=0}$	0.0072		(s_{WL}=0)
$a_{WL.L=0}$	0.4094		
L	0.175	m	(width of the heat conducting plate)
a_{WL}	1.044032		
a_K	0.92		(table A.9)
$R_{\lambda.B}$	0		(because layer 1 has been avoided)
f	1.196774		
a_B	1		
$k_{H.floor}$	4.392962		
$k^*_{H.floor}$	**$R_{\lambda.B}$=0.15**		
B	6.5	W/m^{2}K	
s_u	0.012	(thickness of the layer above the pipe - thickness of plywood)	
λ_E	0.15	(th. Cond. Of the layer above the pipe - therm cond of plywood)	
s_u/λ_E	0.08		
a_T	1.0778		(table A.6 - EN 1264-2)
T	0.2	m	
m_T	-1.66667		(if 0.05<T<0.375 m)
α	10.8	W/m^2·K	
$\lambda_{u.0}$	1	W/m·K	
$s_{u.0}$	0.045	m	
a_U	0.79721		
b_U	0.5		
s_{WL}	0.001	m	
λ_{WL}	226	W/m·K	
k_{WL}	1.8152		

$a_{WL.KwL=inf.L=T}$	1.05		(table A.8f)
$a_{WL.kWL=0.L=T}$	0.435		(table A.8a)
$a_{WL.L=T}$	1.043536		
$k_{WL.L=0}$	0.0072		(s_{WL}=0)
$a_{WL.L=0}$	0.4094		
L	0.175	m	(width of the heat conducting plate)
a_{WL}	1.044032		
a_K	0.92		(table A.9)
$R_{\lambda.B}$	0.15		(because layer 1 has been avoided)
f	1.196774		
a_B	0.559094		
$k^*_{H.floor}$	2.45608		
$\alpha_{cooling}$	6.5	W/m^2	
$\Delta R\alpha$	0.061254		
$R_{\lambda.B}$	**0**		**LAYER 1 IS AVOIDED**
$k_{H.cooling.floor}$	**3.322883**		
q	31.61856	W/m^2	
θ_i	25	°C	
σ	4	°C	
$\Delta\theta_H$	9.515399	°C	
$\theta_{V.des}$	13.34489	°C	
Q (W)	1354.125		
$\%_{floor}$	1		
$\%_{ceiling}$	0		
Q_{floor} (W)	1354.125		
A_{floor} (m^2)	42.8		
q_{floor} (W/m^2)	31.63843		
$\Delta\theta_{H.floor}$ (°C)	9.52138		
$\theta_{V.design.floor}$ (°C)	**13.34**		
cw	4190	J/kgK	
$\alpha_{cooling}$	6.5	W/m^2	
s_u	0.012	m	
λ_u	0.15	W/mK	

$R_{\lambda.B}$	0		
R_o	0.315846	m^2K/W	
$R_{u.ground\ floor}$	10.19292	m^2K/W	
$R_{u.first\ floor}$	1.595513	m^2K/W	
$A_{ground\ floor}$	35.25	m^2	
$A_{first\ floor}$	8.6	m^2	
$\theta_{u.ground\ floor}$	9	°C	
$\theta_{u.first\ floor}$	25	°C	
$m_{cooling.ground\ floor}$	**0.06526**	**239.94**	**kg/h**
$m_{cooling.first\ floor}$	**0.019436**	**74.97**	**kg/h**
		314.91	**kg/h**
		0.315	m3/h

14.2.5 Radiant system dimensioning – EN15377-1 (2008) – Cooling

θ_i	25.5	°C
R_i	0.322862	m^2K/W
q_i	31.61856	W/m^2
θ_m	15.29158	°C
σ	4	°C
θ_V	**13.2**	**°C**

14.2.6 $R_{con.i}$ and T_{supply} comparison

$R_{con.i}$ ($m^2 \cdot K/W$)	$\theta_{V.des}$ (°C)	R_i ($m^2 \cdot K/W$)	σ (°C)	θ_i (°C)	θ_R (°C)	θ_m (°C)	$q_{EN15377}$ (W/m^2)
0.01	29.89	0.18	5.00	21.00	24.89	27.05	33.14
0.02	30.21	0.19			25.21	27.38	33.14
0.03	30.52	0.20			25.52	27.71	33.14
0.04	30.84	0.21			25.84	28.04	33.14
0.05	31.16	0.22			26.16	28.38	33.14
0.06	31.48	0.23			26.48	28.71	33.14
0.07	31.80	0.24			26.80	29.04	33.14
0.08	32.12	0.25			27.12	29.37	33.14
0.09	32.44	0.26			27.44	29.70	33.14
0.10	32.76	0.27			27.76	30.03	33.14
0.11	33.09	0.28			28.09	30.36	33.14
0.12	33.41	0.29			28.41	30.70	33.14
0.13	33.73	0.30			28.73	31.03	33.14
0.14	34.06	0.31			29.06	31.36	33.14
0.15	34.38	0.32			29.38	31.69	33.14

14.3 Weather shield simulations

Copenhagen

Tmax_weathershield	30.51	°C
Tmax_ext	26.60	°C

Tavg_July_weathershield	16.74	°C
Tavg_July_ext	16.6	°C

Paris

Tmax_weather_shield	33.4	°C
Tmax_ext	30.00	°C

Tavg_July_weathershield	19.5	°C
Tavg_July_ext	19.4	°C

14.4 Heat capacity calculation – VDI 2078

14.4.1 Structure 0

SOUTH VERTICAL WALL

Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
Material				
Th. cond.	[W/mK]	0.11	0.032	0.11
Specific HC	[kJ/kgK]	1.6	0.9	1.6
Density	[kg/m3]	560	35	560
Thickness	[m]	0.03	0.3	0.03
Rough Area	[m2]	19.47	19.47	19.47
Window area	[m2]	2.52	2.52	2.52
Net wall area	[m2]	16.95	16.95	16.95
Volume	[m3]	0.5085	5.085	0.5085
Resistance	[m2-K/W]	0.2727273	9.375	0.272727273

$C_{eff.VDI\ 2078}$	455616	J/K
$m_{VDI\ 2078}$	284.76	kg/m^2

SOUTH 26° tilted WALL	Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
	Material				
	Th. cond.	[W/mK]	0.11	0.032	0.11
	Specific HC	[kJ/kgK]	1.6	0.9	1.6
	Density	[kg/m3]	560	35	560
	Thickness	[m]	0.03	0.3	0.03
	Rough Area	[m2]	16.52	16.52	16.52
	Window area	[m2]	1.4	1.4	1.4

$C_{eff.VDI\ 2078}$	**406425.6**	**J/K**
$m_{VDI\ 2078}$	254.016	kg/m^2

WEST VERTICAL WALL	Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
	Material				
	Th. cond.	[W/mK]	0.11	0.032	0.11
	Specific HC	[kJ/kgK]	1.6	0.9	1.6
	Density	[kg/m3]	560	35	560
	Thickness	[m]	0.03	0.3	0.03
	Rough Area	[m2]	45.52	45.52	45.52
	Window area	[m2]	0	0	0
	Net wall	[m2]	45.52	45.52	45.52

(sum of west facades of ground floor. first floor and tech room)

area				
Volume	[m3]	1.3656	13.656	1.3656
Resistance	[m2-K/W]	0.2727273	9.375	0.27272727 3

$C_{eff.VDI\ 2078}$		1223577.6	J/K
$m_{VDI\ 2078}$		764.736	kg/m^2

NORTH VERTICAL WALL (ext wall of ground floor zone - EXT WALL)

Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
Material				
Th. cond.	[W/mK]	0.11	0.032	0.11
Specific HC	[kJ/kgK]	1.6	0.9	1.6
Density	[kg/m3]	560	35	560
Thickness	[m]	0.03	0.3	0.03
Rough Area	[m2]	9.72	9.72	9.72
Window area	[m2]	1.82	1.82	1.82
Net wall area	[m2]	7.9	7.9	7.9
Volume	[m3]	0.237	2.37	0.237
Resistance	[m2-K/W]	0.2727273	9.375	0.272727273

$C_{eff.VDI\ 2078}$	212352	J/K
$m_{VDI\ 2078}$	132.72	kg/m^2

NORTH VERTICAL WALL (ext wall of tech room - INT WALL INS)

Layer #		PLYWOOD	OAK WOOD	GLASS FIBER	OAK WOOD	PLYWOOD
Material						
Th. cond.	[W/mK]	0.11	0.16	0.032	0	0.11
Specific HC	[kJ/kgK]	1.6	2.7	0.9	3	1.6
Density	[kg/m3]	560	700	35	##	560
Thickness	[m]	0.015	0.02	0.08	0	0.015
Rough Area	[m2]	9	9	9	9	9
Window area	[m2]	0	0	0	0	0
Net wall area	[m2]	9	9	9	9	9
Volume	[m3]	0.135	0.18	0.72	0	0.135
Resistance	[m2-K/W]	0.1363636	0.125	2.5	0	0.136363636

$C_{eff.VDI\ 2078}$	461160	J/K
$m_{VDI\ 2078}$	201.6	kg/m^2

NORTH 62° tilted WALL (first floor - EXT WALL)

Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
Material				
Th. cond.	[W/mK]	0.11	0.032	0.11
Specific HC	[kJ/kgK]	1.6	0.9	1.6
Density	[kg/m3]	560	35	560
Thickness	[m]	0.03	0.3	0.03
Rough Area	[m2]	13.68	13.68	13.68
Window area	[m2]	0.8	0.8	0.8

Net wall area	[m2]	12.88	12.88	12.88
Volume	[m3]	0.3864	3.864	0.3864
Resistance	[m2-K/W]	0.2727273	9.375	0.272727273

$C_{eff.VDI\ 2078}$	346214.4	J/K
$m_{VDI\ 2078}$	216.384	kg/m^2

EAST VERTICAL WALL (ground floor and first floor - EXT WALL)

Layer #		PLYWOOD	GLASS FIBER	PLYWOOD
Material				
Th. cond.	[W/mK]	0.11	0.032	0.11
Specific HC	[kJ/kgK]	1.6	0.9	1.6
Density	[kg/m3]	560	35	560
Thickness	[m]	0.03	0.3	0.03
Rough Area	[m2]	36.63	36.63	36.63
Window area	[m2]	0.9	0.9	0.9
Net wall area	[m2]	35.73	35.73	35.73
Volume	[m3]	1.0719	10.719	1.0719
Resistance	[m2-K/W]	0.2727273	9.375	0.272727273

(it's less of the western area because of the doors)

$C_{eff.VDI\ 2078}$	960422.4	J/K
$m_{VDI\ 2078}$	600.264	kg/m^2

INTERNAL WALLS (non insulated)	Layer #		PLYWOOD	OAK WOOD	PLYWOOD
	Material				
	Th. cond.	[W/mK]	0.11	0.16	0.11
	Specific HC	[kJ/kgK]	1.6	2.7	1.6
	Density	[kg/m3]	560	700	560
	Thickness	[m]	0.02	0.06	0.02
	Rough Area	[m2]	16.265	16.265	16.265
	Window area	[m2]	0	0	0
	Net wall area	[m2]	16.265	16.265	16.265
	Volume	[m3]	0.3253	0.9759	0.3253
	Resistance	[m2-K/W]	0.1818182	0.375	0.181818182
	C	[J/K]	291468.8	1844451	291468.8
	C_apparent	**[J/K]**	**2427388.6**	149240	
	U_transmittance	[W/m2-K]			
	UA	[W/K]			

$C_{eff.VDI\ 2078}$	**1213694.3**	**J/K**
$m_{VDI\ 2078}$	540.2705	kg/m^2

INTERNAL WALL (insulated)	Layer #		PLYWOOD	OAK WOOD	GLASS FIBER	OAK WOOD	PLYWOOD
	Material						
	Th. cond.	[W/mK]	0.11	0.16	0.032	0	0.11
	Specific HC	[kJ/kgK]	1.6	2.7	0.9	3	1.6
	Density	[kg/m3]	560	700	35	700	560
	Thickness	[m]	0.015	0.02	0.08	0	0.015
	Rough Area	[m2]	9.625	9.625	9.625	10	9.625

Window area	[m2]	0	0	0	0	0
Net wall area	[m2]	9.625	9.625	9.625	10	9.625
Volume	[m3]	0.144375	0.1925	0.77	0	0.144375
Resistance	[m2-K/W]	0.1363636	0.125	2.5	0	0.136363636
C	[J/K]	129360	363825	24255	363825	129360
C_apparent	**[J/K]**	**1010625**	105000			
U_transmittance	[W/m2-K]					
UA	[W/K]					

$C_{eff.VDI\ 2078}$	**246592.5**	**J/K**
$m_{VDI\ 2078}$	107.8	kg/m^2

14.4.2 Structure 1

SOUTH VERTICAL WALL

Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
Material					
Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
Density	[kg/m 3]	1500	1280	35	1500
Thickness	[m]	0.015	0.1	0.3	0.015
Rough Area	[m2]	19.47	19.47	19.47	19.47

Window area	[m2]	2.52	2.52	2.52	2.52
Net wall area	[m2]	16.95	16.95	16.95	16.95
Volume	[m3]	0.25425	1.695	5.085	0.25425
Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571

$C_{eff.VDI\ 2078}$	1950165.3	J/K
$m_{VDI\ 2078}$	2550.975	kg/m^2

SOUTH 26° tilted WALL

Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
Material					
Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
Density	[kg/m3]	1500	1280	35	1500
Thickness	[m]	0.015	0.1	0.3	0.015
Rough Area	[m2]	16.52	16.52	16.52	16.52
Window area	[m2]	1.4	1.4	1.4	1.4
Net wall	[m2]	15.12	15.12	15.12	15.12

	area					
	Volume	[m3]	0.2268	1.512	4.536	0.2268
	Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571
			$C_{eff.VDI\ 2078}$		1739616.48	J/K
			$m_{VDI\ 2078}$		2275.56	kg/m^2

	Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER	
WEST VERTICAL WALL	Material						
	Th. cond.	[W/m K]	0.7	0.25	0.032	0.7	
	Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1	
	Density	[kg/m 3]	1500	1280	35	1500	
	Thickness	[m]	0.015	0.1	0.3	0.015	
	Rough Area	[m2]	45.52	45.52	45.52	45.52	(sum of west facades of ground floor. first floor and tech room)
	Window area	[m2]	0	0	0	0	
	Net wall area	[m2]	45.52	45.52	45.52	45.52	
	Volume	[m3]	0.6828	4.552	13.656	0.6828	

	Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571
			$C_{eff.VDI\ 2078}$		5962664.8	J/K
			$m_{VDI\ 2078}$		6850.76	kg/m^2

NORTH VERTICAL WALL (ext wall of ground floor zone - EXT WALL)	Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
	Material					
	Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
	Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
	Density	[kg/m3]	1500	1280	35	1500
	Thickness	[m]	0.015	0.1	0.3	0.015
	Rough Area	[m2]	9.72	9.72	9.72	9.72
	Window area	[m2]	1.82	1.82	1.82	1.82
	Net wall area	[m2]	7.9	7.9	7.9	7.9
	Volume	[m3]	0.1185	0.79	2.37	0.1185
	Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571
			$C_{eff.VDI\ 2078}$		1034821	J/K
			$m_{VDI\ 2078}$		1188.95	kg/m^2

NORTH VERTICAL WALL (ext wall of tech room - INT WALL INS)	Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
	Material					
	Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
	Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
	Density	[kg/m 3]	1500	1280	35	1500
	Thickness	[m]	0.015	0.05	0.2	0.015
	Rough Area	[m2]	9	9	9	9
	Window area	[m2]	0	0	0	0
	Net wall area	[m2]	9	9	9	9
	Volume	[m3]	0.135	0.45	1.8	0.135
	Resistance	[m2-K/W]	0.021428571	0.2	6.25	0.021428571

$C_{eff.VDI\ 2078}$		700830	J/K
$m_{VDI\ 2078}$		778.5	kg/m^2

NORTH 62° tilted WALL (first floor - EXT WALL)

Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
Material					
Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
Density	[kg/m 3]	1500	1280	35	1500
Thickness	[m]	0.015	0.1	0.3	0.015
Rough Area	[m2]	13.68	13.68	13.68	13.68
Window area	[m2]	0.8	0.8	0.8	0.8
Net wall area	[m2]	12.88	12.88	12.88	12.88
Volume	[m3]	0.1932	1.288	3.864	0.1932
Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571

$C_{eff.VDI\ 2078}$	**1687151.2**	**J/K**
$m_{VDI\ 2078}$	1938.44	kg/m^2

EAST VERTICAL WALL (ground floor and first floor - EXT WALL)

Layer #		INT PLASTER	LIGHT CONCRETE	GLASS FIBER	EXT PLASTER
Material					
Th. cond.	[W/m K]	0.7	0.25	0.032	0.7
Specific HC	[kJ/kg K]	1.1	0.83	0.9	1.1
Density	[kg/m 3]	1500	1280	35	1500
Thickness	[m]	0.015	0.1	0.3	0.015
Rough Area	[m2]	36.63	36.63	36.63	36.63
Window area	[m2]	0.9	0.9	0.9	1.9
Net wall area	[m2]	35.73	35.73	35.73	34.73
Volume	[m3]	0.53595	3.573	10.719	0.52095
Resistance	[m2-K/W]	0.021428571	0.4	9.375	0.021428571

(it's less of the western area because of the doors)

$C_{eff.VDI\ 2078}$	4680272.7	J/K
$m_{VDI\ 2078}$	5768.0775	kg/m^2

14.4.3 Ground floor – active floor

Layer #		PLYWOOD	ALLUMINUM	OAK WOOD	GLASS FIBER	OAK WOOD	GLASS FIBER	OAK WOOD	GLASS FIBER	PLYWOOD FINISHING
Material										
Th. cond.	[W/mK]	0.11	200	0.16	0.032	0.16	0.032	0.16	0.032	0.11
Specific HC	[kJ/kgK]	1.6	0.86	2.7	0.9	2.7	0.9	2.7	0.9	1.6
Density	[kg/m3]	560	2700	700	35	700	35	700	35	560
Thickness	[m]	0.012	0.001	0.03	0.04	0.03	0.2	0.03	0.05	0.015
Rough Area	[m2]	35.4	35.4	35.4	35.4	35.4	35.4	35.4	35.4	35.4
Window area	[m2]	0	0	0	0	0	0	0	0	0
Net wall area	[m2]	35.4	35.4	35.4	35.4	35.4	35.4	35.4	35.4	35.4
Volume	[m3]	0.4248	0.0354	1.062	1.416	1.062	7.08	1.062	1.77	0.531
Embedded pipes HC	J/K	3588578.5								
Resistance	[m2-K/W]	0.1090909	0.000005	0.1875	1.25	0.1875	6.25	0.1875	1.5625	0.136363636

$C_{eff.VDI\ 2078}$	**6058578.056**	**J/K**
$m_{VDI\ 2078}$	1076.868	Kg/m^2

14.4.4 Ground floor – technical room

Layer #		PLYWOOD	OAK WOOD	GLASS FIBER	OAK WOOD	GLASS FIBER	OAK WOOD	GLASS FIBER	PLYWOOD FINISHING
Material									
Th. cond.	[W/mK]	0.11	0.16	0.032	0.16	0.032	0.16	0.032	0.11
Specific HC	[kJ/kgK]	1.6	2.7	0.9	2.7	0.9	2.7	0.9	1.6
Density	[kg/m3]	560	700	35	700	35	700	35	560
Thickness	[m]	0.012	0.05	0.04	0.03	0.2	0.03	0.05	0.015
Rough Area	[m2]	3	3	3	3	3	3	3	3
Window area	[m2]	0	0	0	0	0	0	0	0
Net wall area	[m2]	3	3	3	3	3	3	3	3
Volume	[m3]	0.036	0.15	0.12	0.09	0.6	0.09	0.15	0.045
Resistance	[m2-K/W]	0.1090909	0.3125	1.25	0.1875	6.25	0.1875	1.5625	0.136363636

$C_{eff.VDI\ 2078}$	315756	J/K
$m_{VDI\ 2078}$	125.16	Kg/m^2

14.4.5 First floor – active part

Layer #		PLYWOOD	ALLUMINUM	OAK WOOD (real)	OAK WOOD (fic)	OAK WOOD (fic)	GLASS FIBER	OAK WOOD	PLYWOOD
Material									
Th. cond.	[W/mK]	0.11	200	0.16	0.64	0.64	0.032	0.16	0.11
Specific HC	[kJ/kgK]	1.6	0.86	2.7	2.7	2.7	0.9	2.7	1.6
Density	[kg/m3]	560	2700	700	175	175	35	700	560
Thickness	[m]	0.01	0.001	0.03	0.06	0.06	0.05	0.06	0.01
Rough Area	[m2]	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
Window area	[m2]	0	0	0	0	0	0	0	0
Net wall area	[m2]	5.6	5.6	5.6	5.6	5.6	5.6	5.6	5.6
Volume	[m3]	0.056	0.0056	0.168	0.336	0.336	0.28	0.336	0.056
Embedded pipes HC	J/K	580189.3313							
Resistance	[m2-K/W]	0.090909091	0.000005	0.1875	0.09375	0.09375	1.5625	0.375	0.090909091
						$C_{eff.VDI\ 2078}$	960888.5313	J/K	
						$m_{VDI\ 2078}$	164.08	kg/m^2	

14.4.6 First floor – non-active part

Layer #		PLYWOOD	OAK WOOD	GLASS FIBER	PLYWOOD
Material					
Th. cond.	[W/mK]	0.11	0.16	0.032	0.11
Specific HC	[kJ/kgK]	1.6	2.7	0.9	1.6
Density	[kg/m3]	560	700	35	560
Thickness	[m]	0.01	0.06	0.09	0.01
Rough Area	[m2]	3	3	3	3
Window area	[m2]	0	0	0	0
Net wall area	[m2]	3	3	3	3
Volume	[m3]	0.03	0.18	0.27	0.03
Resistance	[m2-K/W]	0.090909091	0.375	2.8125	0.090909091

$C_{eff.VDI\ 2078}$	367080	J/K
$m_{VDI\ 2078}$	142.8	kg/m^2

14.4.7 Embedded system capacity

GROUND FLOOR (ground floor)

LAYER	L [m]	Density [kg/m3]	HC [J/kgK]	Capacity [J/kg]
PEX	167	950	2000	3588578.46

	Fold	Embrace	
Area	59	35.3	
Pipe lenght [m]	278	166.3288	167

FIRST FLOOR (first floor to ground floor)

LAYER	L [m]	Density [kg/m3]	HC [J/kgK]	Capacity [J/kg]
PEX	27	950	2000	580189.331

	Fold	Embrace	
Area	59	5.6	
Pipe lenght [m]	278	26.38644	27

The calculation of the additional thermal capacity due to the presence of the embedded system has been based on the SDE 12 house. the FOLD. using (Kazanci & Skrupskelis, 2012) as reference.

14.4.8 Structures 2 and 3

Calculations for structure 2 and 3 have been based on the following concrete data:

LIGHT DENSITY CONCRETE

Th. cond.	[W/mK]	0.25
Specific HC	[kJ/kgK]	0.83
Density	[kg/m3]	1280

MEDIUM DENSITY CONCRETE

Th. cond.	[W/mK]	0.7
Specific HC	[kJ/kgK]	0.88
Density	[kg/m3]	1760

HIGH DENSITY CONCRETE

Th. cond.	[W/mK]	1.6
Specific HC	[kJ/kgK]	0.92
Density	[kg/m3]	2082

The results are listed below.

14.4.9 Thermal capacity - Results

14.4.9.1 Structure 0

$C_{eff.VDI\ 2078}$	**13228357**	**J/K**
	307636.2	**J/K·m^2**
	85.45	**Wh/K·m^2**

$m_{VDI\ 2078}$	**4611.459**	**kg**
	107.24	**kg/m^2**

14.4.9.2 Structure 1

$C_{eff.VDI\ 2078}$	**26465851.22**	**J/K**
	615484.912	**J/K·m^2**
	170.97	**Wh/K·m^2**

$m_{VDI\ 2078}$	**24162.89425**	**kg**
	561.93	**kg/m2**

14.4.9.3 Structure 2

$C_{eff.VDI\ 2078}$	36788560.34	J/K
	855547.9148	$J/K\cdot m^2$
	237.65	$Wh/K\cdot m^2$

$m_{VDI\ 2078}$	36232.92625	kg
	842.63	kg/m2

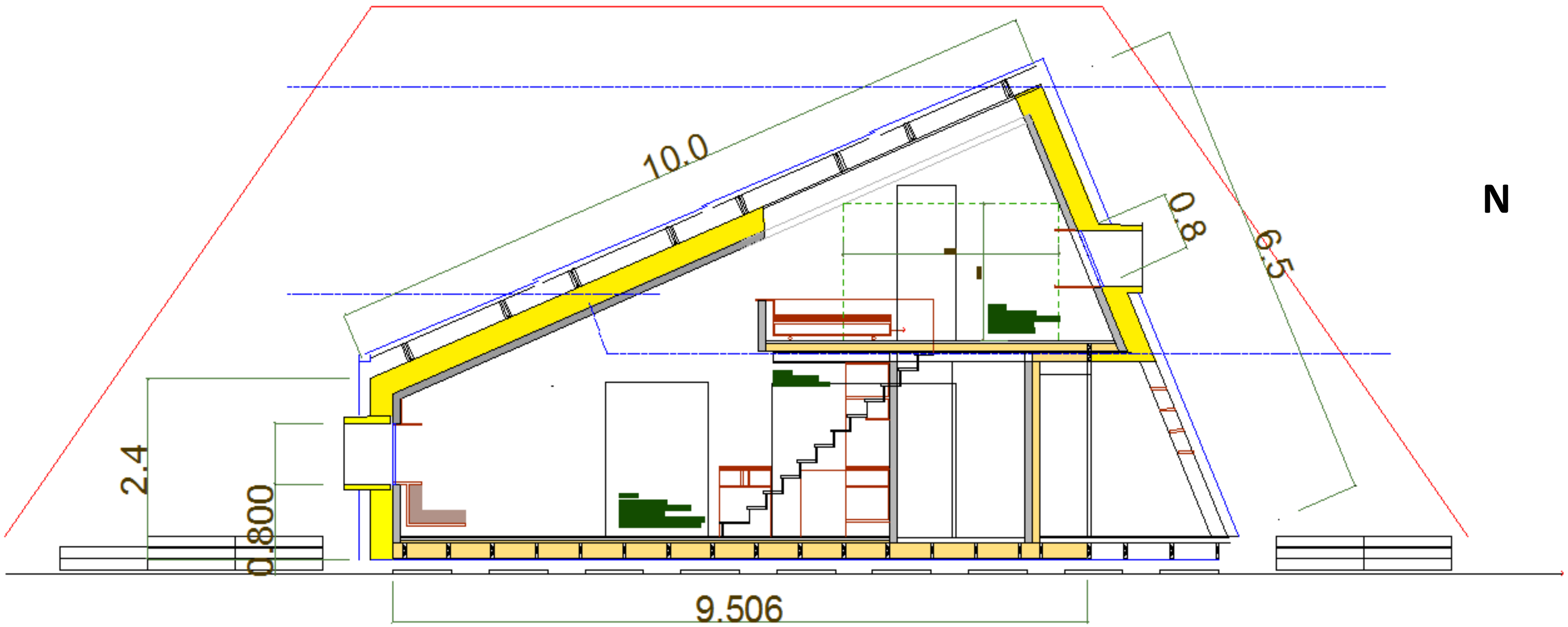
14.4.9.4 Structure 3

$C_{eff.VDI\ 2078}$	49253236.3	J/K
	1145424.1	$J/K\cdot m^2$
	318.17	$Wh/K\cdot m^2$

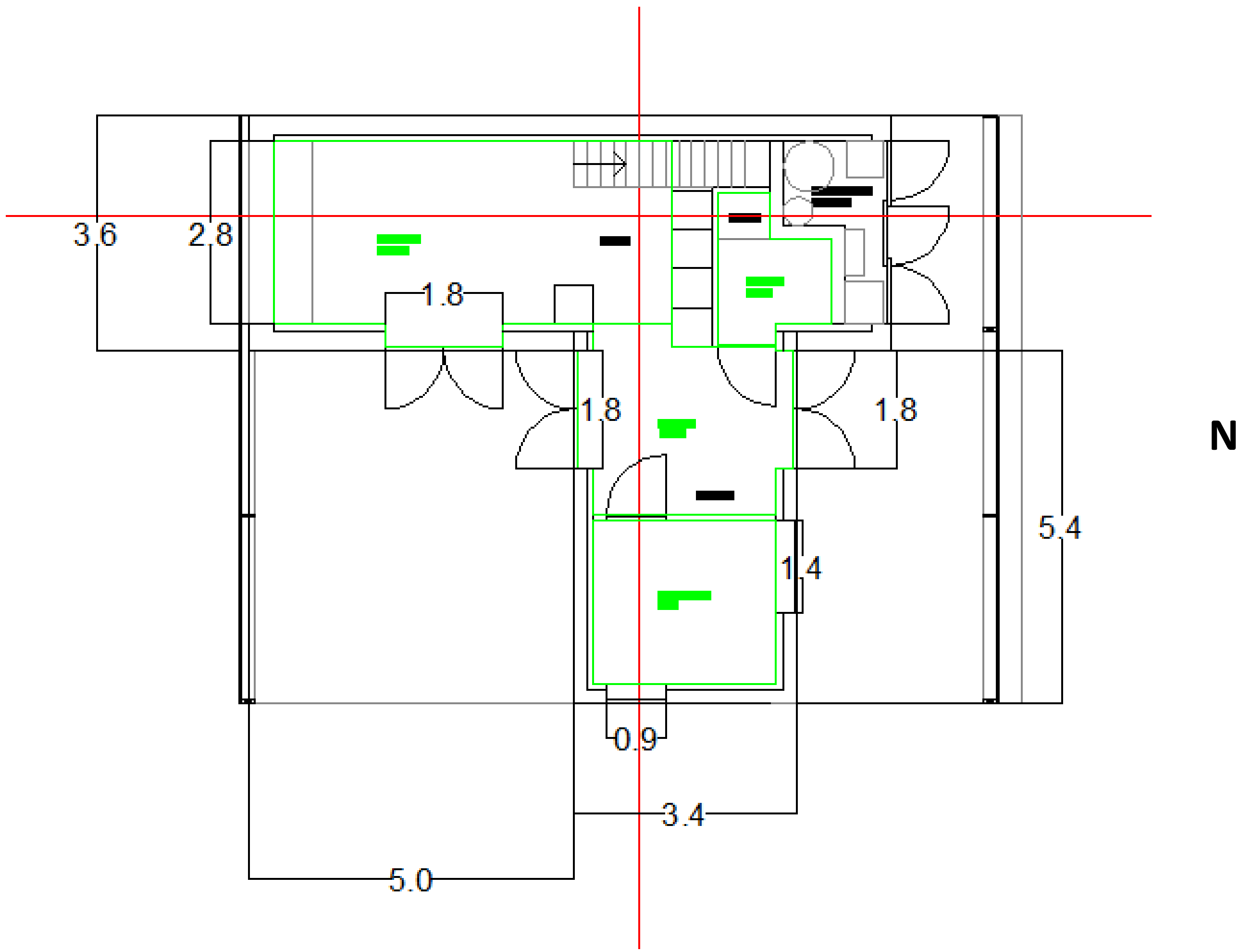
$m_{VDI\ 2078}$	50899.756	kg
	1183.72	kg/m2

14.5 Drawings

14.5.1 Overview



14.5.2 Ground floor



14.5.3 First floor