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Analysis of a District Energy system containing Thermal Energy Storage and Heat Pumps

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Abstract

In the context of meeting the targets set by 2050 for reducing greenhouse gas emissions, District Energy (DE) systems are considered to be a proven solution. This is essentially due to their ability to re-use energy that would otherwise be wasted, and its compatibility with a variety of other technologies, such as Thermal Energy Storage (TES) and renewable energy sources. When available, thermal energy storage provides greater flexibility, reliability, as well as energy security and it can be used to optimize equipment responsible for thermal energy production, as for instance, heat pumps.

The main objective of this project is to study the influence of the introduction of short-term thermal storage in a DE, where heat and cold requirements are supplied by a combination of seasonal TES and heat pumps. To be specific, the focus is to analyze to what extent can short-term TES be used to shift the heat pumps electrical heating loads, from peak to off-peak periods, and quantify the influence of this strategy on energy production and electricity consumption.

In order to do this, space heating and cooling demand data regarding a group of buildings is determined in Dymola/Modelica, and the energy systems performance is evaluated by using an analytical MATLAB model. The results obtained show that the introduction of short-term storage allowed to shift some of the thermal load from peak to off-peak periods. This operation led to a significant reduction in the individual electricity costs for the heat consumers (11.5% to 37.5%), which were determined based on electricity prices from the Dutch EPEX day-ahead spot market. Regarding electricity consumption and total heat production, it was noticed that the introduction of short-term storage led to an increase in the total heat output from heat pumps (~7%), mainly due to higher thermal losses. However, the global heat pumps coefficient of performance (COP) also increased (~14%), which resulted in less electricity consumption (~-13%), despite of the higher heat production.

Keywords: District Energy, Seasonal Thermal Energy Storage (STES), Short-term Thermal Energy Storage, Load Shifting, MATLAB, Dymola.

Resumo

No contexto de atingir as metas estabelecidas até 2050 para a redução da emissão de gases de efeito de estufa, as redes urbanas de energia são já consideradas uma solução provada. Isto deve-se essencialmente ao facto de estas possibilitarem recuperar energia que seria de outra forma desperdiçada. Para além disso, possibilitam a integração de diferentes fontes de energia renovável, ou outras tecnologias, como é o caso de armazenamento térmico de energia. Quando presente, armazenamento de energia térmica confere maior flexibilidade, bem como maior segurança energética e pode ser usado para otimizar o equipamento responsável pela produção de energia térmica, como por exemplo bombas de calor.

O principal objetivo deste projeto é estudar a influencia da introdução de armazenamento de energia térmica de curta duração numa rede urbana de energia, em que o abastecimento de calor é suprido por uma combinação de armazenamento de energia térmica sazonal e bombas de calor. Em particular, pretende-se analisar de que forma pode o armazenamento de curta duração, que consiste num tanque de água, ser usado para deslocar a produção de calor (carga térmica), de períodos de cheia ou ponta, para períodos de vazio, e quantificar as consequências desta estratégia (*load shifting*) na produção de energia térmica e consumo de eletricidade.

Para que isto seja possível, o consumo de energia relacionado com aquecimento e arrefecimento de um grupo de edifícios é determinado usando um modelo implementado em *Modelica*, e a análise dos sistemas de energia é feita através de um modelo analítico, implementado em *Matlab*. Os resultados obtidos mostram que a introdução de armazenamento térmico de curto prazo, permitiu deslocar parte da carga térmica de períodos de pico, para períodos de vazio. Isto levou a uma redução significativa (11.5% a 37.5%) nos custos individuais de eletricidade para os consumidores de calor, que foram determinados com base nos preços de mercado de eletricidade holandês (EPEX), e em tarifas de distribuição e transmissão. Paralelamente, verificou-se um aumento na produção total de calor (~ 7%), principalmente devido a maiores perdas térmicas face ao sistema sem armazenamento de curta duração. No entanto, a eficiência global das bombas de calor (COP) também aumentou (~ 14%), o que resultou ainda assim num menor consumo global de eletricidade (~ -13%), apesar da maior produção de energia.

Palavras-Chave: Redes Urbanas de Energia, Armazenamento térmico de curta duração, Armazenamento térmico sazonal, *Load Shifting*, *MATLAB*, *Dymola*.

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Chapter 1 - Introduction

Energy is closely linked to every aspect of modern human life. It is a key element for the economic growth by contributing to the development of different sectors, within a nation [1]. During last decades, there has been a nonstop increase in the global energy demand, particularly due to rapid development of nations located in Asia and in the Middle East [2]. In addition, the global share of oil, natural gas and coal, used as primary energy, remains dominant (figure 1.1).

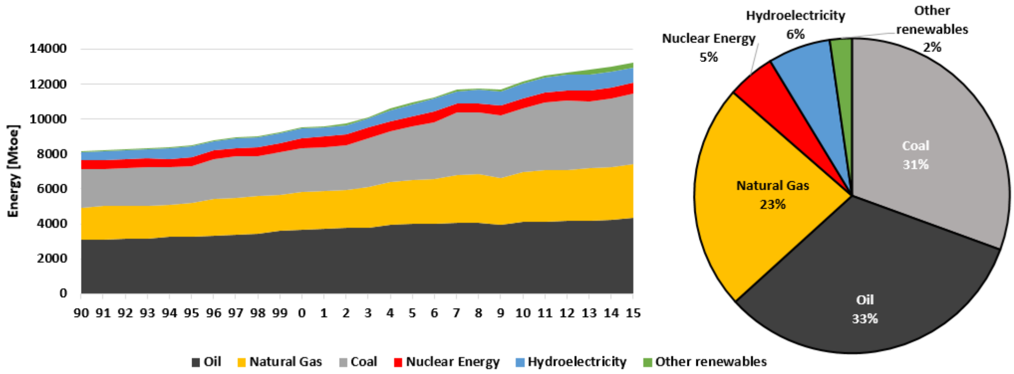

Figure 1.1 - World primary energy consumption: (left) per source in Mtoe, 1990 – 2015; (right) Share per source in 2015. [3]

There is today the awareness that in order to sustain the living standards in developed nations, as well as, to improve the social and economic status in the developing countries, future energy systems must become more sustainable from the economic and environmental points of view.

International Energy Agency (IEA) recommends a reduction in greenhouse gas emissions by 80% below 1990s levels in 2050 [4]. An important step to achieve this, is to reduce buildings related emissions. The building sector alone, including households and services, is responsible for more than one-third of the global final energy consumption in Europe [5].

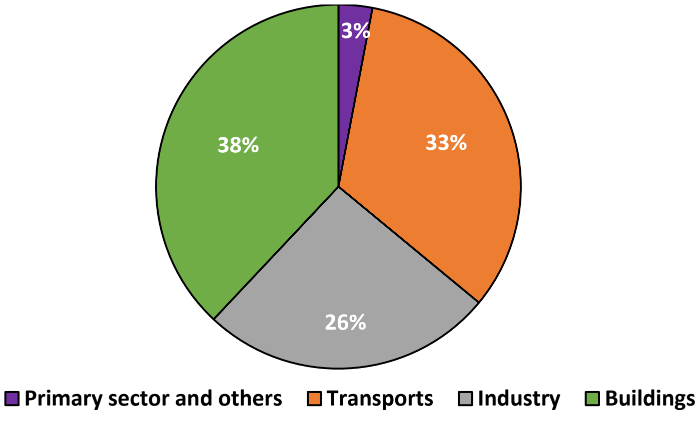

Figure 1.2 - Share on Final Energy consumption per sector in Europe, 2015.[6]

In this context, IEA refers that stricter building codes and efficiency standards, in combination with the development of a set of low-carbon technologies, are key actions [6], [7]. Among others, energy efficient equipment/envelope and a greater use of renewables, heat pumps, energy storage, and most importantly, a decarbonized power sector are pointed as the main drivers of decarbonization in buildings.

Furthermore, one of the topics currently discussed is the way to ensure space heating and cooling in the future. Thermal needs account for nearly 60% of the global energy consumption in buildings, therefore representing the largest opportunity to reduce energy demand and CO_2 emissions, within the sector [5].

District Energy (DE) is a proven energy solution to this issue [8]. It basically consists on using an underground piping network, to distribute thermal energy from a central source, to a group of consumers [9]. By doing this, the energy supply can be considered on a district scale, which allows to identify and use energy that is already available, and that would otherwise be wasted. For example, the water of an aquifer, or the internal excess heat of a shopping center [10].

In addition, DE systems can integrate various technologies. Energy Storage (ES) is one technology often present in these systems. It can be utilized to temporarily retain energy for later use, and it can be applied in a number of different applications. In general, these systems can either be electric or thermal. Additionally, one can categorize them regarding its location and storage duration, as described in the following table.

Table 1.1 - Energy storage classification regarding storage location and duration. [12]

Location	**Centralized**, if the storage unit is connected to all or the majority of the components of the system. **Distributed**, if storage is applied in smaller scale and associated to particular production plant, building or other specific part of the system.
Duration	**Short-term**, if energy is retrieved on an hourly, daily or weekly basis. **Long-term,** if energy is recovered from season to season (Seasonal storage) or even larger periods.

As it is showed in table 1.1, the existing storage duration ranges from hours to months. Therefore, energy storage can help to address the typical mismatches between times with high energy production and times with high energy requirements. For instance, considering solar thermal energy, the highest production occurs during summer, which is not parallel with the highest demand, that occurs during winter.

Moreover, combining electrical heating and cooling equipment with Thermal Energy Storage (TES) has advantages. To be specific, TES can be used for electric load management, by shifting electrical heating and cooling demands from peak to off-peak periods. In other words, during off-peak times, heating or cooling can be generated using electric equipment (e.g. heat pump), stored in a short-term thermal energy storage unit, and then used during peak-hours. This process brings benefits both for customers and utilities. Customers can have more efficient systems and save money if they take advantage of different electricity prices, and the same applies to utilities, that can spread the demand over more time.

1.1. Scope and Method

The main objective of this project is to investigate the influence of introducing short-term thermal energy storage into a district energy system. In particular, the focus is to analyze to what extent can short-term TES be used to manage the electrical load of water-to-water heat pumps, that are used to supply space heating requirements, by using as sink and source the water of a centralized aquifer system (seasonal thermal storage).

It is expected that the addition of short-term storage will reduce the individual electricity costs for the energy consumers, that can take advantage of low-price electricity to produce and store thermal energy during off-peak hours, for later use when electricity prices and global demand are higher. Furthermore, this strategy should allow to decrease the capacity of the installed energy producing equipment, leading to less investment costs and better global efficiency. Therefore, the thermal energy production, and respective electricity consumption are two important performance indicators, that will be used to support analysis. A secondary objective is also defined. It consists on comparing two district system configurations, both containing short-term storage; One, in which short-term storage is present at each building in a distributed manner, and another considering that storage and energy production equipment are centralized, and connected to the entire district.

The first step towards the objectives defined is to do a bibliographic review, focusing on thermal energy storage and all the three main components of District Energy, which are the thermal energy production units, the distribution system and finally, the energy consumers. The aim is also to provide some insight on current and future development in DE and TES applications and technologies. The most relevant information is presented in the first part of this report.

The second part of this project consists on defining and implementing a methodology in order to evaluate the above referred topics. First, to generate heating and cooling demand data, a neighborhood case study was defined. It is based on a small set of buildings which are combined to form a street section. This representative district typology was taken from IEA-EBC Annex 60 (International Energy Agency – Energy in Buildings and Communities) Programme [11]. The demand data is determined using Modelica simulation language on Dymola software. After this, the calculated demand values are inputted into a numerical MATLAB model, in which different system configurations are implemented. This way, sets of performance indicator values resulting from having or not short-term storage, in a centralized or distributed manner, can be obtained, compared and used to support conclusions.

1.2. Thesis outline

This master thesis report is structured as follows:

- **Chapter 2** gives an overview on District Energy systems, embracing and describing its three essential aspects, the energy generation, energy distribution and energy consumers.
- **Chapter 3** addresses energy storage, focusing on short-term and seasonal thermal energy storage. It also provides particular information on the heat pump – seasonal TES combination.
- **Chapter 4** describes the case study buildings and energy system configurations implemented in this study, as well as the software tools utilized;
- **Chapter 5** focus on describing the methodology followed to determine heating and cooling demands in Dymola/Modelica, as well as, energy systems implementation in Matlab.
- **Chapter 6** compares the energy performance of the three system configurations implemented in Matlab. These are, the reference case without short-term storage; system 1 with distributed short-term TES and heat pumps; and finally, system 2 consisting on centralized short-term TES and energy production.
- **Chapter 7** summarizes and conclude this report and identifies directions for future work.

Chapter 2 - District Energy Systems

Cities must play a central role in the transition to sustainable energy. In fact, they represent more than 70% of the global energy demand, of which approximately half is for heating and cooling purposes [8]. In this context, district energy (DE) systems are a proven energy solution, referred to as a cost-effective technology, capable of reducing the use of primary energy and greenhouse gas emissions [4]. It provides already approximately 13% of the building thermal demand in the European Union [12].

A district energy network or district heating and cooling is a system where thermal energy, centrally produced and/or stored, is delivered to different consumers by using steam, chilled or hot water as energy carrier [9]. The water distributed is generally at a temperature level adequate to its final purpose (rather than being generated on site at each facility); Nonetheless, it can be a low temperature thermal source ready to be used by another production equipment (e.g. Heat Pump) at the consumer site.

District systems may be beneficial both from economic and environmental points of view. In the first place, centrally produced energy assures higher efficiency by replacing less efficient equipment in individual buildings, which consequently leads to lower primary energy consumption. In addition, emissions are easier to control and, in aggregate, are lower because of the higher quality of equipment and improved level of maintenance/operation [9], [13]. Moreover, by producing energy away from the consumer, harmful noises or odors produced by the equipment do not affect the final user, not to mention that the usable space in building increases, since it is no longer necessary to have energy producing related apparatus [9].

This technology is planned considering local fuels, resources or opportunities, and the associated costs are distributed among all customers. Furthermore, centralizing the operation, allows to create synergies between the production and supply of thermal energy and electricity, and other municipal systems such as sanitation or sewage treatment [8]. On the other hand, a large initial capital investment is often required not only for planning and designing, but also in both infrastructure and production utilities. District energy project costs vary greatly and depend on local construction environment and site conditions, such as soil type, availability of materials, labor rates and others [9].

In what concerns the maturity of these systems, one can identify four generations. Through the first three, the techniques evolved towards lower distribution temperatures, new materials and prefabricated equipment, including metering and monitoring tools [8]. The latest fourth-generation, which is currently under research, is first of all, capable of further decrease grid loss, making it feasible to connect to areas with low energy density (e.g. areas with many low-energy buildings). In addition, it uses even more diverse sources of heat, including low-grade waste heat, and can allow consumers to supply energy as well (figure 2.1).

For these systems, energy storage and information technology equipment, capable of communicating real data between supply and demand, are crucial components. When combined, they create the flexibility required to integrate high levels of variable renewable energy sources into the grid, and, really define fourth-generation district energy as an integrated part of smart grid systems [14].

In this sense, different authors [8], [15] discuss the role of 4th generation districts in future energy systems. In general, they concluded that DE is the most relevant solution to reach the existing environmental and energy goals. For this reason, the continuous expansion of existing networks, as well as, using individual reversible heat pumps on the remaining buildings, is advised.

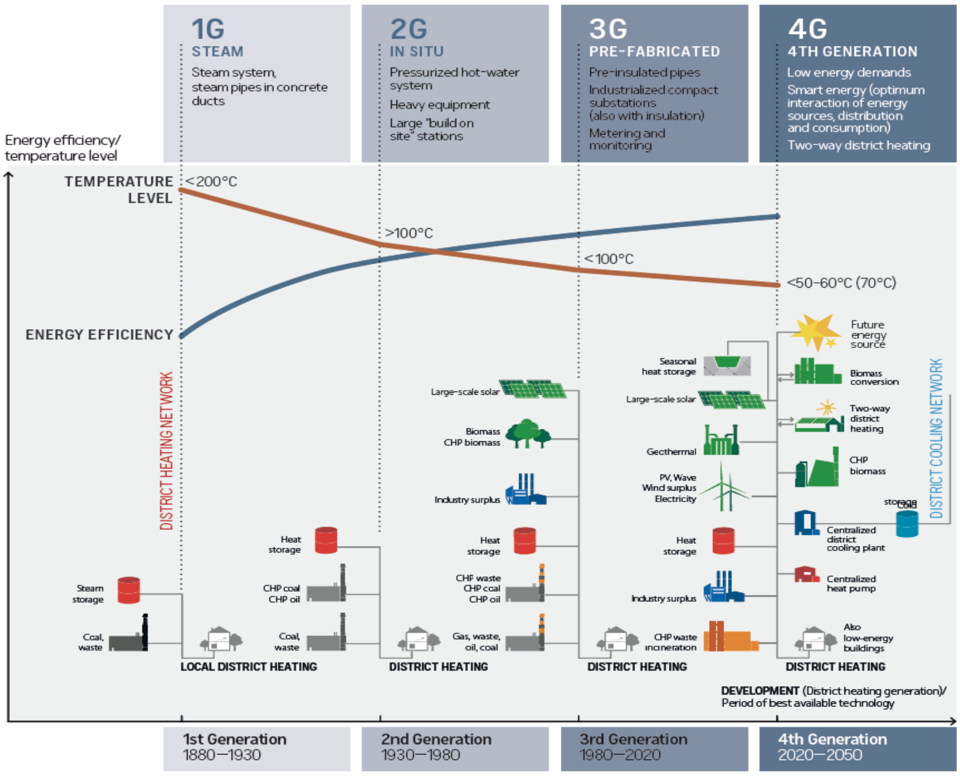

Figure 2.1 - Historical development of district energy networks, to the modern day and into the future. [9]

Finally, in what regards the configuration of these systems, a wide range of different approaches are possible. Even so, according to the ASHRAE handbook, [9] a district energy basically consists of three primary modules: the central source or production plant, the distribution network and the consumer interface units. These components are described in more detail in the sub-sections below.

2.1. Heat and cold Generation

District energy systems are adaptable to a large variety of energy sources and are designed taking into consideration the local resources. This way, it is possible to decrease the dependency on scarce or imported fuels. Moreover, the fundamental idea is to use energy that is already available and that would otherwise be wasted, through recovery and recycling [12].

2.1.1. Heating systems

Figure 2.2 illustrates the energy supply composition for generated district heating in Europe [12]. The results differ from country to country, but in the majority of them the heat generation is essentially based on recycled heat. This parcel includes the surplus heat from electricity production (CHP), waste-to-energy plants and industrial processes. Additionally, approximately two-thirds of the energy delivered by heat pumps, is also considered as recycled heat. The remaining part, is either based on the direct use of renewable energy sources and installations others than CHP, or in fossil fuel boilers and the other one-third of the heat originating from heat pumps.

In the larger systems, more than one source or technology can be used. In this case, a priority regime has to be established in order to guarantee that the system operation is as optimized as possible [16]. As such, the cheaper technologies are generally used during base load periods, whereas other more expensive are most commonly available as stand-by and peak load providers only.

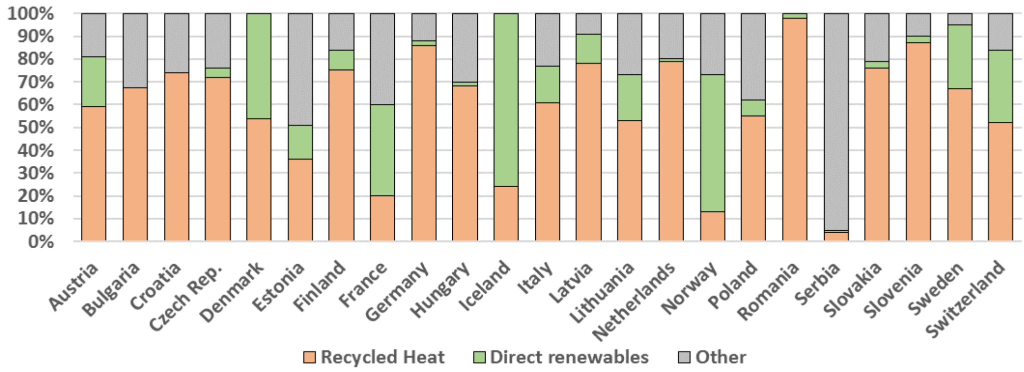

Figure 2.2 – Energy supply composition for district heat generated among European countries in 2015. [12]

In light of its flexibility, one can find a range of heat generation technologies associated to district energy systems. This topic is relatively well-documented, and various up-to-date public reports are available [17], providing extensive information in what regards technical and economic aspects. For this reason, this chapter focuses mainly on features within the scope of the project.

The table below list the main district heating technology options, along with its fuel source.

Table 2.1 - Principal district heating technologies and its respective fuel source. [8]

Technical solution	Fuel source
Combined heat and power (CHP)	Gas, biomass, coal, biogas, waste
Heat Pump	A heat source (e.g. ambient air, water or waste heat) and electricity
Solar thermal	Sun
Geothermal	Heat from water underground
Boiler	Gas, biomass, oil products, electricity
Waste-to-energy plant	Municipal solid waste and other combustible wastes
Waste heat recovery	Waste heat from an industrial process or low-grade heat from sewage

Combined heat and power plants (CHP) are specific combustion units, capable of supplying both heat and electricity simultaneously. The heat is generated as a by-product of electricity production, which would otherwise be rejected or wasted. By recovering it, this technology is able not only to operate with higher efficiencies than conventional power plants, [13] but also to supply heat directly to consumers. In what concerns its priority regime, it is used to provide the baseload generation and, in order to achieve improved results [18], it is typically combined with thermal energy storage technologies. CHP is the most common solution implemented to supply areas with concentrated heat demand, and in fact, is responsible for almost half of the district heating production in Europe [12].

It is possible to operate a CHP plant with different levels of electricity and heat outputs by varying the ratio of produced power to heat, within a technical limited feasibility range [13]. Therefore, it can be operated to meet a specific heat demand or to follow electricity prices. In other words, it is possible to maximize the production of either heat or electricity, according to what is most advantageous at any time. For instance, coupled with storage, the operator can take advantage of low electricity demand during night periods, to maximize heat production and store the surplus for later use.

Heat pumps (HP) are electrical units that, if coupled with thermal energy storage, are capable of performing similar operations. In fact, the retrofit of heat pumps with water storage to allow for their effective use in demand management services, is one key recommendation from IEA [6].

As expressed in table 2.1, heat pumps require a heat source in order to produce heat or cold, at adequate temperature levels. The thermal source is generally low temperature heat from the ambient air, water, ground or waste heat. This low-grade energy is converted to high-grade by putting in a required amount of compressor work, through electrical energy. The operating principle of this equipment will be later discussed in more detail, nonetheless, its main components are the electrically driven compressor, the condenser, the expansion valve and the evaporator.

The efficiency of a heat pump is determined by the Coefficient of Performance (COP). This parameter indicates the ratio of produced to used energy, and it mainly depends on factors such as, the temperatures of heat sink and heat source, the efficiency of its compressor, and the type of its working medium [19].

Above all, the temperature of the heat source and heat sink are very important factors for the COP value. To be specific, lowering the temperature difference between the heat source and heat sink, results in higher COPs, as indicated by equation 2.1.

$$COP = \eta_C \left(\frac{T_{sin}}{T_{sin}-T_{sor}}\right) \tag{2.1}$$

Where, T_{sin} and T_{sor} are the heat sink and heat source temperatures respectively, and η_C is the relation between the HP's efficiency under real conditions and the theoretical maximum reachable (Carnot Cycle) [19].

A low temperature heating system and high temperature heat source is then beneficial. In fact, for every degree reduction in the heat sink temperature (T_{sin}), the COP improves 1-2 % [19]. And in turn, for every degree enhancement in the heat source temperature (T_{sor}), the COP improves by 2-4% [19]. This makes heat pumps attractive to be combined with technologies, such as waste heat recovery, geothermal, solar thermal and thermal energy storage in low to medium temperature applications [20].

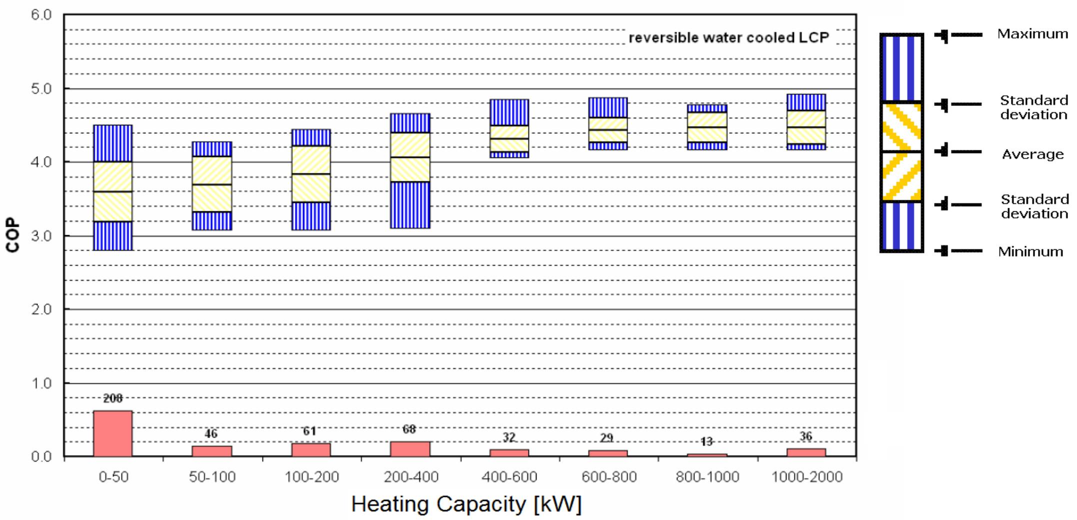

Figure 2.2 - COP of water-to-water heat pumps according to the heating capacity range. The numbers above red bars represent number of units reported. [21]

This technology can deliver more useful energy than the required energy to operate it, which means that its efficiency can be higher than 1. IEA Annex 48 – subtask 1 [21], provides statistical performance data regarding an universe of heat pumps. It refers that, the average COP of an air-to-air heat pump can be up to 4, however, water-to-water heat pumps can reach even higher efficiencies. In addition, as it can be seen in figure 2.2, large capacity heat pumps typically present higher COP values.

Another important factor when characterizing the performance of a heat pump is how it behaves in partial load conditions. This aspect has to be considered, otherwise, the overall efficiency of the system may be affected. The available data on this topic proves that a proper sizing of heat pumps has great importance, in particular for water-to-water systems [21]. The next figure demonstrates the relation between the partial load factor (PLF) and the partial load ratio (PLR), for a water-to-water heat pump. It shows that if the HP is not sized properly, its efficiency can drop substantially in the milder seasons. However, the loss of efficiency associated to partial load running can be avoided by having thermal energy storage, which acts like a thermal buffer.

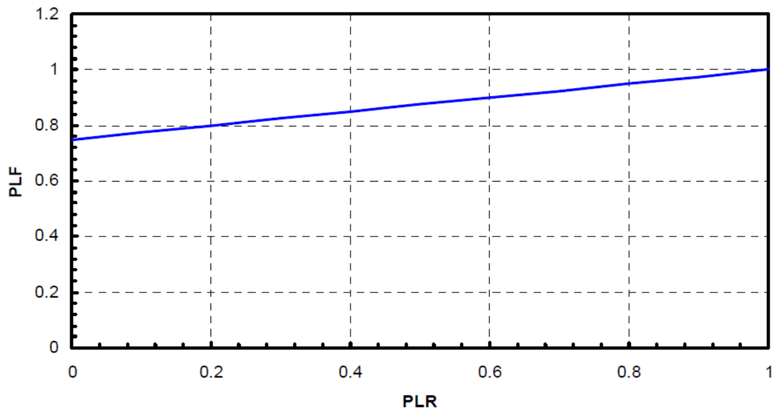

Figure 2.3 - Partial load factor vs. partial load ratio for water-to-water HP. [21]

To sum up, heat pumps are efficient technology for supplying both heating and cooling demands, particularly if coupled with thermal storage. Nonetheless, any technical solution mentioned on table 2, can be applied, according to which one is more adequate to each location.

As told already, solar thermal energy is commonly integrated with DH. Different solar thermal technologies exist [22]; however, it basically consists on using solar collectors to produce steam or hot water. In what regards its deployment, it can be implemented as a large centralized solution or as a distributed technology, providing only a minor contribution to the total heating load in a particular location of the grid. In addition, this system may also produce heat source for heat pumps, as described by these authors [20] [23].

Large-scale solar heating has a significant number of projects implemented [24]–[26]. The most relevant is the *Solarthermie-2000* [27], and it consists on a German program, in which pilot and demonstration projects have been realized. The project was developed in order to support implementation of solar and seasonal thermal storage concepts.

Another renewable source of thermal energy is the ground itself. Geothermal technologies essentially make use of the more constant ground temperatures, at higher depths. This immunity to seasonal variations, make geothermal also optimal to be combined with reversible heat pumps [28]. During the heating season, the deep soil is warmer than outside air, and is used as heat source. Then again, in Summer is colder, and it serves as low grade cooling energy.

Due to its characteristics, geothermal technologies are cheap, operational stable and typically have a long lifetime. For these reasons, and given its immunity to seasonal variations, geothermal is typically used as base load provider [8]. This technology is considered to be underdeveloped, and an ongoing project is being carried out by the European Commission, in order to support further expansion [29].

Finally, to illustrate the potential of residual and waste energy, the next figure shows the existing district heat network connected in Amsterdam, and sources of unused heat [8]. The red lines represent the distribution grid. The yellow squares denote the consumer sites, and the orange circles are suppliers of heat. The map also shows possible future residual or waste heat sources from: hospitals (green circles), data centers (blues circles) and offices (purple circles).

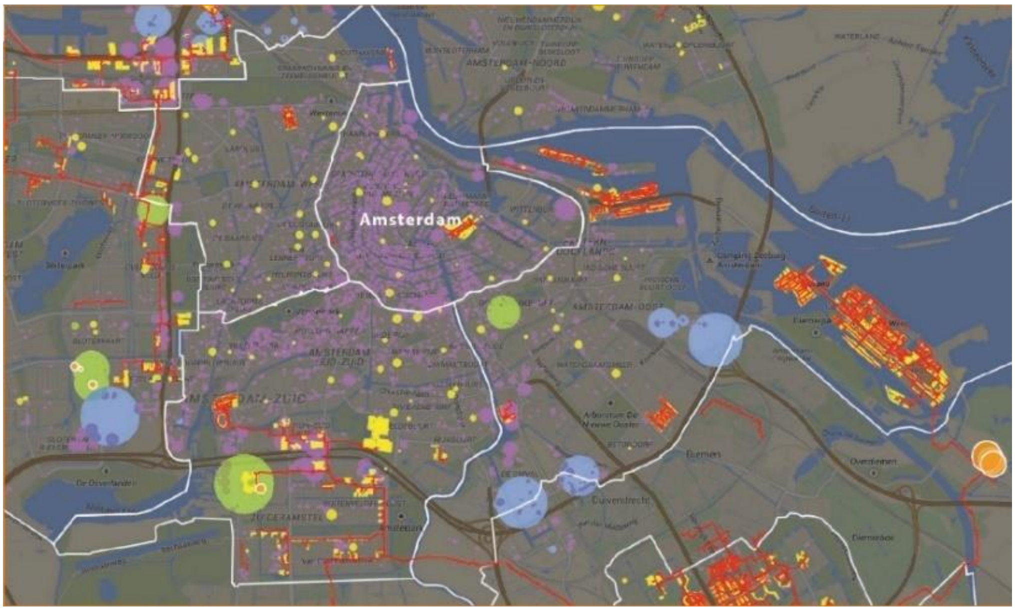

Figure 2.4 – Existing district network in Amsterdam, and sources of unused heat. [8]

On balance, the already existing sources of unused energy are many, and as a result, one can only expect the number of future district energy connections to increase.

2.1.2. Cooling systems

Despite the existence of successfully implemented projects [8], [30], district cooling (DC) is not as commonly applied as district heating (DH). In fact, during last years, the expansion of DC in Europe is barely insignificant [30], keeping a constant supply of around 1% of the total cooling market.

On the contrary, global cooling demand is progressively becoming more relevant, as for example, the energy consumption related to space cooling increased [8] more than 60% during the past two decades. In this context, one can expect the number of operating DC systems to increase, as they provide an alternative way to improve cooling supply in terms of environmental impact and efficiency.

Similar to DH, district cooling systems produce and distribute chilled water, or brine, to a group of buildings. The water is carried by a piping network to the different locations, and it provides cooling by absorbing excess heat. The respective heated medium usually returns to the cooling plant, where is cooled again. Nonetheless, depending on the return temperature, this low-grade energy may also be stored on a long-term thermal storage unit, in order to be used on the next season as low-grade heat.

Each district system is unique and based on local conditions. For this reason, depending on local resource availability and energy prices, district cooling can use a number of technologies (table 2.2).

Table 2.2 - Main district cooling technologies and its respective fuel source [8].

Technical solution	Fuel source
Electrical chillers	Electricity, cold air or water.
Absorption chiller	Surplus heat from waste incineration, industrial processes or renewable source, and electricity for pumping.
Heat Pump	Electricity, cold air or water.
Free Cooling	Cold water from lakes, rivers or aquifers, and electricity for pumping.

An electrical chiller, or compression chiller is a cooling solution, normally air or water-cooled from natural sources. The main components of this chiller are the evaporator, the electrically driven compressor and the condenser. The excess heat is absorbed by the refrigerant, that evaporates and is then compressed by an electrical compressor to pass through the condenser, in which the refrigerant releases the heat. This process is also used by heat pumps to produce cold.

The electricity demand increase in many European nations over the last decades is, among other reasons related to the operation of electrical refrigeration equipment, particularly in the Summer peaks [31]. In addition, cooling is becoming more relevant as a factor of thermal comfort in the residential sector and is already essential to the services, which constantly require cooling to keep indoor temperature levels, or to other refrigeration processes [32].

Instead of electricity, absorption chillers use heat as primary energy. The cooling process is quite complicated; however, cooling is essentially generated by water being vaporized at low pressure [30]. This technical solution can be found in cities having access to waste energy from incineration or other industrial processes. Moreover, if coupled with a CHP unit this technology allows to centrally produce electricity, heat and cold (trigeneration plant). The absorption cycle is particularly interesting during Summer. During this period, the demand for heat in the district is low, and these facilities allow to increase the production of cold water, with higher efficiencies, by using the available excess heat in the system [31].

In the context of addressing the electricity demand increase related to cooling, free/natural cooling systems may be important technical solutions. It essentially consists on utilizing locally available natural cold-water reservoirs, such as lakes, aquifers, rivers or the sea, as long as their temperatures are low enough. Therefore, using air as natural cooling medium, limits the usability to northern latitudes, where it is cold outdoors for the main part of the year.

The amount of natural cooling that can be produced mostly depends on the water temperature variation during the year, and the district cooling supply and return temperatures. Furthermore, even if the main idea of DE is to utilize unused energy, sometimes in larger districts this is not enough, and some kind of peak production unit is also needed. In this case, depending on what is most advantageous, a compression chiller, heat pump or absorption chiller is often installed, in order to produce a sufficient DC supply temperature [30].

In short, when available, free cooling consists of a very simple, cost efficient and environmental friendly energy solution. Moreover, although other district cooling technologies run on electricity, their power consumption is in most cases lower than in distributed solutions. Therefore, DC is definitely capable of reducing the cooling related electricity demand, particularly in Summer peaks.

2.2. Heat and Cold Distribution

The second major component of district energy is the insulated piping network that carries the water or steam. This infrastructure consists of a two-circuit system – flow and return pipes. The flow, transports the water from the source to the consumers, and the return is responsible for taking the water back to the source. In addition, the supply of heat is independent from the supply of cold, which means that, there is a piping network for heating and another for cooling. Moreover, the piping system is often the most expensive part of the district energy [9].

The type of distribution grid is mainly determined by site conditions such as, the layout of roads or the arrangement of the houses to be connected. However, the network size and the number of energy sources, are also decisive factors [33]. Essentially, there are three grid configurations, as showed in the next figure.

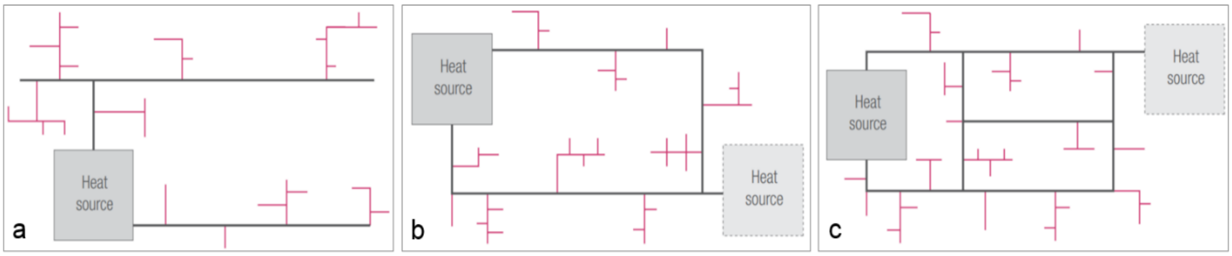

Figure 2.5 – Types of district energy networks: (a) Radial systems; (b) Ring networks; (c) Meshed networks. [33]

Due to its simplicity, radial systems are mostly used for smaller districts. The short piping paths and smaller diameter, result in lower construction costs and heat losses. The disadvantage of this configuration however, is that future extensions are only possible to a small extent.

On the other hand, ring networks are often used to supply larger areas with one or more thermal sources. With this configuration, not only different energy sources can be integrated, but it also results in higher supply security, as most consumers can be reached via two piping paths. This makes future extension or adding new equipment easier.

The last arrangement consists on meshed networks, which are ring grids nested inside each other. These offer optimum supply security and better expansion opportunities. Because of the related high investment costs, this configuration is mostly implemented in large distribution networks.

A number of aspects have to be taken into consideration when designing a distribution network. According to the district heating and cooling connection handbook [16] the dimensioning of the distribution grid is mainly ruled by three factors:

- The temperature difference between supply and return;
- The maximum mass flow rate (velocity and pipe diameter);
- The required differential pressure to overcome the flow resistance, and reach the most remote consumers.

The thermal capacity of the district is determined by both rate of water flow and temperature difference between supply and return. A large temperature differential means that less water has to be dispatched, in order to carry the same energy; in other words, the mass flow rate can be lower. This allows to use smaller pipes, leading to a reduction in the heat losses/gains, pumping power and initial capital investments.

The system temperature difference (supply/return) is typically limited to 8-11ºC for cooling, and 15-30ºC for heating. The maximum allowable flow velocities are governed by pressure drop constrains and other hydraulic disturbances, but in general, velocities higher than 2.5 – 3 m/s should be avoided [16].

In what concerns the water flow, it can be either constant or variable [9]. Constant flow is typically applied only on smaller systems, where simplicity of design and operation are important, or where distribution pumping costs are low. In addition, constant flow distribution is also applied to in-buildings circuits, with separated pumps. Variable flow is a more efficient solution. By using meters to track the load and variable speed water pumps, a high temperature difference (supply/return) can be maintained, since the pump can reduce flow and pressure, during partial load time.

When establishing design flows for thermal distribution systems, the diversity of the consumer demands should be considered. In particular, when in time do the maximum demands of the various consumers occur, and how the consumption is geographically distributed in the district, are two questions to address. By doing this, the energy supply and main distribution piping may be sized for a maximum thermal load.

Regarding thermal losses, it mainly depends on three parameters [16]:

- Amount of pipe insulation;
- Pipe dimensions (diameter and length);
- Supply and return temperatures;

The amount of pipe insulation generally results from a balance between capital investment and operation performance, as better insulation means more expenses. In short, reduced diameter and length, low temperatures and high insulation values are beneficial.

Many reports address thermal energy distribution in district systems. Since the decreased thermal demand from low-energy buildings affects the cost-effectiveness of traditional districts, authors have been focusing on optimized design concepts, mainly based on lower distribution temperatures and new equipment. Dalla Rosa *et al.* [34] showed the advantage of low supply and return temperatures and their effect on energy efficiency. The total primary energy use was 14.3% lower when compared to a standard network. Moreover, the thermal distribution losses were halved. This study also concludes that a district heating design optimized for low-temperature operation, is superior to a design optimized for low-flow operation.

Another relevant report addressing low temperature distribution was developed in the context of a Danish Energy Agency R&D Programme [35]. It defines low temperature distribution as a system in which, all components can operate in the range between 40-65ºC for supply, and 25-45 ºC for return flow, while meeting consumer demands for indoor thermal comfort. Moreover, this report states the benefits of low temperature heating, as well as how this concept can be applied in new or existing districts. A number of case studies is also presented. Finally, methods for determining distribution thermal losses [9], [36], as well as, friction pressure losses [37], [38] are also available in literature.

2.3. Heat and Cold Consumption

The interface between the district energy and the building systems is commonly referred to as the consumer substation. It consists on a number of control valves, measurement instruments and heat exchangers [16]. The heat exchanger is responsible for transferring heat from the distribution piping, to the consumer system or vice versa, creating a hydraulic separation. Nonetheless, a direct connection is also possible in which the district water is distributed, within the building, directly to terminal equipment such as a fan coil unit. Controls are another major component of the substation. An adequate control strategy can reduce energy costs and optimize conditions for the equipment, while maintaining comfortable temperatures in the building [16].

The consumer's behavior largely influences the energy load [34]. In addition, the energy demand also depends on building constructive elements, such as, walls, windows, size, etc. For designing and operation purposes, the building's global thermal demand is a central calculation. When existing, this parameter should be established based on historical energy usage. Otherwise, the expected building thermal demand may also be predicted, within a range of error.

In order to determine the thermal demand of a building, a dynamic building model is generally used. This model can be implemented in an adequate software and based on a number of inputs, it returns the expected space heating or cooling demand, along with other useful data. Different standard models exist, but it is very popular to use an equivalent circuit model, such as the ones, described in the German standard VDI 6007-1 of 2007 or ISO 13790 [39], [40]. Because of the increased complexity and number of objects, when modelling at district level, the idea is to use the simplest method possible, which requires lesser computational resources, while keeping the results in an acceptable range of error for each application.

Another very important calculation is the building's peak load. The district has to be designed to provide enough capacity to deliver each building's peak thermal needs, during peak periods. If the system is under designed, the customers will be uncomfortable. In turn, if the system is overdesigned, the costs to deliver energy increase, which may affect the economics of the entire system.

In addition to occupant's behavior, internal gains and constructive elements, weather is also a significant factor when determining the above referred parameters. Heating and Cooling demands and peaks depend on conditions like the outdoor temperatures and the durability of these, solar radiation, wind direction or humidity. In fact, humidity is particularly relevant when calculating cooling demands [30].

The cooling demand is basically the amount of heat to be removed from a building, to maintain it at indoor design temperature. As the warmer air to be air-conditioned has greater moisture content than chilled air, a significant amount of cooling load in buildings is attributed to dehumidifying air in the form of latent heat [41]. Figure 2.6 shows an example of a daily profile of latent and sensible cooling loads in an apartment building in Hong Kong, where it can be seen that latent heat is comparable to sensible load during some periods of the day.

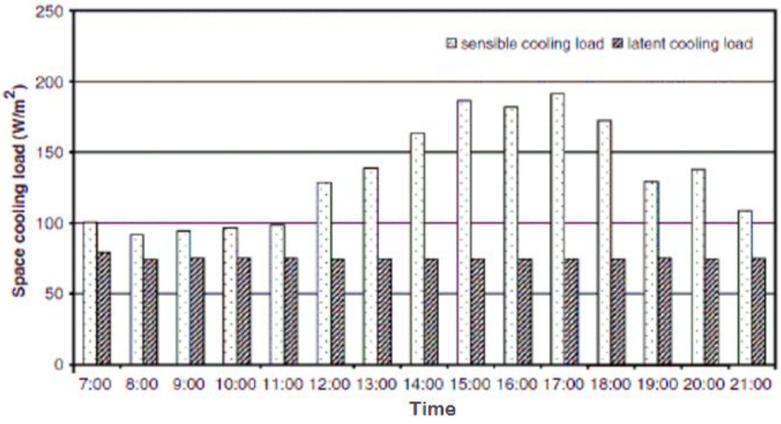

Figure 2.6 - Daily distribution of sensible and latent cooling loads for an apartment building in Hong Kong. [41]

Moisture is mainly introduced into the building through people, equipment and air infiltration [41]. Therefore, a substantial share of the cooling demand in a building is created by heat from the usage inside the building, and not due to the hot weather outside. In fact, contrary to general perception, the location of existing district cooling systems illustrates that the difference in climatic conditions between the countries in Europe does not play such an important role with regard to the specific cooling demand, as DC is actually more frequently established in countries with colder climate, than in warmer climate ones [30].

2.4. Final remarks

Latest generation DE is an energy efficient technology capable of reducing primary energy use and greenhouse gas emissions. This system is very flexible and is designed taking into consideration local resources and potential sources of already existing unused energy. It works like a technology hub, allowing to combine different equipment for greater performance. Moreover, a number of heating and cooling technical solutions are available and can be used, according to which one is more advantageous at each location.

When analyzing a district energy, a number of parameters have to be considered. For instance, the global thermal demand and respective peaks, supply and return temperatures, mass flow rates, pressure requirements or equipment efficiency are important variables.

Finally, as thermal demand density decreases in areas of low-demand buildings, researchers propose new distribution methods, based on low temperature distribution. This approach can reduce primary energy use and thermal losses. In addition, with low temperature heating, heat pumps are more attractive, since its main limitation is that it can only elevate water temperature to a given extent. With this system, heat pumps may efficiently supply both heating and cooling demands, sometimes even without the need for peak units, especially if combined with any low-grade energy source, immune to seasonal variability.

Chapter 3 - Energy Storage

Energy Storage (ES) can be defined as any installation or method, at any place of the power or district system, in which it is possible to store energy, keep it stored, and later retrieve it when necessary [42]. The modern concepts of ES emerged essentially from the energy crisis in the 1970s. More recently, the pursuit for solutions to achieve energy system decarbonization has been attracting increased awareness to its advantages.

First of all, storing energy allows to dissociate energy supply and demand, thus increasing the overall system reliability. This can be particular important since the most limiting aspect of renewable energy production is, its intermittent behavior [42], [43]. In fact, Daim et *al.* [44] refers that in order for renewable energy resources to become completely reliable as primary sources, energy storage is essential.

Moreover, decoupling supply and demand, also result in increased effective generation capacity and energy security. In other words, the demand is rarely constant over time, and the excess generation during low-demand periods can be used to charge a storage unit, therefore increasing the overall capacity for high demand periods. This also implies that cheap energy during off-peak periods is being stored, and can be later recovered. Regarding the production equipment, this process results in a more flattened energy output, which allows smaller capacity to be installed, with lower partial load operation and better efficiency [43].

Another benefit is that energy storage can be utilized to combine different forms of energy, thus conferring greater flexibility. For instance, in recent decades, TES systems have demonstrated the capability to shift electrical heating and cooling loads, from peak to off-peak hours [45]. This process has many advantages both for customers and utilities. Customers have more efficient systems and save money, if they take advantage of different electricity prices, and the same applies to utilities that can spread the demand over more time.

Each storage technology has unique characteristics and is different in terms of its appropriate application and scale. Due to this versatility, the appropriateness of any storage system is generally evaluated, based on certain parameters [42]. Some of these are listed in the following table, along with a brief description.

Table 3.1 - Criteria for selecting energy storage technologies. [44]

Parameter	Description
Power density	Rated output power divided by the volume of the whole storage system.
Energy density	Stored energy divided by the volume of the whole storage system.
Self-discharge	Portion of energy that is dissipated over a given amount of non-use time.
Response time	Speed at which the energy is released or absorbed.
Capacity	The available quantity of energy in the storage system, when charged.
Efficiency	Performance of the process of storing and retrieving energy.
Lifetime	The energy storage cost depends on investment costs and projected lifetime.
Others	Monitoring & control equipment, economics and other operating constrains.

In general, energy storage systems are classified as either thermal or electrical [43]. This project focuses on thermal energy storage, however, a short overview on electrical energy storage is also provided on the next sub-chapter.

3.1 Electrical Energy Storage

Electrical energy storage (EES) embraces a number of technologies, which directly or indirectly, store energy through an electric input and output. As electricity is the most versatile and preferred form of energy for many applications, the importance of further developing this topic is obvious. In fact, it can be crucial to enable the widespread [44] of large-solar or wind-based power generation [46], as well as, electric vehicles, for example.

Electricity can be stored with different methods: [42] Mechanically by pumping water to higher reservoirs, compressing air or increasing the rotational speed of electromagnetic flywheels; chemically by using components such as batteries, flow batteries or fuel cells; and finally, electrically by modifying electric or magnetic fields in capacitors or super conducting magnets. The main EES technologies are grouped by storage method in the next figure.

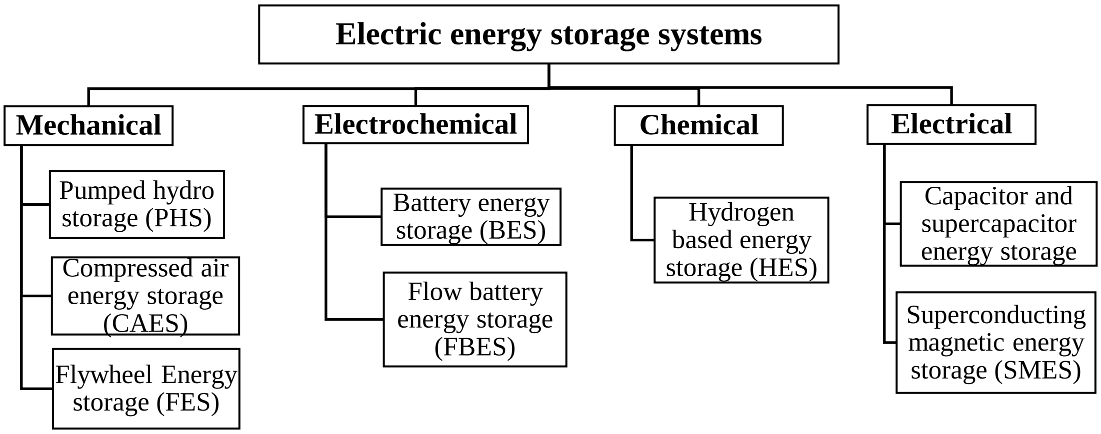

Figure 3.1 - Principal electrical energy storage technologies [44]

Electrical energy storage systems are broadly described in literature [48]–[50]. Among other authors, T. Kousksou et. *al.* [42] released an article providing a comprehensive overview on EES. The study focuses on state of the art systems and installations, as well as, the main characteristics of energy storage. The following table summarizes a small piece of information present in this report. As mentioned before, the suitability of each technology for a specific application is assessed through the analysis of technical characteristics.

Table 3.2 - Comparison of some technical characteristics of electrical energy storage systems [42].

	Efficiency [%]	Capacity [MW]	Energy density [Wh/kg]	Response time	Lifetime [years]
Pumped hydro	75-85	100-5000	0.5-1.5	Fast (ms)	40-60
Compressed air	50-90	3-400	30-60	Fast	20-60
Flywheel	90-95	0.25	10-30	Very fast (< ms)	~15
Battery	60-90	0-40	30-240	Fast	5-20
Flow battery	75-85	0-15	10-50	Very fast	5-15
HES (fuel cells)	20-50	0-50	800-10000	Rapid (< 1 s)	5-15
Capacitors	60-65	0.05	0.05-5	Very fast	~5
Supercapacitors	90-95	2.5-15	2.5-15	Very fast	20+
SMES	95-98	0-10	0.5-5	Very fast	20+

3.2 Thermal Energy Storage

The concept of thermal energy storage is to store energy by cooling, heating, melting or solidifying a material. Various TES categories exist, based on different criteria, such as, storage temperature, duration, etc. [43]. The next figure lists the criteria within the scope of this project, along with the respective categories.

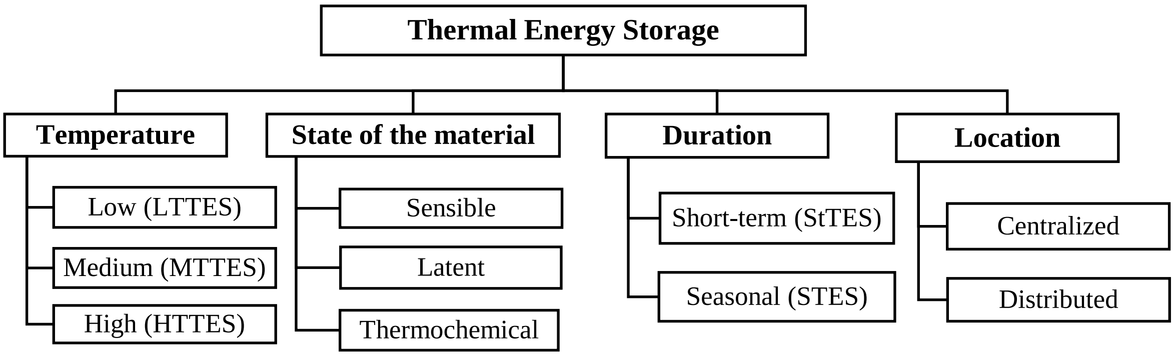

Figure 3.2 - Thermal energy storage categories, according to criteria relevant to this project.

In what concerns the location in the district, TES can either be centralized or decentralized. The first is generally a large capacity unit, connected to all or the majority of the components in the system. The decentralized or distributed approach is when storage is applied in a smaller scale, associated to a particular building or specific part of the network. Both concepts are analyzed in this project. In addition, regarding the temperature level, this project focuses on low temperature thermal energy storage (LTTES), therefore medium and high temperature TES will not be addressed. Thermal energy storage can also be classified according to the state of the storage material. The next paragraphs provide an overview on sensible, latent and thermochemical TES.

In sensible storage systems, energy is stored by changing the temperature of a storage medium. The amount of energy stored in the medium is given by [43]:

$$\mathrm{Q} = \mathrm{m} \cdot c_p \cdot (T_f - T_i) \qquad [\mathrm{J}] \qquad (3.1)$$

Where, m is the mass of the storage medium [kg], c_p is the medium's specific heat capacity [$J \cdot Kg^{-1} \cdot K^{-1}$], T_i is the medium initial temperature [ºC] and T_f is the final temperature [ºC]. For some systems, it is also useful to describe mass as function of its volume (V) [m^3] and density (ρ) [$kg \cdot m^{-3}$], as follows:

$$\mathrm{m} = \mathrm{V} \cdot \rho \qquad [\mathrm{kg}] \qquad (3.2)$$

In addition, the maximum storage capacity may also be determined using equation 3.1, if the final and initial temperatures are replaced by the fully charged and fully discharged temperatures, respectively [19].

Different authors [19], [50] report that the storage medium is essentially determined based on the required temperature level, storage duration, and rate at which energy must be released or extracted. For this reason, the choice of sensible storage materials is very dependent on the following properties:

- *Volumetric thermal capacity* ($\rho \cdot c_p$), which quantifies the ability of a given volume of material to store sensible thermal energy;
- *Thermal Diffusivity* ($\lambda \cdot \rho^{-1} \cdot c_p^{-1}$), which measures the thermal inertia of a material. A bigger value means that the material will rapidly react to a temperature change.

In addition to high volumetric thermal capacity and low thermal diffusivity values, stability when charging and discharging and low material costs, are also important characteristics of sensible storage mediums. In this context, water, concrete, rocks, gravel, sand or the ground itself are reliable, simple and low-cost materials for storing energy [50].

Among the above referred mediums, water is considered to be the most favorable material for sensible energy storage due to its high specific heat and high capacity rate, while being charged and discharged. Even so, all mediums have their own advantages and disadvantages [19]. Basically, the thermal capacity of liquids is higher than that for solids. In turn, solids can tolerate a higher range of temperatures since they will not freeze or boil, and these cannot leak from the container.

In short, sensible TES is a matured method, applied in different contexts, generally related to space heating and cooling or domestic hot water supply. The required temperatures for these applications range from 15 – 60 ºC, in low temperature systems. This makes water and other rock sort materials very attractive, as sensible storage mediums. Over the past years, a number of studies have been carried out focusing on sensible thermal storage. The major part of the efforts aimed to the resolution of specific issues (e.g. new water tank concepts) and new materials [50], [51].

Another method to store thermal energy is in latent form, using phase change materials (PCMs). This technology can offer higher energy density than sensible storage, however it stills in a yearly stage of development, especially in what concerns its implementation in district systems [52]. The energy is stored as heat of fusion when melting occurs, at a nearly constant temperature (see figure 3.3 – left). If the temperature decreases to the phase change temperature, the material changes back to solid phase, and releases the stored energy. The amount of energy stored in PCMs is given by [53]:

$$\mathrm{Q} = \mathrm{m} \cdot \left[c_{p-solid}(T_m - T_i) + a_m \cdot \Delta h + c_{p-liquid}\left(T_f - T_m\right)\right] \qquad [\mathrm{J}] \qquad (3.3)$$

Where T_i, T_m, T_f are the initial, melting and final temperatures [ºC], respectively, c_p is the specific heat capacity [$J \cdot kg^{-1} \cdot K^{-1}$], a_m is the melted fraction of the material, and Δh is the enthalpy change at the phase change temperature [$J \cdot kg^{-1}$]. Materials used for storing energy in latent form (PCMs), should have a high heat of fusion and a phase change temperature within the operating range of the thermal system. The right side of figure 3.3, shows various PCMs and their phase change temperatures [54].

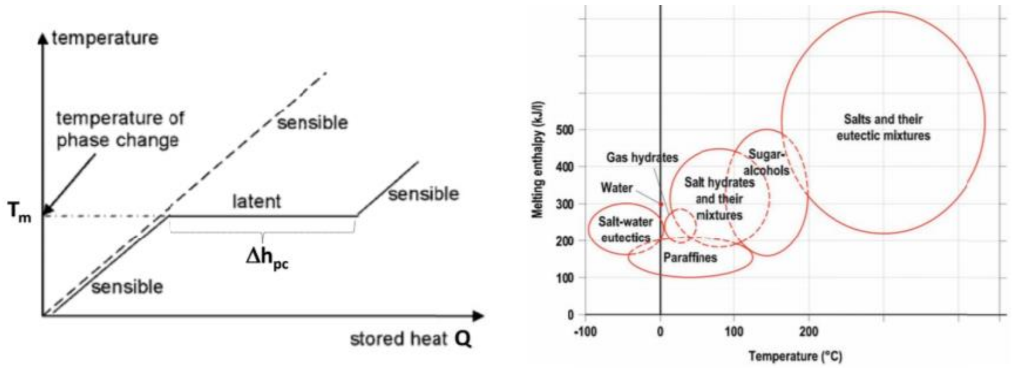

Figure 3.3 – Left - Temperature and stored heat for a solid-liquid phase change, compared with sensible heat [53]; Right - Classes of materials that used as PCMs and their typical range of phase change temperatures and melting enthalpy [54].

Based on the literature collected, appears to be potential for the integration of latent thermal storage in district networks. Nonetheless, at this scale this technology is very undeveloped, as most of present work is based on small scale prototypes or in computational models [52]. Examples of this line of work are the five PCM-related projects resulting from IEA-SHC task 32 [55]. According to the task participants, screening for better PCMs with higher heats of fusion and heat transfer rates, should be addressed in the future. By contrast, ice storage for cooling purposes is already a relatively mature latent technology, currently in commercialization phase [43].

Finally, a more recent method to store thermal energy is thermochemical energy storage. It is based on reversible chemical reactions between two substances, which are energy demanding in one direction and energy yielding in the reversed way. No reaction occurs as long as the two materials are stored separately, allowing to store thermal energy until the two substances are mixed together [53].

This method is seen as the most promising alternative to store thermal energy. With this technology, it is possible to store energy at ambient temperature, for the desired duration, with negligible heat losses, making these systems suitable for long-term applications. In addition, it has the highest energy density when compared to sensible and latent solutions [52].

Current work focuses on developing new stable, non-polluting and cost-effective materials [53]. However, this technology is still on yearly stages of development. Prototypes and demonstration projects have already been developed, mainly related to long-term thermal storage. Examples of these are the watergy and SWEAT (Salt water energy accumulation and transformation) projects [52].

The case study considered in this project focus on sensible thermal storage. This method is used for both short-term and long-term applications. The following sub-sections address this topic.

3.2.1. Short-term thermal storage

A very common technical solution for short term sensible TES is the water tank, often operating in daily storage cycles. Tanks are ideal for water storage since they are cheap and easy to produce. The major challenge with water tanks however, is the space they occupy. In fact, the available space plays a decisive role when sizing the storage unit, particularly in domestic applications [17]. By increasing the temperature of one kilogram of water 1ºC, its energy content increases approximately 4.18 kJ. Similarly, in order to cool the same amount by 1 degree, an equal quantity of energy is released/absorbed. To illustrate, for a temperature difference of 20ºC, the storage capacity is approximately 84 kJ/kg. This means that, to deliver a heat output of 1 kW a full week, a water tank of about 7200 liters is needed.

In addition to space availability, the optimum size of the storage also depends on other factors, such as, weather, building heat loads, equipment efficiencies, temperature levels, costs, and most importantly, the application for which is designed [56]. Short-term TES can be used to optimize energy systems in different ways and scales.

To begin with, it may be utilized to meet fast fluctuations on the thermal demand, which is particularly useful when the heating or cooling equipment has slow response time, as it is the case with some boilers, for instance. Moreover, thermal storage is often combined with solar thermal technologies. In this context, it can help to neutralize the instability of the weather, as for example, the appearance of clouds [17].

Furthermore, the capacity of heating and cooling equipment is normally selected to match part of a design-day load, when the requirements for heating or cooling are close to maximum. However, these peak design loads occur for only short periods of time, resulting in excess capacity on average days. In new systems or retrofit interventions, short-term storage allows to select the capacity of the thermal equipment closer to the average, rather than peak conditions [43]. This not only improves the utilization of base load generating equipment, but also reduce the reliance on peaking units that have high operating costs.

Another example of a short-term TES application, is the use of thermal storage to shift electrical heating and cooling loads from peak to off-peak periods [50]. Producing equipment can operate at night, when the global demand and cost of electricity are relatively low. The stored energy can then be used throughout the day, when demand and costs are higher. This does not necessary result in lower energy consumption, but reduce the peak power demand of the overall network. In addition, the final consumers can take advantage of the low electricity tariffs.

A. Arteconi *et al.* [45] reviewed state of the art technologies and methods for demand-side management (DSM), which is defined as "the planning, implementation and monitoring of those utility activities designed to influence customer use of electricity, in ways that will produce desired changes in the utility's load shape". These authors refer that, there is a significant potential for short-term TES to manage electrical loads of electric heating and cooling equipment, as it emphasizes figure 3.4. The potential is even bigger in the tertiary sector, in which electrical heating and cooling account for about 44% of total electricity consumption, among end-use equipment in the EU-15.

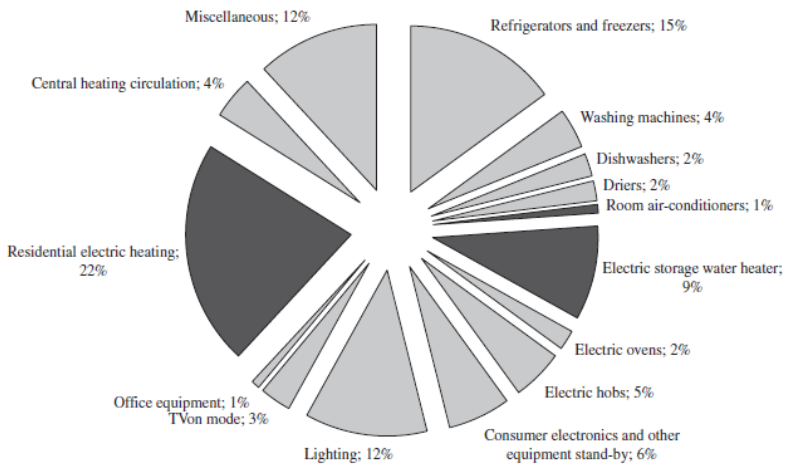

Figure 3.4 - Breakdown of electricity consumption among residential end-use equipment in EU-15 (black slices represent electricity consumption for heating and cooling that can be managed by TES) [45].

An example of a real DSM application, is the peak reduction using residential hot water heaters, in France [6]. In this country, more than one-third of the households use electric water heaters equipped with meters, that allow customers to respond to the country's peak-pricing structure. A remote start/stop function option that allows grid operators to remotely control these water heaters, as well as consumer information campaigns on electricity pricing structures, were crucial to achieve results. The consumers we encouraged to use less energy during peak hours, by offering them more attractive electricity tariffs during off-peak periods. Nowadays, French utilities claim that TES has helped the country to optimize the use of the nation's generation capacity.

A similar demonstration project was conducted by a Danish electricity transmission and systems operator (Energinet.dk), involving 300 homes with heat pumps. The goal was to analyze the potential for operating heat pumps with increased flexibility by having hot water stores, while creating a dynamic demand responding to electricity prices [17].

Shifting the peak load in time is often called "peak-shifting" or "peak shaving" and can be made essentially according to three operational strategies [50], [57], proposed and discussed by I. Dincer [43]. The strategies are full-storage shift, partial-storage load leveling and partial-storage demand limiting.

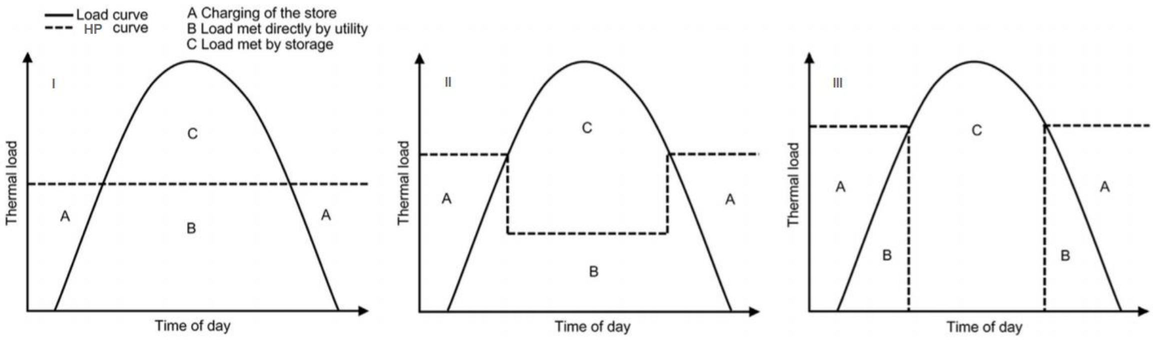

Figure 3.5 - Strategies for peak shifting: (I) partial-storage load leveling; (II) partial-storage demand limiting and (III) full storage [57].

A full storage or load-shifting strategy shifts the entire load to off-peak periods (Fig. 3.5-III). In this method, the producing equipment is designed to operate at full capacity during all or the majority of non-peak hours, in order to charge the TES. The demand is then met by discharging the storage, while heating and cooling equipment is idle. However, for this to be possible, the capacity of energy producing equipment is generally selected according to the design-day load. Additionally, a fairly large storage volume is often required. This method can be advantageous in systems where the peak loads are high, but short in duration and when there is a significant discrepancy between peak and off-peak tariffs.

For example, Dincer refers [50] that a commercial building whose electrical demands drops substantially after five o'clock in the afternoon (17:00), in an electric utility territory where peak tariffs apply between 13:00 and 21:00, usually can economically apply a full-storage strategy. In other words, the thermal demands during the four peak hours period between 13:00 and 17:00, can be met with a cost effective short-term thermal storage unit, without oversizing energy producing equipment. On the other hand, busy facilities such as hotels or hospitals are less likely candidates for full-storage, as their load profiles are continuous and flatter. Less time is, therefore, available for charging the storage. However, these buildings are often suitable to partial storage strategies.

In a partial-storage method, the producing equipment also operates during peak periods, in order to meet part of the thermal load. The rest is met by drawing energy from the storage, that was produced during off-peak times. This allows to install heating or cooling units with smaller capacity than the design load. In addition, the equipment will also have more operating hours, as the demand can be distributed for more time.

Partial storage TES may operate as demand-limiting or load-leveling. In a demand-limiting strategy (Fig. 3.5-II), the producing machinery operates at a lower capacity during peak hours, and is often controlled to limit the peak demand. In this case, the sizes of the storage and production equipment are chosen to get the most cost-effective solution, regarding peak load characteristics and electricity tariffs. This is most advantageous where the peak loads are much higher than the average load.

When using a load leveling strategy (Fig. 3.5-I), heating and cooling equipment is designed to work continuously, at a more or less constant power over the entire day. In this case, the required installed thermal capacity may be lower, therefore cheaper, and the size can be chosen so that energy producing units can operate at nominal efficiency most of the time.

Partial storage is generally the most economic option, representing the majority of thermal storage installations. This method allows to incorporate smaller equipment with reduced costs. In addition, during operation at less than peak design loads, partial storage can work as full-storage systems. For instance, if a partial load TES is designed for heating in Winter, the same storage unit may be used to shift the entire load (full-storage) on some milder spring or autumn days.

3.2.2. Seasonal thermal storage

Seasonal thermal energy storage (STES) is a technology used to store heat or cold, for periods up to several months. It is most commonly implemented in district energy systems or greenhouses, often combined with solar heating or heat pumps [52]. The idea is to store thermal energy from season to season, for instance, collecting and storing heat during summer to later use it in space heating, throughout cold periods.

Sensible, latent and thermochemical STES concepts exist, however, sensible is the most implemented method, as it is more mature, reliable and inexpensive at the present. This technical solution is referred to have greater potential in practical applications, than short-term storage, however it is more technologically challenging [52]. It requires larger storage volumes and has greater risk of heat losses. As in any TES application but with particular importance for large STES projects, the material chosen for storage must be economical, reliable and ecological. As a result, water and rock-sort materials, such as gravel, pebbles, and bricks, have been largely selected for storage media, in large-scale projects.

At a depth of eight to nine meter or more, the ground temperature is more or less constant throughout the whole year, having a value close to that of the average annual air temperature [58]. Therefore, the ground can act as a natural insulator, and for this reason, seasonal storage technologies often consist on underground or buried systems. The most common seasonal storage concepts are the water tank storage, the rock bed storage, the gravel-water pit storage, the duct thermal energy storage (DTES), and the aquifer thermal energy storage (ATES) [19], [52].

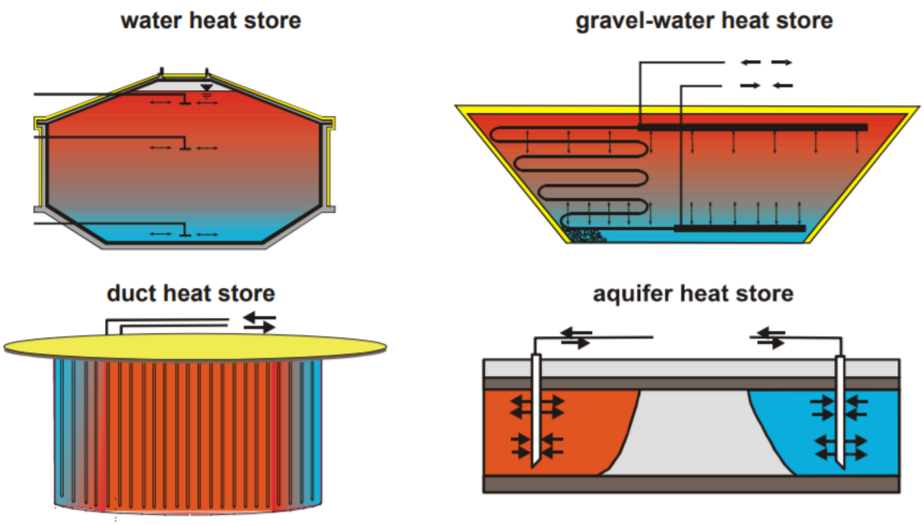

Figure 3.6 - Schematic representation of common technologies for seasonal thermal energy storage. [59]

Water tanks are artificial structures generally made of stainless steel or reinforced concrete. They are usually surrounded by thick insulation and buried underground, in order to decrease heat losses. Tanks operate in a stratified manner with water at the top of the tank being hotter that in the bottom, due to thermal buoyancy. The subsequent mixing effect caused by temperature difference, may negatively affect the maximum and minimum temperatures, leading to loss of efficiency. To minimize this phenomenon, current research focus on methods for maintaining the inside water on a stable thermal stratification. In addition to this, another popular research in water tank storage lies in the reduction of heat losses, by having new tank designs and insulation materials [52].

In rock bed storage, the rock (pebble, gravel or bricks) bed is usually circulated with a heat transfer fluid (water or air) to exchange heat, gained in summer and released in winter. Compared to water, rocks can endure much higher temperatures. However, due to the low energy density, rock systems require much larger volumes to achieve the same amount of thermal energy storage. In this context, the gravel/water concept was proposed. In this storage system, both water and rock are used as storage mediums. This can be viewed as a compromise between the proprieties of water and rocks. Literature refers that this technology could be operated at temperatures up to 90ºC [52].

In duct thermal energy storage, the ground itself is directly used as storage medium. The ground is excavated and drilled to insert vertical or horizontal tubes. These serve as heat exchangers, the soil is the storage material, and water is generally the energy carrier. This method allows to store both heat and cold. The energy is stored in a radial manner, with the cold being stored farther from the center, and the heat in the center [59]. The temperature levels at which operates ranges from -3 to 20 ºC [19]. However, due to its low energy storage density when compared to water tanks, this system requires 3 to 5 times more volume to store the same amount energy (table 3.3). In addition, literature refers that another disadvantage of this system is the small heat transfer rate, when charging and discharging [52].

Similar to DTES, in an aquifer thermal energy storage, wells are drilled into a suitable aquifer in order to install pipes and inject or extract groundwater, using water pumps. There are hot and cold wells, as this technology allows to store both heat and cold. These are separated in order to keep the warm and cold water from mixing. Additionally, from the literature collected [60], [61], ATES usually work within the temperature range of 5-35 ºC, which means that it is possible to supply direct cooling without the need of producing equipment [62]. Furthermore, as in this case water is used to store thermal energy, the heat transfer rate is much more efficient when compared to DTES.

In what regards the operational strategy, during summer water is extracted from a cold well, heated by a chosen heat source, and injected into a hot well. Later, in the heating season, the cycle is reversed and the hot water is extracted from the warm well, cooled by a heat sink and injected into the cold well.

In contrary to water or gravel-water tanks that can be built at almost every location, DTES and ATES require special geological conditions. In addition, these systems need a larger volume (table 3.3) and 3 to 4 years to reach the typical performances [19], [60], after implementation. However, these are cost-effective solutions, capable of storing both heat and cold, with relatively low maintenance.

Table 3.3 - Energy storage density and medium of the most common seasonal storage systems. [19]

	Water tank	**Gravel-water**	**Duct TES**	**Aquifer TES**
Storage medium	Water	Water/ gravel	Soil/Rock	Water, sand/ gravel
Energy storage density	60-80 kWh·m^{-3}	30-50 kWh·m^{-3}	15-30 kWh·m^{-3}	30-40 kWh·m^{-3}

3.3. Combination of TES and Heat Pumps

In this project, space heating and cooling supplied by a combination of STES, short-term TES and heat pumps, is covered. In particular, an aquifer thermal energy storage is used as low-grade thermal sink and source for heat pumps (ATES-HP).

As mentioned in last chapter, the operation of ATES means that water is extracted from a well and is heated or cooled before being re-injected into the same aquifer, which allows to store both heat and cold (Figure 3.7). A heat pump in cooling mode can charge the ATES by absorbing the excess heat in the buildings, and injecting it into the warm well. Later, during cold months, the heat stored is extracted and used as low-grade heat source for the heat pump, when producing heat. Throughout the winter, by extracting warm water and injecting cold water resulting from heat production, the aquifer's surrounding soil and water temperature decreases, which will be beneficial for the next summer. As the heat pump runs year by year, the heat charge and discharge of the aquifer will become periodic.

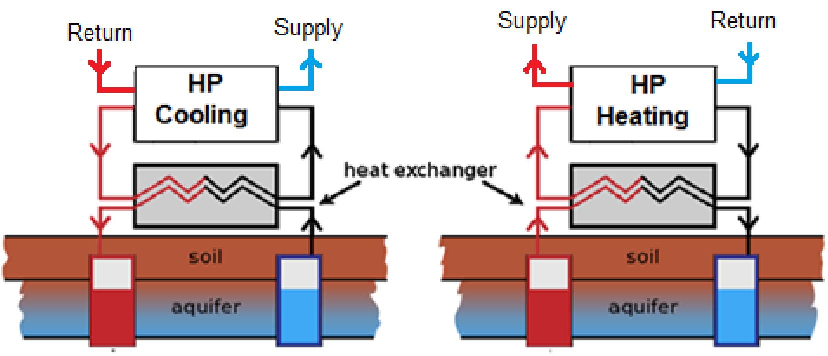

Figure 3.7 - Schematic representation of an ATES system operation when combined with heat pump: (left) – heat injection and cold extraction (Summer); (Right) – heat extraction and cold injection (Winter) [63].

In essence, this operation allows to store excess thermal energy characteristic of each period of the year, and use it during the opposing season, in a cycle. This greatly improves the performance of heat and cold supply using heat pumps. Firstly, by using ATES as sink, energy that would otherwise be wasted is being stored from season to season. Secondly, the energy stored is used as thermal source for the heat pump, greatly improving its coefficient of performance. To illustrate, a combination of aquifer thermal storage and water-to-water heat pumps is analyzed by Paksoy *et. al.* [64], and they referred a 60 % increase in the COP of the water-to-water heat pump, when compared to the COP of a conventional HP, using ambient air. Moreover, Tanaka *et. al.* [10] evaluated the performance of three district energy configurations, one containing no storage, a second system with STES and a third system based on short-term energy recovery. They found that seasonal storage could reduce energy consumption by about 26%, while the short-term system could decrease it by about 16%, when compared with the system with no thermal storage installed.

In addition to the ones above referred, other authors focused on heat pumps in combination with seasonal thermal storage, particularly in the context of low-temperature heating. Having a low-temperature heating system, such as, floor heating or low temperature radiators, combined with low-temperature storage can lead to more efficient systems. The lower required temperatures result in reduced heat losses and can also further support the heat pump in terms of COP, as it needs less compressor power to upgrade the temperature to a suitable level.

The main objective of this project is to assess the performance of a district energy containing an ATES-HP system, when short-term storage is also used for peak shifting applications. Therefore, the short-term TES may influence the amount of energy extracted and injected into the aquifer. It is important to have a balance between the amount of heat and cold stored, in order to keep storage temperatures in a periodic cycle and within acceptable values. However, as it was described in chapter 3, short-term TES is also very beneficial to energy systems. In addition to all the already referred advantages of short-term storage (lower required installed capacity, peak-shifting, etc.), when combined with a heat pump the operation can be further optimized. Using heat pumps in low temperature heating systems, allows to considerable reduce the storage discharge temperatures, and it is claimed [19] that this reduction helps keeping the water stratified, which is beneficial.

In short, combining seasonal and short-term thermal storage with heat pumps allows to optimize the performance of an energy system as a whole. This have many advantages for both small and large applications. This thesis addresses an ATES, however any other STES method may be equally used in combination with low temperature heating or cooling equipment, resulting in lower energy costs and greater efficiency.

Chapter 4 - Case Study and System Configurations

This chapter describes the case study neighborhood that is used in this project. It is based on a layout adapted from IEA-EBC Annex 60 Programme [11], which consisted on modelling a set of buildings and energy systems at the district level. Moreover, it discusses the three configurations used to provide space heating and cooling to the case study neighborhood, as well as the relevant performance indicators, in which conclusions are based. Finally, the software tools and methods used to model the buildings and energy systems are presented.

4.1. Neighborhood case study

In this thesis, the neighborhood case study proposed in annex 60 common exercise was adapted to create a small set of buildings. It consists of 14 residential and 2 office buildings, which are combined to form the single street section schematized in the next figure.

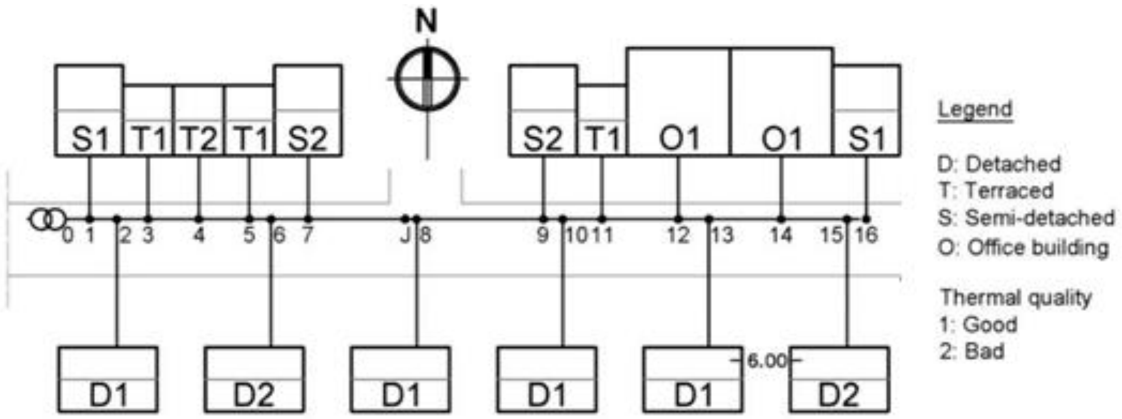

Figure 4.1 - Neighborhood case-study layout.

As defined in annex 60 common exercise, the buildings are classified in four typologies: detached (D), semi-detached (S) and terraced (T) houses, as well as office (O) buildings. Furthermore, two sub-topologies are made to distinguish between the levels of thermal insulation: (1) denoting a level of insulation required by the EPB (Energy Performance of Buildings directive) of 2010 in Flanders, and (2) representing a level of insulation similar to the one found on buildings of the period 1946-1970 in Flanders, based on the IEE Tabula (Typology Approach for Building Stock Energy Assessment) project [65].

All residential buildings are composed by a ground floor, where the occupants are during day time (day-zone), a 1st floor with the bedrooms (night zones), and one unheated attic. In what regards the office building typology, it has 5 floors, and each one consists of two working areas in opposing façades, separated by an unheated hallway.

Information such as district distribution distances between the connection nodes of the different houses, detailed blueprints of all typologies, as well as insulation values and other parameters related to construction elements, were also provided in the common exercise paper. Since this document is no longer available online, the most relevant information is summarized in appendix A of this report.

4.2. System Configurations

At this point it may be useful to remind the global goals of this thesis. In short, the main objectives are:

a. Investigate in terms of electricity costs and energy performance, the impact of having decentralized short-term TES for peak-shifting applications, in a DE containing seasonal TES utilized as sink and source for water-to-water heat pumps;
b. Compare in terms of electricity costs and energy performance, the decentralized system approach referred in (a), with a full centralized thermal energy production and storage method.

In order to this, a virtual test environment consisting on a function based MATLAB numerical model is created. Three DE system configurations are implemented, simulated, and a number of pre-defined performance indicators are obtained and compared. The configurations defined in this project are (figure 4.2):

- **Reference Case**, that consists on using centralized aquifer thermal storage (ATES) as sink and source for decentralized heat pumps at each building;
- **System 1**, which is the same configuration as in reference case, however in this case, each building also has a water tank used for peak-shifting applications;
- **System 2**, in which thermal energy is produced in a central plant away from consumers, using a large capacity heat pump connected to the ATES, and a large buried water tank.

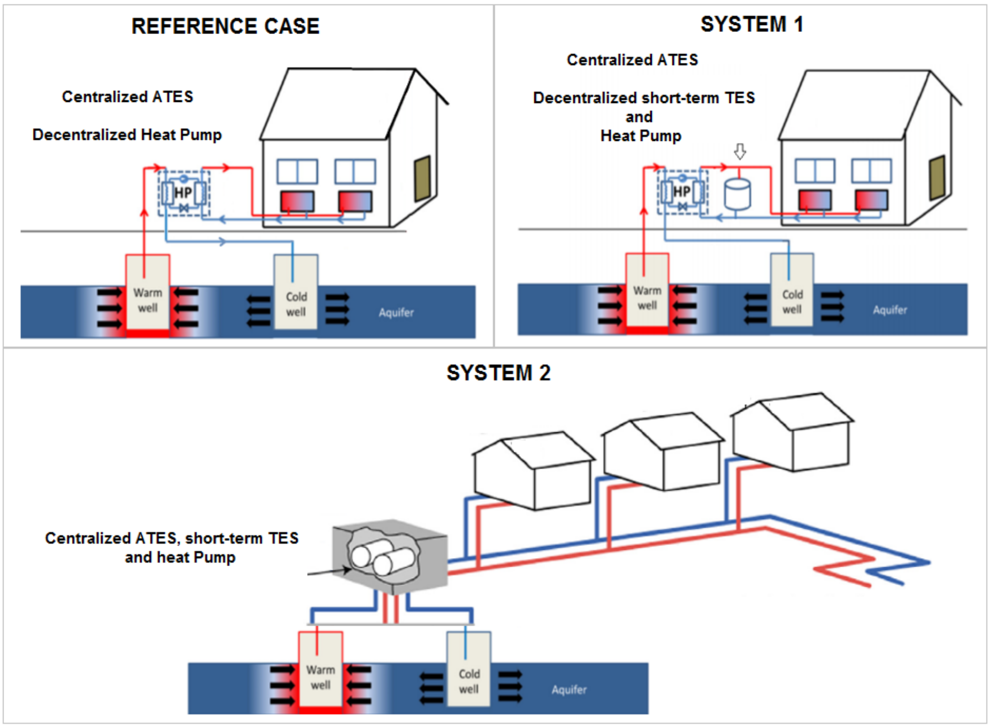

Figure 4.2 – Configurations used to supply the space heating and cooling requirements of the neighborhood considered in this project. Adapted from [19] and [52].

The reference case system is based on the DE used to supply the thermal needs of the buildings in Technical University of Eindhoven (TU/e) campus. As the name implies, this configuration is meant to provide reference performance indicator values, representing a conventional system without short-term storage.

On the contrary, in system 1 each building has a short-term storage unit. In this case, controls are applied in order to operate heat pumps in a more continuous way. The control strategy should allow the system to react to electricity prices, in order to take advantage of low electricity tariffs during off-peak periods, while minimizing on/off switching and maximizing efficiency.

By simulating the reference case and system 1 under the same conditions, then comparing a set of pre-established representative performance indicator values, the impact of adding decentralized short-term TES can be measured/quantified. The same applies to system 2, that can be compared to system 1 in order to achieve the objective referred in (b).

The most relevant performance indicators used in this project are:

- Heat pump's coefficient of performance (COP);
- Electricity consumption (for energy production and water pumping);
- Thermal energy production;
- Energy injected/extracted from ATES;
- Electricity costs.

Furthermore, in order to deal with the complexity and high number of components present when analyzing an energy system at the district scale, computational software is used as a tool to determine the referred performance indicators. This subject is the topic of the next section.

4.3. Simulation software tools

The analysis of an energy system at the district scale involves a large number of complex multidisciplinary components. For this matter, computational models implemented in simulation software, are often used by researchers and companies, as tools to deal with the complexity of reality. Furthermore, the approach taken to investigate energy systems at the neighborhood level, should be different from the one adopted when studying smaller and more specific components.

Literature collected on this topic [39], [40], [66], points advantages on modelling large scale systems by using simplified methods and assumptions, which can produce simulation results accurate enough for the purpose of study, while keeping development and simulation time within an acceptable range. In addition, authors refer that it is helpful to use modular approaches with interconnected models of the different subsystems. This way, it is possible to combine and modify any subsystem as required, without the need to make changes in other parts of the whole model.

Moreover, it was noticed that researchers in this field usually adopt an object-oriented modelling philosophy. First of all, it facilitates the reusability of models. Furthermore, it eliminates the need for routine development to input and output values, and for the management of large amount of data involved in building and energy systems simulations, therefore leading to smaller code size. Additionally, object-oriented programming, generally allows to have a more natural representation of systems and an increased level of modularization, simplifying the process of modelling different disciplines, such as electrical, thermodynamics or control systems.

Based on the above referred characteristics, Dymola (Dynamic Modeling Laboratory) simulation environment [67], using Modelica programming language, was initially selected for this project. Modelica [68] is an open-source object-oriented modelling language, used to model dynamic multi-domain physical and control systems. It allows the reusability of models and components. In fact, various open source libraries are available, providing a large number of models and functions, from many domains. Examples of this are the Modelica Standard Library (MSL) or the Buildings Library developed by Lawrence National Laboratory (LBNL). This allows users to save development time, and to focus more on physical process, rather than on their mathematical expressions.

Following this, the initial plan was to determine building's space heating and cooling demands, as well as the performance criteria concerning the different energy system configurations, using Dymola. Unfortunately, incompatibilities resulting from combining models from different libraries during implementation, as well as having models with a level of detail superior to the required for this study, led to unsustainable development and simulation time. Because of this, and also forced by the fact that Dymola's license was soon expiring, it was decided to take a different approach. The thermal energy demand data related to the neighborhood buildings, that had already been successfully determined using Dymola/Modelica, were kept. However, another simulation software had to be selected to evaluate the energy performance of the different system configurations, using the already determined demand profiles as external inputs (figure 4.3).

Various suitable simulation software were considered at this point. The choice was between OpenModelica, TRNSYS or MATLAB. Due to software costs, lack of suitable libraries or step learning curves for the available time, it was concluded that MATLAB (Matrix Laboratory) was the best option to take. This decision was also taken in light of my previous experience with this software, and by the fact that an already existing software license, was available.

MATLAB is a multi-paradigm numerical computing environment developed by MathWorks. It provides an easy-to-use interface, where problems and solutions can be expressed as they are written mathematically. In addition, a number of accessory tools and packages are available, as for example, Simulink, which adds multi-domain simulation capabilities and a model-based design for dynamic systems. However, this time, it was decided to take the simplest approach possible for the purpose of this study. Therefore, an analytical approach is the main method used to determine the required system performance indicators. This is done by executing text files with MATLAB code (scripts and functions). Through a number of sequential operations, each text file is used to conduct steady-state calculations concerning the various aspects of the district energy and configurations assumed.

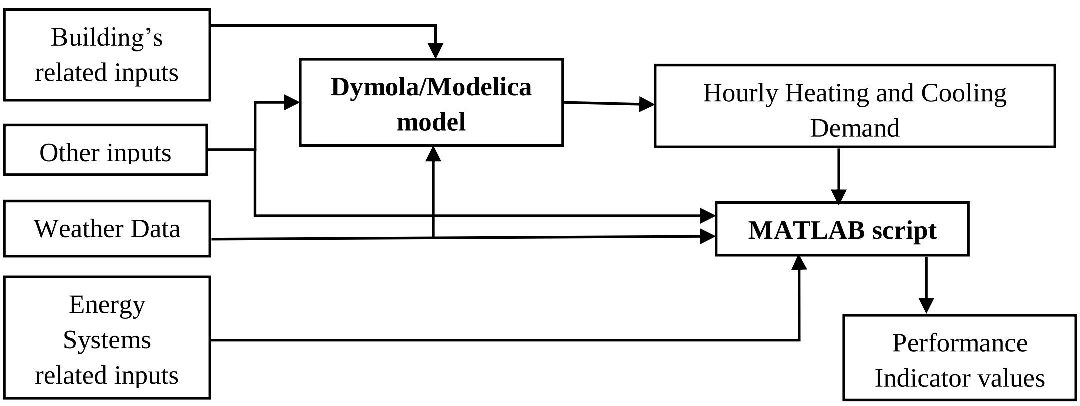

Figure 4.3 – Chain of the described global methodology.

Chapter 5 - Implementation in Dymola and MATLAB

The methodology followed to implement the components of the essential district energy aspects is presented and described in this chapter. Firstly, the Dymola model used to determine heating and cooling demands, resulting from simulating the different neighborhood case study buildings is addressed. As mentioned before, each building typology has its own construction and usage characteristics.

Furthermore, energy systems implementation in MATLAB is also discussed. This is done by gradually presenting the main components of the district energy. It begins with heat and cold production, in which heat pump and free cooling calculations are performed. After this, there is a sub-section presenting the methodology followed to implement thermal energy storage, and its respective control strategies. Afterwards, the matlab code responsible for executing thermal and hydraulic calculations regarding energy distribution, is described.

In short, each script (text file containing matlab code) is programmed to use a number of inputs, in order to determine the required output values. Both MATLAB scripts and functions allow to reuse sequences of commands, by storing them in program files. Scripts are the simplest type of program, since they store commands exactly as they would be typed in the command line. In turn, functions are more flexible, more easily extensible, and they remove the need to manually update the script, each time that a specific input value changes. In this project, main functions, and secondary functions are implemented. The first are used to represent large parts of the system, as for instance the entire heating calculations, while secondary functions are nested inside main functions, and are used to perform accessory operations. In addition, matlab functions are also used in this project to share calculated data between the different scripts.

Finally, once all individual components of the computational model are specified, the last part of this chapter consists on discussing how matlab functions are called and combined together to form the complete system script. When executed, this file determines the performance indicators concerning each district configuration.

5.1. Heating and Cooling Demand

During the literature review phase, it was possible to verify that there are many possible ways to determine space heating and cooling demands. While some authors take very simple procedures [60], others adopt more complex methods [40] [66].

The first approach to represent the buildings of the neighborhood case study was based on a simple RC (resistance-capacitance) thermal model. The implementation in Dymola/Modelica was carried out, mainly using existing blocks from the Modelica Standard Library 3.2.1 [69]. The model (fig. 5.1) consists on a simplified mixed volume of air, a prescribed heat source for the internal gains, and a heat conductor for steady-state heat conduction to the outside. The convective heat transfer coefficient is lumped into the heat conductor model. Moreover, a temperature sensor block is placed in this model in order to measure the indoor air temperature, that is used to regulate heating and cooling operation. To arrive at the building's demand, the model calculates the amount of heat input necessary to keep the temperature of the mixed volume of air, at a given temperature set-point.

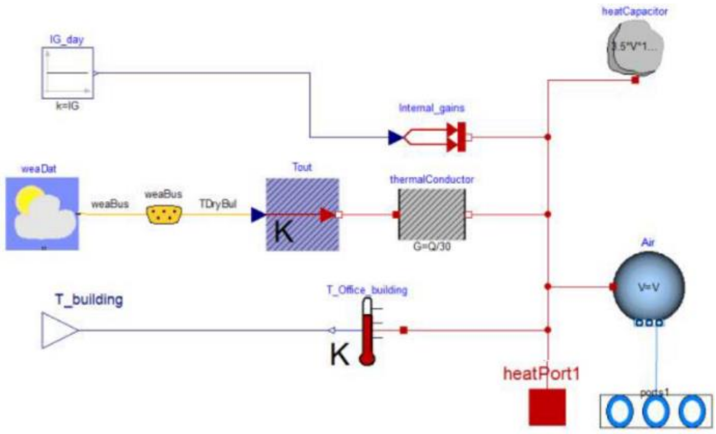

Figure 5.1 - Schematic view of the RC building model in Dymola (initial approach).

Space heating and cooling demands resulting from this model were compared with another set of results provided by an author [70], who was also investigating the thermal demand of the same neighborhood case study. In his project [70], he implements a more complex model using packages from Buildings library (Lawrence Berkley Laboratory). This open-source library basically meets the needs for simulation in building systems. It provides users with a collection of models that can be used, extended and adjusted as needed, making it a very powerful tool when simulating heating, ventilation, air-conditioning (HVAC) and control systems.

Throughout the comparisons, it was noticed that there were large discrepancies between the demand results obtained from each model. In addition, several attempts to calibrate the RC model were made, only achieving however, the reduction of a small margin of error. At this point, it was decided to use his implementation methodology instead, since it utilizes validated packages and is certainly more accurate. A complete description of the implementation is provided in his master thesis report [70], however, an overview is presented in the next paragraphs.

The Modelica model used to determine space heating and cooling demands in this project, is hierarchically structured in two layers (fig. 5.2). First of all, the bottom layer contains the block responsible for computing heat balance inside the building. Each building is defined using two thermal zones (night and day), each one implemented with the *ThermalZones.MixedAir* model (fig. 5.3), from buildings library. Additionally, all building's related parameters such as geometry, material properties and infiltration rates, are here declared and inputted to the mixed air model. Further information on the mixed air model itself, or on how it can be employed, may be consulted on the buildings library documentation website [71] and on this [72] conference proceedings report.

Secondly, in the top layer, other control based components or inputs such as weather data, occupancy profiles, internal gains, surrounding soil temperature and shading control, are implemented. These are also inputs of the *ThermalZones.MixedAir* model. This layer also contains the block responsible for calculating heating and cooling demands. It receives feedback data from the heat balance model, and determines the amount of heat input/output required to keep the indoor air temperature above or below a given temperature set-point, depending respectively on whether heating or cooling is necessary. The total building demand is calculated by summing up the demand from all the heat ports/nodes of all its thermal zones.

As said before, the neighborhood case study contains buildings with different usage and construction characteristics. This structured implementation facilitates when dealing with this amount of input data. Moreover, it also simplifies the reusability or re-adjustability of models.

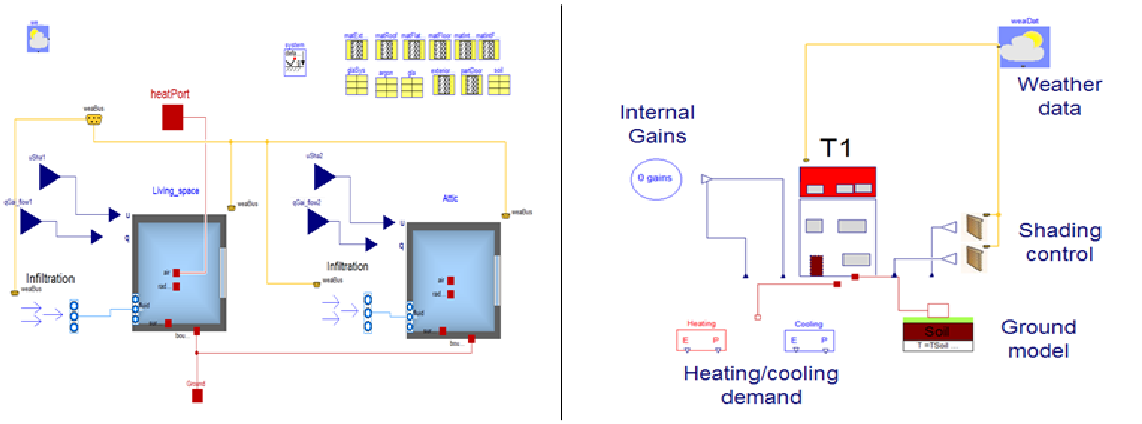

Figure 5.2 – Model structure in Dymola: (left) is the bottom layer, and (right) is the top layer. The dark blue lines refer to internal gains and controls, blue represent fluids, red heat transfer, and yellow weather data. Source: [70]

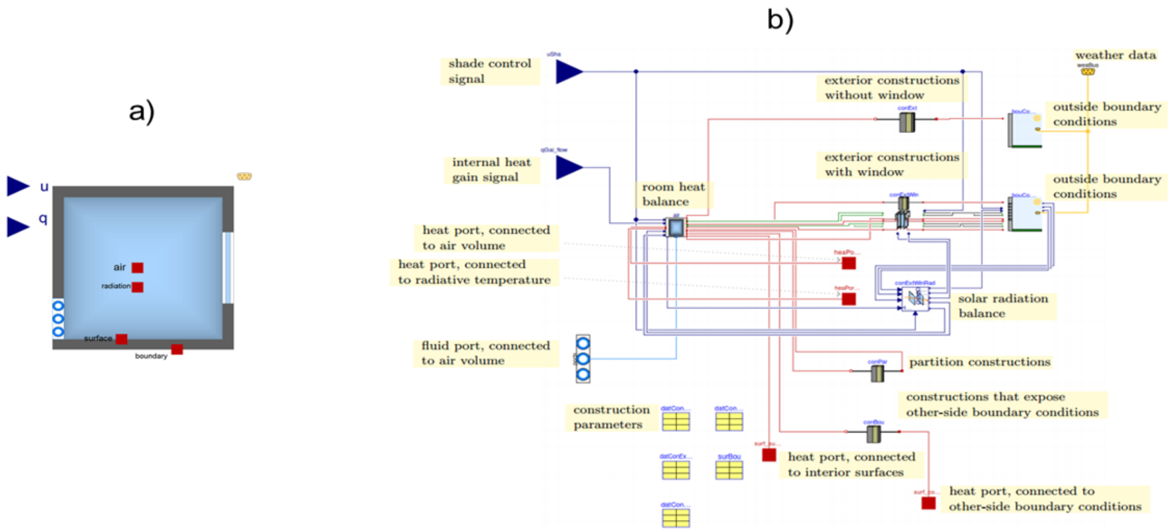

Figure 5.3 – MixedAir model from Buildings library: (a) – Icon layer view. (b) – Diagram layer view. Source: [70]

As shown in figure 5.3(b), *MixedAir* model calculates heat balance, depending on conditions like outside air temperature, solar radiation, as well as user occupancy, and the use of lights or other equipment contributing to internal heat gains. Therefore, the temporal resolution of the model's output depends on the coarsest resolution of these boundary condition inputs. In this project, these consist on hourly data, thus simulations were carried out for an entire year, with an hourly output interval. Each neighborhood building is simulated individually.

The climate reference year data used in simulations is based on 20 years of historical data measured in Brussels, Belgium. It consists on an hourly data file (Brussels_064510_IWEC.epw) obtained from EnergyPlus website [73], in which it is possible to find a database of climatic data files, in various formats and organized by location.

Finally, the next sub-chapter discusses occupancy behavior and other related aspects, and how these are taken into consideration in this project.

5.1.1. Occupant's behavior and other inputs

Occupant's behavior is defined as the presence of occupants in the buildings, and the actions they take that influence the indoor environment. Regarding this topic, literature collected allows to conclude two major points. Firstly, it can largely influence the thermal load of a building, and secondly, it is very hard to predict and properly account for. The most common method to represent the occupant's behavior is by using predefined profiles. This was the method adopted in this project, and implementation was made mainly using the Modelica Standard library 3.2.1. Detailed tables of the different profiles/schedules can be consulted in appendix A.

Regarding residential buildings, the number of occupants can be one, two or four, according to the building's typology. Furthermore, based on a similar project of another author [53], two general patterns are defined:

- (1) Occupants are at work during weekdays, and at home during evening (from 18h) to early morning (8h). This pattern accounts for 19% of the Dutch households [74]. E. g., heating is on from 18h to 24h;
- (2) People are present during the whole day. This pattern accounts for 48% of the Dutch households [74]. E. g., heating is on from 8h to 24h.

Alternatively, in office buildings people are present during weekdays, from 8h to 19h. The number of people present at each hour, is defined as the maximum, or a fraction of the maximum people allowed per square meter (see appendix A). This method is based on Dutch buildings regulation, and a limit of 0.1 people/m^2 is considered.

The heat generated by the human body and introduced into indoor environment is determined using the following expression:

$$Q_{people} = \frac{\text{skin surface area} \times \text{skin heat emission} \times \text{nr of people}}{Floor\ area} \qquad [\mathrm{W}m^{-2}] \qquad (5.1)$$

The skin heat emission greatly depends on the body metabolic rate, that varies with the level of physical activity. For thermal considerations, the metabolic heat generated is generally measured in MET (Metabolic Equivalent of Task) units. One met is correspondent to 58.2 Wm^{-2}, and is defined as the rate of energy produced per unit surface area, of an average person seated at rest. For all calculations, it is considered a constant value of 1.4 met, which approximately correspond to 80 Wm^{-2}. In addition, it is assumed an average skin surface of 1.8 m^2.

In what regards temperature setpoints, heating is activated when indoor temperature is lower than 20.5ºC, until it reaches 21ºC. In turn, cooling is supplied if air temperature in the thermal zone is above 23.5ºC, until it reaches 23ºC. During night time (24h – 8h), the heating set-point is reduced to 16ºC. However, only office and semi-detached buildings are climatized during night.

Internal gains resulting from lightning were assumed to have a constant value of 10Wm^{-2}. Moreover, other activities such as cooking or TV/computer use, are represented using three levels of constant dissipation rate values, that vary throughout the day. This was made in resemblance to this [53] similar project, and the gain levels considered are low gains (2 Wm^{-2}), average gains (4 Wm^{-2}) and high gains (6 Wm^{-2}).

Finally, concerning external blinds, they are essentially used to reduce overheating problems. However, the blinds should also allow solar gains to be stored in the building to reduce the heating energy demand, during cold days. For all the district case study buildings, the following blinds control rules are defined:

- Raise blinds, if solar irradiation and outside temperature are less than 300 Wm^{-2} and 22ºC, respectively;
- Lower Blinds, if solar irradiation and outside temperature are higher than 300 Wm^{-2} and 23ºC, respectively;

5.2. Heat and Cold Generation

This sub-chapter discusses the methods utilized to determine the parameters related to heating and cooling equipment. It focuses on the strategies, assumptions and equations implemented on MATLAB. The relevant matlab code regarding heat and cold generation, can be consulted in Appendix B.

5.2.1. Heating System

All calculations regarding heat generation are made by *Space_heating.m* script. In order to distinguish between reference case, system 1 and system 2, three scripts were created. The code implemented in each file is essentially the same, however for systems 1 and 2, short-term storage had to be accounted for. The global methodology followed to create the referred script files is described in this sub-chapter. The technique used to implement thermal storage however, is later described in chapter 5.3.

The approach taken consists on assuming that at each hour, the required thermal demand is provided using water radiators, supplied by a water-to-water heat pump-aquifer combination (HP-ATES). The implemented code can be divided in three main parts. To begin with, data is inputted and nominal system conditions are declared. Then, the required supply and return temperatures for a given water mass flow and heat demand, are computed. After this, thermodynamic calculations regarding a vapor compression cycle based heat pump are made, and results are determined (COP, primary energy use, etc.).

Part 1 – Inputs and Nominal conditions

Space_heating.m script starts by reading external inputs. These are the heating demand calculated in Dymola/Modelica and the weather data obtained from EnergyPlus website, both converted to text files. In addition, seasonal thermal storage temperatures (T_{ATES}) and electricity prices are also inputs. After this, nominal and design conditions are defined. The system is designed for the coldest hour of the year, in which peak demand occurs. During this hour, heating will be running on nominal conditions, that are summarized in the following table.

Table 5.1 – Nominal conditions defined for the design hour.

Nominal Conditions	
Radiator inlet/supply temperature	$Tin_{nominal} = 50$ ℃
Radiator exhaust/return temperature	$Tout_{nominal} = 30$ ℃
Environment temperature	$Tair_{nominal} = 20.5$ ℃
Difference between mean radiator and environment temperature	$\Delta T_{nominal} = 19.5$ ℃
Nominal radiator power	$Q_{nominal} = Peak\ demand\ [W]$

The nominal water mass flow rate, $m_{water\_nominal}$, is also be determined by using the following equation:

$$m_{water\_nominal} = \frac{Peak\ demand}{c_{p-water} \cdot (Tin_{nominal} - Tout_{nominal})} \quad \text{[Kg/s]} \qquad (5.1)$$

Where, $c_{p-water}$ is the water's specific heat capacity [J·kg^{-1}·ºC^{-1}]. In all calculations is assumed that, when heating is required the water mass flow rate is constant, and equal to its nominal value. In turn, during periods with no demand, it is defined as zero. The radiator's heat output is regulated by varying the inlet and outlet water temperatures.

Part 2 – Radiator´s inlet and outlet temperatures

Based on these [75], [76] guidelines, a method to determine inlet and outlet radiator temperatures was implemented. It is based on the fact that, for each hour (h), the heat output of a radiator (Q) can be described by a relationship (characteristic curve) that depends on the temperature difference between the radiator and the surrounding environment (ΔT):

$$\mathrm{Q(h)} = Q_{nominal}\left[\left(\frac{\Delta T(h)}{\Delta T_{nominal}}\right)^{1.3}\right] \quad [W] \qquad (5.2)$$

Rearranging equation 5.2 gives:

$$\Delta T(h) = \left[\left(\frac{Q(h)}{Q_{nominal}}\right)^{\frac{1}{1.3}}\right] \cdot \Delta T_{nominal} \quad [℃] \qquad (5.3)$$

With equation 5.3, the temperature difference (ΔT) that must be imposed between the radiator and the surrounding air, in order to supply a required space heating demand (Q), can be calculated. Using this result, the radiator's mean temperature may be approximated by:

$$RMT(h) = \Delta T(h) + Tair_{nominal} \quad [℃] \qquad (5.4)$$

In resemblance to equation 5.1, the water temperature drop across the radiator can be determined as:

$$[T_{in}(\mathrm{h}) - T_{out}(h)] = \frac{Q(h)}{m_{water} \cdot c_{p-water}} \quad [℃] \qquad (5.5)$$

Based on the results from equation 5.5, the hourly inlet and outlet radiator temperatures, for a given inputted heat demand are calculated. By summing half of the difference between T_{in} and T_{out}, to the average radiator mean temperature (RMT), the supply temperature is obtained. In turn, the exhaust temperature is implemented as the radiator's mean temperature minus half of that same difference.

$$T_{in}(\mathrm{h}) = RMT(h) + \frac{[T_{in}(\mathrm{h}) - T_{out}(h)]}{2} \quad [℃] \qquad (5.6)$$

$$T_{out}(\mathrm{h}) = RMT(h) - \frac{[T_{in}(\mathrm{h}) - T_{out}(h)]}{2} \quad [℃] \qquad (5.7)$$

To illustrate, the next figure provides a plot with the computed water inlet/outlet temperatures, for the semi-detached (S1) building typology, during the first two days of the simulated year. It can be seen that inlet/outlet temperatures greatly vary during this period, in accordance with outside air temperature and heat demand.

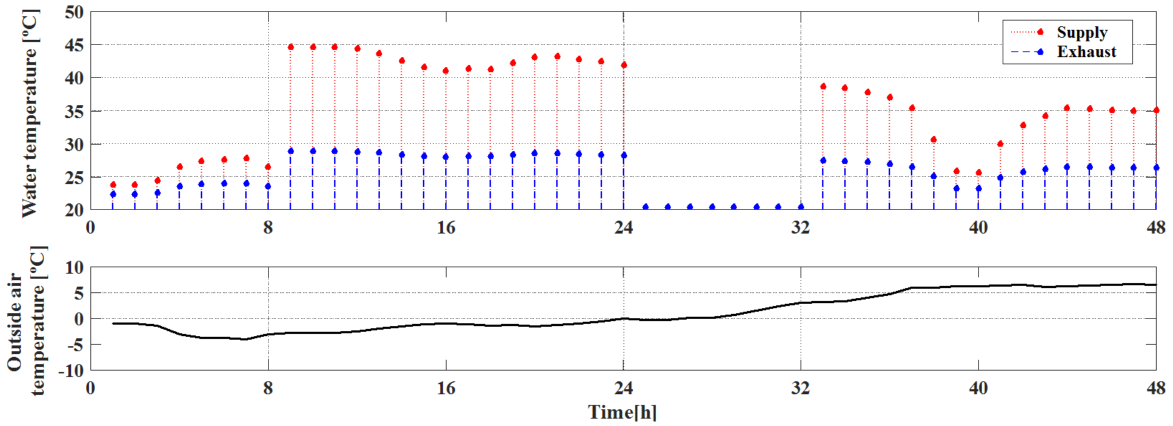

Figure 5.4- Top: Calculated supply (red) and return (blue) water temperatures for the first 48h of the simulated year, using S1 building typology demand data; Bottom: Outside air temperature for the first 48h.

Part 3 – Vapor compression cycle based water-to-water heat pump

The vapor compression cycle based heat pump is mainly composed of four units, i.e. the condenser, the compressor, the evaporator and the expansion valve (fig. 5.5). In the evaporator, heat removed from the water of the ATES starts by evaporating the heat pump's working fluid at low temperature and pressure (1-2). After this, the compressor increases pressure, at the expense of electricity consumption (2-3). In the condenser, the working fluid then condenses at high temperature and pressure, providing useful heat to the circulating colder water used to supply heat (3-4). Finally, the condensed working fluid is expanded back to the evaporator (4-1), and the cycle is repeated.

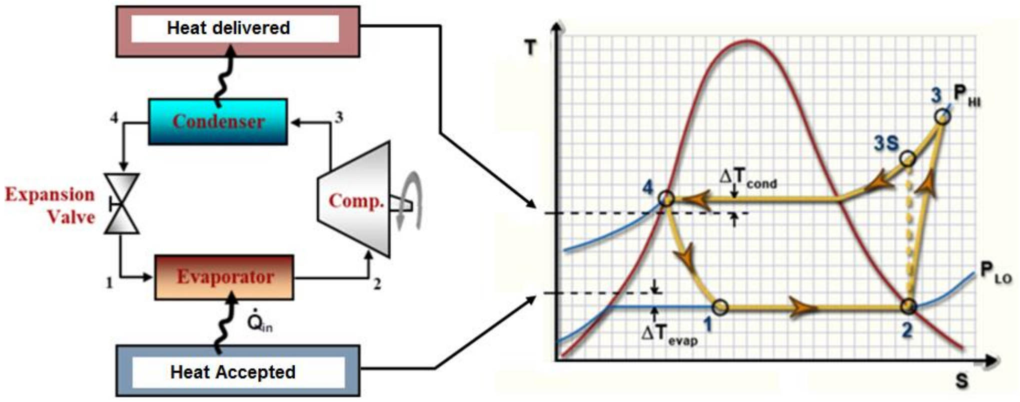

Figure 5.5 – Vapor Compression cycle (left): Schematic representation; (right): Temperature – Entropy diagram.

The methodology implemented to analytically estimate the COP and primary energy use, is based on the fact that these can be calculated, if the thermophysical properties of the working fluid as well as the evaporator and condenser temperatures (T_{evap} and T_{cond}), are known.

According to the engineering practice, the temperature difference between the refrigerant and the heat transfer medium water in the heat exchanger, (i.e. condenser and evaporator) is about 4-8ºC [60]. However, in order to assure good heat transfer rates, a temperature difference of 10ºC is adopted in this project. In short, the condenser and evaporator temperatures, at each hour are defined as:

$$T_{cond}(h) = T_{in}(\mathrm{h}) + 10 \qquad [°\mathrm{C}] \qquad (5.8)$$

$$T_{evap}(h) = T_{ATES}(\mathrm{h}) - 10 \qquad [°\mathrm{C}] \qquad (5.9)$$

The term T_{in} is the radiator inlet temperature, that is previously calculated using equation 5.6. In turn, T_{ATES} is the temperature of the (warm) water stored in the aquifer thermal energy storage (ATES). This temperature is one the external inputs given to matlab, and is later described.

At this point is useful to refer the assumptions made:

- The working fluid considered is R134a.
- Based on the numeric notation adopted in fig. 5.5, R134a stream fluid (1) is on vapor-liquid equilibrium phase, stream (2) is saturated vapor, streams (3) and (3s) are superheated vapor and flux (4) is saturated liquid.
- There are no pressure drops throughout the evaporator or condenser (isobaric);
- The compressor operates adiabatically with a constant isentropic efficiency ($\eta_{S,\ Comp}$) of 0.8;
- The expansion through the valve is an isenthalpic process;

It is conceivable to say that the heat pump must keep a heat output (Q_{HP}) at least equal to the required thermal demand, which is dependent on the calculated radiator water inlet/outlet temperature difference and mass flow rate, as follows:

$$Q_{HP} = m_{water} \cdot c_{p-water} \cdot (T_{in} - T_{out}) \qquad [\mathrm{W}] \qquad (5.10)$$

Where, $c_{p-water}$ is the water's specific heat capacity [J·kg^{-1}·ºC^{-1}]. However, as it was said before, it is in the condenser that the heat-pump's working fluid delivers the generated heat (3-4). Therefore, for steady-state calculations, and considering a single-inlet, single-outlet condenser, the heat output (Q_{HP}) can be also described by the equation 5.11 [77]. In this case, m_{R134a} is the mass flow rate of the working fluid; H_3 and H_4 are respectively, the fluid´s initial and final enthalpy [J/Kg], during the condenser phase.

$$Q_{HP} = -Q_{34} = m_{R134a} \cdot (H_3 - H_4) \qquad [\mathrm{W}] \qquad (5.11)$$

Expression 5.11 can be solved for the working fluid's mass flow rate:

$$m_{R134a} = \frac{Q_{HP}}{(H_3 - H_4)} \qquad [\mathrm{Kg/s}] \qquad (5.12)$$

Where, H_3 (superheated-vapor flux enthalpy) can be written as:

$$H_3 = H_2 \frac{(H_{3s} - H_2)}{\eta_{S,\ Comp}} \qquad [\mathrm{J/Kg}] \qquad (5.13)$$

Regarding H_4 (saturated-liquid enthalpy), its value can be found in a thermophysical proprieties table or database for the R134a fluid, by assuming that fluid stream (4) is saturated liquid at the condenser temperature (T_{cond}). H_2 and H_{3s} are determined in a similar way. The method consists on taking into consideration the vapor-compression cycle characteristics, the condenser and evaporator temperatures, as well as the r134a properties. For this to be possible, all the required data concerning r134a, were obtained from the NIST webbook [78], and inputted to MATLAB as functions, by using polinomial or linear regression techniques. The r134a propriety curves implemented are:

- Temperature vs. Enthalphy, for liquid saturated;
- Temperature vs. Entropy, for vapor saturated;
- Enthalpy vs. Temperature, for vapor saturated;
- Saturation Pressure vs. Temperature, for vapor saturated;
- Isobaric Entropy vs. Temperature curves, from 0.5 MPa to 3.5 MPa for vapor saturated;
- Isobaric Entropy vs. Enthaly curves, from 0.5 MPa to 3.5 MPa for vapor-liquid.

The implementation of these curves can be consulted in detail, by looking the respective matlab code provided in Appendix B. In essence, these were implemented using “for” and “if/else” matlab statements.

After all the required properties are determined, for all the stages of the vapor-compression cycle, results from equations 5.12 and 5.13 can be obtained, and they may be used to determine compressor work (W_{Comp}) and COP, by using the next equations.

$$W_{Comp} = m_{R134a} \cdot (H_3 - H_2) \qquad [\mathrm{W}] \qquad (5.14)$$

$$COP_{HP} = \frac{Q_{HP}}{W_{Comp}} \qquad [-] \qquad (5.15)$$

$$Q_{HP} = Q_{ATES} + W_{Comp} \qquad [\mathrm{W}] \qquad (5.16)$$

With the compressor work and COP, all other relevant results can be obtained, as for instance the amount of energy removed/injected in ATES (Q_{ATES}) or the electricity consumption. It was considered that the heat exchange between HP’s working fluid and radiator supply water occurs with 95% effectiveness, and that the electric conversion efficiency of the compressor is 92%.

Finally, in order to represent the effect of partial load running, previously discussed in chapter 2, it was assumed that the minimum compressor work has always to be, equal or superior to one third of its maximum (nominal value). Therefore, during low demand periods, the compressor work determined is sometimes higher than the actual heat output, which dramatically reduces COP if short-term storage is not applied.

5.2.2. Cooling system

Cooling related calculations are computed in *Space_Cooling.m* script. Its main purpose is to determine how much heat must the ATES system water dissipate, in order to supply the cooling load. The external inputs are the cooling demand obtained from Dymola/Modelica, the outside air temperature and relative humidity from the weather file, as well as the ATES water temperature ($T_{ATES\text{-}Cold}$).

The method applied consisted on implementing cooling as a fresh air supply with heat recovery. The outside air is chilled by using free cooling, based on the cold water stored in the aquifer thermal energy storage (ATES) system, and then introduced into indoor environment. A schematic representation is provided in the next figure.

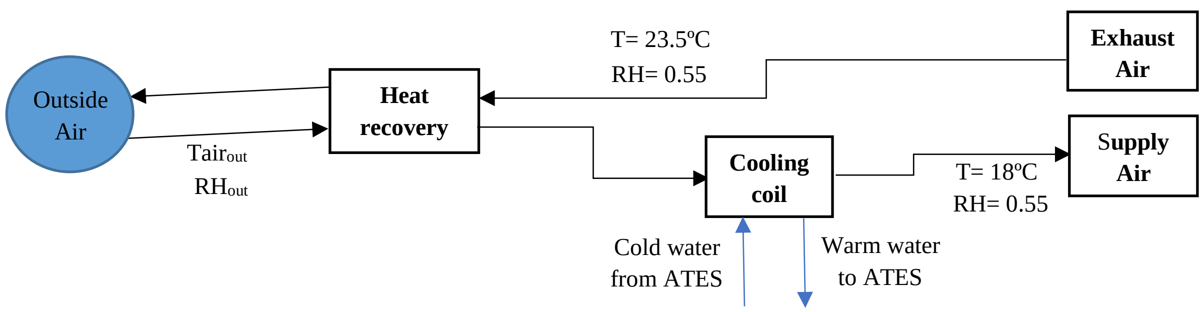

Figure 5.6 –Schematic representation of the cooling process implemented on *Space_cooling.m* script.

In order to meet the required cooling demand, it is defined that a fresh air supply at 18ºC and 55% relative humidity, is kept. Moreover, exhaust air conditions are also considered to be constant. A temperature of 23.5 ºC and a relative humidity of 55% is assumed.

Using a simple heat balance equation, the air mass flow rate (m_{air}) that must be induced in order to remove a specified heat load (Cooling demand) may be approximated by taking into consideration the known supply and exhaust air conditions:

$$m_{air} = \frac{\text{Cooling Demand}}{\left(H_{exhaust} - H_{supply}\right)} \qquad [\text{Kg/s}] \qquad (5.17)$$

Where, $H_{exhaust}$ and H_{supply} are respectively, the enthalpies [J/Kg] of the exhaust and supply air, at the defined temperature and humidity conditions, and cooling demand [W] is the thermal load data obtained using Modelica/Dymola.

For the purpose of determining the enthalpy of the air, based on its known conditions, secondary matlab functions are implemented. For instance, $H_{exhaust}$ and H_{supply} are determined using the matlab function *Enthalpy.m*, called in *space_cooling.m* script. This function is programmed to calculate absolute humidity (w) and enthalpy (H) of the air, based on the inputted dry bulb temperature (T_{db}) and relative humidity (RH):

```
function [w, h] = Enthalpy (Tdb, RH)                (5.18)
```

Additionally, using another function, the relative humidity (RH) can be determined, based on the dry bulb temperature (T_{db}) and absolute humidity (w). These functions were created by implementing air proprieties data curves obtained from Shapiro, 6th edition [79]. The matlab code relative to this is provided in appendix B (secondary functions). With these functions, all relevant physical properties of the air, throughout the cooling process are obtained.

Therefore, after determining the air mass flow rate using equation 5.17, heat recovering is addressed. During this process, the warm exhaust air removed from indoor environment, is used to slightly cool the outdoor hot air used as supply air. Taking into account that exhaust air's temperature is at 23.5ºC, it is assumed that the cooling surface in the heat exchanger unit (used to heat recovery), is always above or equal to the dew point temperature. Thus, air is cooled without any change in its specific humidity. The recovered heat (Q_{HR}) is then calculated, by assuming a counter-flow heat exchange process, with constant effectiveness (E=0.7):

$$Q_{HR} = \mathrm{E} \cdot m_{air} \cdot c_{p-dry_{air}} \cdot \left(T_{out} - T_{exhaust}\right) \qquad [\text{W}] \qquad (5.19)$$

Where, $c_{p-dry_{air}}$ is the dry air's specific heat capacity [$\text{J}\cdot\text{Kg}^{-1}\cdot{}^{\circ}\text{C}$], T_{out} is the outside ambient air temperature [ºC] and $T_{exhaust}$ is the exhaust air temperature.

At this point, by knowing the amount of heat transferred during heat recovery (Q_{HR}), and assuming that the specific humidity does not change, the air's relative humidity, enthalpy and temperature after this process can be determined, using the already referred matlab functions.

Once this is made, the remaining heat that must be removed from the air, in order to assure its supply conditions, may also be computed (eq. 5.19). This amount of energy is dissipated using the cold water from the ATES.

$$Q_{coil} = m_{air} \cdot \left(H_{after\_recovery} - H_{supply}\right) \qquad [\text{W}] \qquad (5.20)$$

The ATES water mass flow is also calculated in this script, with the purpose of being later used to determine pressure losses, and respective pumping requirements. For this, it was assumed that water flow is regulated, based on keeping its leaving temperature as 16ºC (eq. 5.21).

$$m_{water-ATES} = \frac{Q_{coil}}{(16 - T_{ATES})} \quad [\text{Kg/s}] \qquad (5.21)$$

Finally, in order to exemplify some the outputs that can be obtained through *space_cooling.m* the next figure is presented. It provides air supply and ATES water mass flow rates (a), the temperature of the air during the various steps of the cooling process (b), and finally the heat removed by the ATES water (c), for the detached (D2) building typology, in a typical Summer day (July).

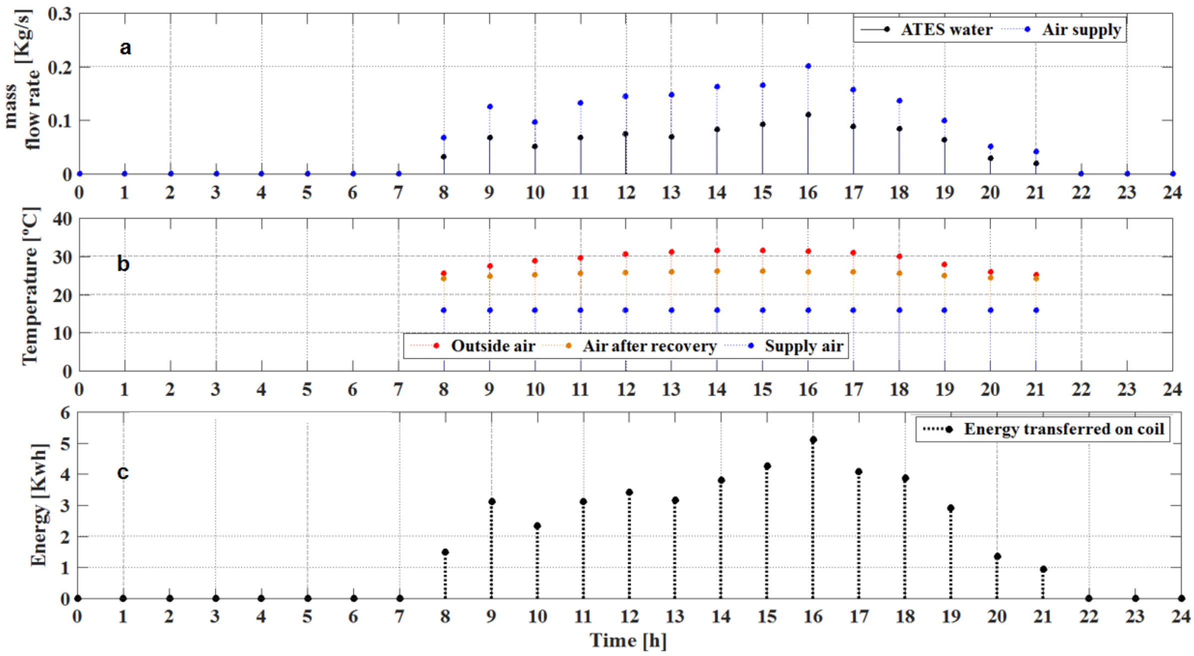

Figure 5.7 – (a): Air supply (blue) and ATES water (black) mass flow rate; (b) Temperature of the air during the cooling process; (c) Energy transferred from air to water (ATES) on the cooling coil. For a typical summer day, building typology D2

5.3. Thermal Energy Storage

In this project, both seasonal and short-term thermal storage are considered. The seasonal storage consists on an Aquifer thermal energy (ATES) system, while short-term TES is assumed to be an insulated water tank.

In what regards the ATES, it would be very difficult to properly represent it using MATLAB since its performance depends on a large number of uncertainties, such as site historical data. Instead, the hourly approximated temperatures of the ATES system installed in Technical University of Eindhoven, for an entire year, were provided by a local researcher. These are externally inputted to matlab as text files (STES_cold.txt & STES_hot.txt).

Concerning short-term storage (water tank), literature shows that a heat balance analysis can be made considering water thermal stratification (thermal buoyancy), through the use of discretization methods [23]. Nonetheless, other simpler approaches are also applied, assuming fully a mixed water tank [80], [81]. This is the technique adopted to implement short-term storage in this study.

The implementation can be divided in four parts:

(1) Water tank sizing and design conditions;
(2) Storage loss factor [W/K] determination;
(3) Control rules;
(4) Heat balance calculation.

1 – Water tank sizing and design conditions

In order to obtain the required water storage mass, or volume, based on the inputted heat demand, a method that could be applied to all building typologies was established. The design conditions assumed are listed in the next table.

Table 5.2 – Storage related design conditions.

Design Conditions	
Maximum storage temperature	$Tst_{max} = 60$ ℃
Minimum storage temperature	$Tst_{min} = 27$ ℃
Partial storage factor	$f_{partial} = 0.5$
Maximum cumulative 24h demand	$Q_{24-nominal}$ [J]

Using matlab's "for" and "if/else" statements, the cumulative heating demand for the following 24h is calculated, for each hour. The nominal energy capacity of the storage ($E_{storage-nominal}$) is then defined as half of the maximum 24 h consecutive heat requirements of the year. This means that a partial load storage strategy is being followed.

$$E_{storage-nominal} = f_{partial} \cdot Q_{24\_nominal} \quad \text{[J]} \qquad (5.22)$$

After the design energy capacity ($E_{storage-nominal}$) is computed, the required water mass to store that energy ($m_{water-storage}$) can be determined, based on the design conditions:

$$m_{water-storage} = \frac{E_{storage-nominal}}{c_{p-water} \cdot (Tst_{max} - Tst_{min})} \quad \text{[kg]} \qquad (5.23)$$

Where $c_{p-water}$ is the water thermal heat capacity [$J \cdot kg^{-1} \cdot {}^{o}C^{-1}$]. Finally, the equivalent volume is given by next equation, where ρ_{water} is the water density:

$$V_{water-storage} = \frac{m_{water-storage}}{\rho_{water}} \quad [m^3] \qquad (5.24)$$

2 – Storage loss factor

The storage heat loss factor ($U_{storage}$) is calculated using the following expression:

$$U_{storage} = U_{insulation} \cdot A_{sup-storage} \quad [\mathrm{WK}^{-1}] \qquad (5.25)$$

Where, $U_{insulation}$ is the heat transfer coefficient [W/m^{2}K] and $A_{sup-storage}$ is the tank surface area. Given the already determined volume of the water tank ($V_{water-storage}$), the surface area ($A_{sup-storage}$) is computed for each building typology, by assuming a cylindric shape. In what concerns insulation, the R-value (thermal resistance) implemented is the R15 value on the ASHRAE standardized insulation rating. After converting it to European units (SI), the equivalent heat transfer coefficient is obtained, and can be used to determine the overall storage loss factor.

$$U_{insulation} \approx 0{,}38 \ \frac{W}{m^2K}$$

3 – Control rules

While the control strategy applied in reference case system consists on supplying energy as it is required, in systems 1 and 2, short-term storage is present, and in this case energy supply is designed to operate based on electricity prices. It is considered that a night electricity tariff (off-peak) is applied from 22h to 8h, and that during the remaining hours a more expensive peak tariff is charged (peak-tariff). The hourly electricity tariffs [€/kWh] are created based on historical electricity day-ahead auction prices, negotiated on the Dutch EPEX SPOT market in 2016 [82]. In order to create indexed tariffs, connection and transmission services tariffs [83] are summed to the hourly prices obtained from EPEX spot market. Further information on the electricity tariffs applied, can be found on appendix C.

It is assumed that the heating load for the next day is known on the previous day. Therefore, for each hour, the heat load of the next 24h (Q_{24h}) can be computed. In addition, heat requirements for the next 2h (Q_{2h}) are also determined.

The storage control procedure is implemented mainly using matlab's "for" and "if/else" statements. Firstly, for each hour, the code starts by checking if the energy stored in the tank is below the tank's nominal capacity. If this is true, further instructions are read, otherwise no heat is produced in that hour. After this, the indexed electricity tariff is read, and based on which tariff applies, peak or off-peak, a different method is applied. The system is designed to use the night tariff, as much as possible. This is done by implementing the storage control rules described in the next figure.

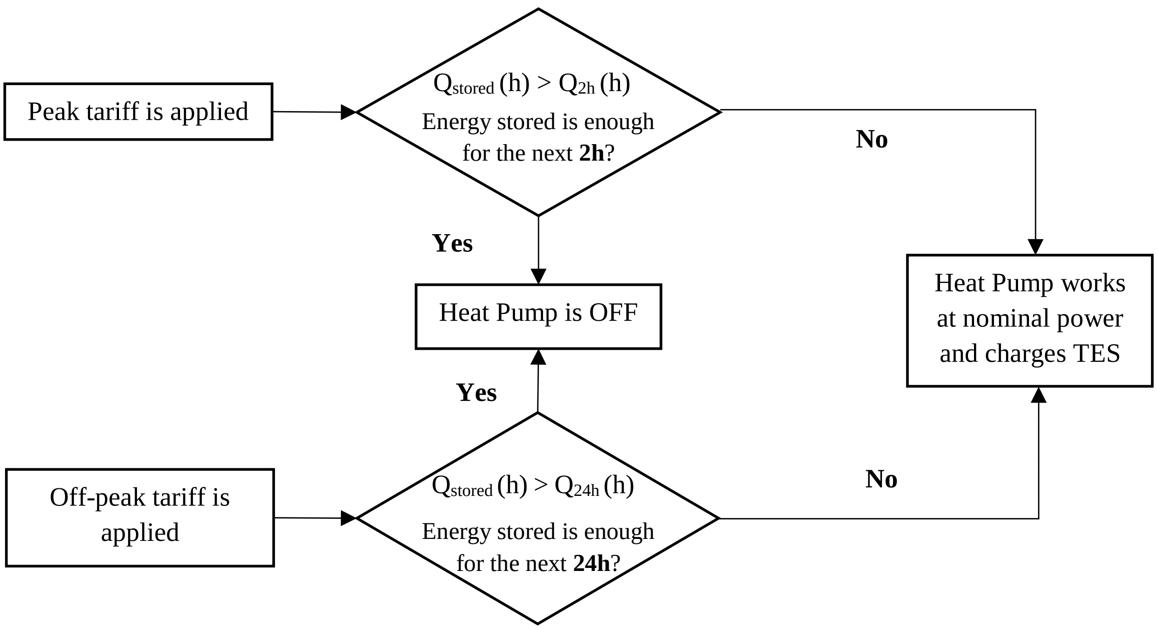

Figure 5.8 - Short-term storage control strategy diagram.

4 – Storage heat balance

The main objective of this part is to compute the water tank heat balance, in order to determine the amount of energy stored at each hour, as well as storage temperature and heat losses. The amount of energy stored $(E_{storage})$, at each hour (h) is calculated as follows:

$$E_{storage}(\mathrm{h}) = E_{storage}(\mathrm{h}-1) + E_{injected}(\mathrm{h}-1) - E_{extracted}(\mathrm{h}-1) - E_{loss}(\mathrm{h}-1) \qquad [\mathrm{J}] \qquad (5.26)$$

Where $E_{storage}(\mathrm{h}-1)$ is the amount of energy that remains available from last hour, $E_{injected}$ is the energy produced by the heat pump and transferred to the storage, $E_{extracted}$ is the energy removed from the water tank in order to supply heat requirements and finally, E_{loss} are the thermal energy losses.

Additionally, for each hour (h), the storage water temperature $(T_{water-storage})$ is calculated by using expression 5.27, where $m_{water-storage}$ is the storage water mass [Kg], and $c_{p-water}$ is the water's thermal heat capacity [J·kg^{-1}·ºC^{-1}].

$$T_{water-storage}(\mathrm{h}) = \left[\frac{E_{storage}(\mathrm{h}) - E_{storage}(\mathrm{h}-1)}{m_{water-storage} \cdot c_{p-water}}\right] + T_{water-storage}(\mathrm{h}-1) \qquad [℃] \qquad (5.27)$$

The most relevant initial conditions implemented are listed in the following table.

Table 5.3 – Main initial conditions implemented in MATLAB, regarding short-term storage heat balance calculation.

Initial Conditions	
Energy stored	$E_{storage}(1) = 0$
Storage temperature	$T_{water-storage}(1) = 27$ ℃
Energy transferred to storage	$E_{injected}$ $(1) =$ Nominal
Energy removed from storage	$E_{extracted}(1) = Demand\ (1)$
Energy Losses	$E_{loss}(1) = Nominal$

It is assumed that, for the first iteration in matlab, the storage is completely discharged, and is charged at the maximum transfer rate (heat pump's nominal heat output).

In order to simulate the charging process, a proper heat pump's output temperature must be calculated. The temperature should not be permanently the maximum, as it would greatly degrade the COP value, however it must be enough to promote the required heat transfer between the tank's water, and the water from the heat pump. For the charging process, a temperature difference of 15ºC is defined, until the maximum supply water temperature of 60ºC is reached. If this occurs, the energy output is then regulated by varying the water mass flow rate.

$$T_{charging}(\mathrm{h}) = T_{water-storage}\ (h) + 15\ ℃ \qquad [℃] \qquad (5.28)$$

Finally, concerning heat losses, it was assumed a constant heat transfer efficiency of 97% between the storage and the heat pump fluids (charging/discharging process), and a 95% efficiency when transferring energy from the water tank to the heating system supply water (radiators). The tank's surrounding air temperature, used to compute energy storage losses, is assumed to vary through the year and is comprised between 16 ºC and 24ºC, system 1. In system 2, the water tank is assumed to be buried, and the underground soil temperature is used (later described).

5.4. Heat and Cold Distribution

The methodology to estimate district distribution pressure and thermal losses using MATLAB, is discussed in this chapter. It is considered that two sets of buried pipes for supply and return exist, one set for cold water and another for hot water. The functions implemented allow to calculate hydraulic and thermal parameters, for a given single pipe section, essentially based on length, depth, water volumetric flow rate and temperature.

5.4.1. Hydraulic Considerations

Pressure losses in water distribution, and the respective pumping electricity consumption are estimated using *Pipe_Hidraulic_Power.m* function. The script starts by reading inputted hourly mass flow rate values (m_{water}), that are given by another matlab function, and then determines the correspondent hourly volumetric flow rate (V) [m^3/s]. Based on these values, the nominal/design condition are defined.

First, the annual maximum volumetric flow rate ($V_{flow\text{-}nominal}$) is determined. With this result, and assuming a maximum water flow velocity of 2.5 m/s ($v_{nominal}$), the pipe diameter (D) is computed:

$$D = \sqrt{\frac{4\,V_{flow-nominal}}{\pi\,v_{nominal}}} \qquad [m] \qquad (5.29)$$

After defining the design conditions, the water velocity (v) for each hour, according to the calculated diameter, can be determined by rearranging expression 5.29.

$$v = \frac{4\ V_{flow}}{\pi\,D^2} \qquad [ms^{-1}] \qquad (5.30)$$

In a piping system, pressure losses occur due to changes in elevation, fluid friction, or presence of equipment (e. g. valves, meters, heat exchangers, etc.). In what concerns the first, pressure losses due to elevation ($\Delta p_{vertical}$) are calculated as follows:

$$\Delta p_{vertical} = \rho_{water}\ g\ \Delta h \qquad [Pa] \qquad (5.31)$$

Where, ρ_{water} is the water density [Kg/m^3], g the acceleration of gravity [m/s^2] and Δh, the vertical elevation [m]. Regarding pressure drop resulting from fluid friction calculations ($\Delta P_{friction}$), ASHRAE fundamentals handbook [37] and Incropera [84] suggests the use of Darcy-Weisabch equation:

$$\Delta p_{friction} = f\left(\frac{L}{D}\right)\left(\frac{\rho_{water}\cdot v^2}{2}\right) \qquad [Pa] \qquad (5.32)$$

Where, L is the pipe length [m] (given to this function as an input), and f [-] is the friction factor, which is dependent on the Reynolds number. Therefore, for each hour, both the Reynolds number (Re) and the friction factor (f), are computed. In equation 5.33, the term ν represents the water kinematic viscosity [m^2/s], and expression 5.34 was chosen because it is valid for a wide range of Reynolds numbers.

$$\mathrm{Re} = \frac{D\,\mathrm{v}}{\nu} \qquad [-] \qquad (5.33)$$

$$\mathrm{f} = [0.79\,log(\,Re - 1.64\,)^{-2}] \qquad [-] \qquad (5.34)$$

Moreover, in what concerns pressure losses induced by equipment ($\Delta \mathrm{p}_{equipment}$), a constant approximated value was assumed. Once all the different pressure drops are known, the total pressure loss (Δp) can be obtained:

$$\Delta p = \Delta p_{vertical} + \Delta p_{friction} + \Delta p_{equipment} \qquad [\mathrm{Pa}] \qquad (5.35)$$

Finally, the hydraulic power ($P_{hidraulic}$) required in the pump, to overcome the flow resistance associated with this pressure drop (Δp) and respective electricity consumption, are computed, assuming a constant electric conversion efficiency of 70%.

$$P_{hidraulic} = \Delta p \cdot V_{flow} \qquad [\mathrm{W}] \qquad (5.36)$$

5.4.2. Thermal Considerations

Regarding heat losses in pipes, calculations are made using *Thermal_Losses_Pipe.m* function. The expression implemented to determine heat losses ($\mathrm{Q}_{loss-pipe}$) is:

$$\mathrm{Q}_{loss-pipe} = U_{pipe} \cdot A_{sup} \cdot \mathrm{LMTD} \qquad [\mathrm{W}] \qquad (5.39)$$

Where, U_{pipe} is the overall pipe heat transfer coefficient [W/m^2K], A_{sup} is the pipe´s surface area [m], that is determined using the inputted pipe length, and finally LMTD, which is the logarithmic mean temperature difference, between surrounding soil and water, for inlet and outlet conditions along the pipe’s length.

$$\mathrm{LMTD} = \frac{\Delta T_{outlet} - \Delta T_{inlet}}{ln\left(\frac{\Delta T_{outlet}}{\Delta T_{inlet}}\right)} \qquad [^{\circ}\mathrm{C}] \qquad (5.38)$$

The logarithmic nature of this average temperature difference is due to the exponential nature of heating or cooling. This method is generally used to determine temperature driving force for heat transfer in flow systems, particularly in pipes and heat exchangers. In order to be able to calculate the LMTD, as well as thermal energy losses, for each hour of the year simulated, an auxiliary equation (5.37) is used, as it is described by Incropera [84].

$$\frac{\Delta T_{outlet}}{\Delta T_{inlet}} = \frac{T_{soil} - T_{in}}{T_{soil} - T_{out}} = \exp\left(-\frac{U_{pipe}\,A_{sup}}{m_{water}\,c_{p-water}}\right) \qquad [^{\circ}\mathrm{C}] \qquad (5.37)$$

Where, T_{soil} is the surrounding soil temperature [ºC], T_{in} and T_{out} are the inlet and outlet water temperatures respectively, $c_{p-water}$ is the thermal heat capacity [$J \cdot kg^{-1} \cdot {}^{o}C^{-1}$] and finally, m_{water} is the water mass flow rate [Kg/s], that is given as input from another matlab function.

In order to determine the soil temperature at the specified depth (1.5 m was assumed), a matlab function was created (*Soil_T.m*). This function is programmed to calculate underground soil temperature, given an inputted depth, and is based on the correlation developed by Kusuda & Achenback [85]. To perform this operation, it is required to know the average soil surface temperature and amplitude, as well as the Julian day in which minimum soil surface temperature occurs (see appendix B – Soil_T for details).

Concerning the overall heat transfer coefficient (U_{pipe}), in real applications the amount of pipe insulation is based on economic and site location aspects. For this reason, it is difficult to make assumptions regarding this topic. Basically, more insulation means less thermal losses, however it also increases investment costs. In order to determine typical insulation values, for both short-term storage (chapter 5.3) and pipes, the various R-values in the ASHRAE's standardized insulation rating [5-25] were plotted against heat losses, for a set of pre-defined conditions. The choice of the insulation value, consisted on choosing the first R-value, from which an increase in insulation does not significantly reduces heat losses, under the pre-defined conditions.

Finally, for the pipe insulation, an R18 value in the ASHRAE's standardized insulation rating is implemented, which is then converted to European units (SI), and used to determine the equivalent heat transfer coefficient (U_{pipe}).

$$U_{pipe} \approx 0{,}31 \ \frac{W}{m^2K}$$

5.5. Complete system and final remarks

The various methods described in the previous chapters cover the majority of the analytical model implemented in MATLAB. Nonetheless, due to the large number of variables and methods considered in this study, it is impossible to address all the constants and assumptions. This can be done by consulting the matlab code in the appendix B.

As previously said, the MATLAB code implemented for the different configurations is practically the same, however some methods slightly change. To begin with, in reference case and system 1, it is assumed that there is a heat pump at each building, thus the water supply and return temperatures are calculated based on a radiator characteristic curve. In system 2 however, the system is implemented as a single centralized heat-pump, which provides heat to all consumers. For this reason, another method is adopted. The supply and return temperature are based on a temperature curve, that depends on outdoor air temperature varying from 35ºC to 55ºC. Additionally, as already said, in system 2 short-term storage (water tank) is considered to be buried at an average depth of 3 m.

The matlab functions described in the last chapters, are called in *District.m*. Three *m* files were created, one for each district configuration. In these files, the relevant matlab variables are obtained from other functions hierarchically below. By doing this, global results considering all building typologies and system configurations are computed.

Chapter 6 - Results and Discussion

Among other data, the hourly space heating and cooling demand values determined in Modelica/Dymola, are given as inputs to the MATLAB functions described in the previous chapter, which are combined to form a simple analytical MATLAB model. Through it, hourly, monthly and annual results, concerning the three district system configurations were obtained. The most relevant annual results, are summarized in table 6.1, that is provided at the end of this chapter.

First of all, when comparing the results from the district configurations with (systems 1 and 2), and without short-term storage (reference case), it is possible to understand that the presence of short-term storage and its respective control strategy, results in a more flattened heat output from the heat pumps. This can be visualized in the next figure, that provides the heat pump's heat output for the reference case (black dots) and system 1 (blue dots), resulting from the calculations regarding the detached building typology (D2), for the second week of January.

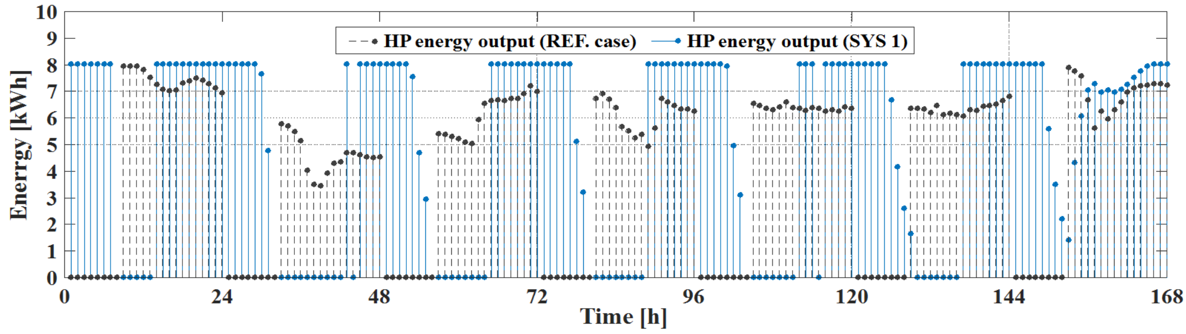

Figure 6.1 – Heat pump heat output for the reference (black) and system 1 (blue); 2nd week of January, D2 typology.

It is possible to observe that when combined with short-term TES, heat pumps operate in a more stable and continuous way, maintaining the heat output at its nominal power during long periods. This contrasts with the equivalent reference case results, in which heat pumps have to follow the hourly heat demand, that varies throughout time.

Moreover, by comparing the time at which heat production occurs, one may also notice that the heating load was successfully shifted from peak to off-peak periods (mismatch between blue and black dots in fig. 6.1), or in others words, from day to night/weekend. By doing this, heat is being produced when the global demand and cost of electricity are lower. This reduced the peak power demand of the entire district and the individual costs for the final heat consumers, by taking advantage of lower electricity tariffs at night. As illustrated in figure 6.2, heat is produced and stored during night and weekends (off-peak), and is then used throughout the day, when global demand and costs are higher.

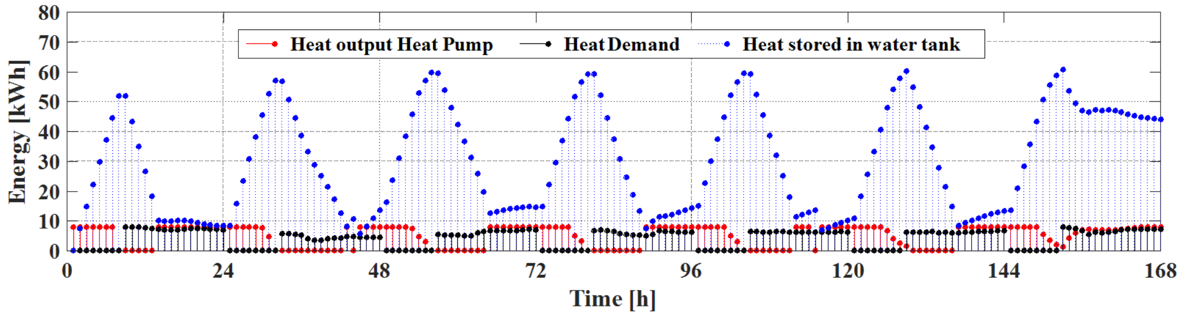

Figure 6.2 – Heat pump's heat output (red), heating demand (black) and energy stored in the short-term storage (blue). 2nd week of January, D2 typology.

The individual electricity costs for the heat consumers were analyzed for the reference case and system 1. In reference case, a simple tariff is applied, while on system 1 a double tariff (day/night) is employed (see appendix C). The results show that the annual electricity costs were 37.5 % lower in system 1, when compared to the reference case, in which 8968€ were spent for space heating purposes. Furthermore, calculations under the same electricity tariff (simple tariff) were carried out, but still electricity costs were 11.5% lower in system 1.

Furthermore, during implementation, the heat pump´s capacity is selected to match the design-day load, when the requirements for heating are maximum. Regarding this, it was found that in system 1, depending on the building typology, the presence of short-term storage allowed to reduce the heat pump’s installed capacity around 25-35%, without undermining heat supply. This supports the hypothesis that by having short-term storage, the capacity of thermal producing equipment may be selected closer to the average load, rather than peak conditions.

In general, the results obtained show that, from the energy point of view, the systems with short-term energy storage perform better than the reference case. The heat pump’s annual average coefficient of performance (COP) increased approximately 14.6% in system 1, and 11.7% in system 2, when compared to the reference case, in which an annual average COP of 3.42 was obtained.

As it can be seen in figure 6.3, this COP improvement is particular relevant throughout the milder Seasons (Spring and Autumn). During these periods, the heating demand is lower and never reaches the maximum for which heat pumps were designed. As it was already referred, if storage is not available, heat pumps have to follow the hourly heat demand, that varies throughout time. This means that, in reference case, heat pumps are often forced to run on partial load conditions during many hours, sometimes at the minimum HP’s heat output, which greatly reduces COP.

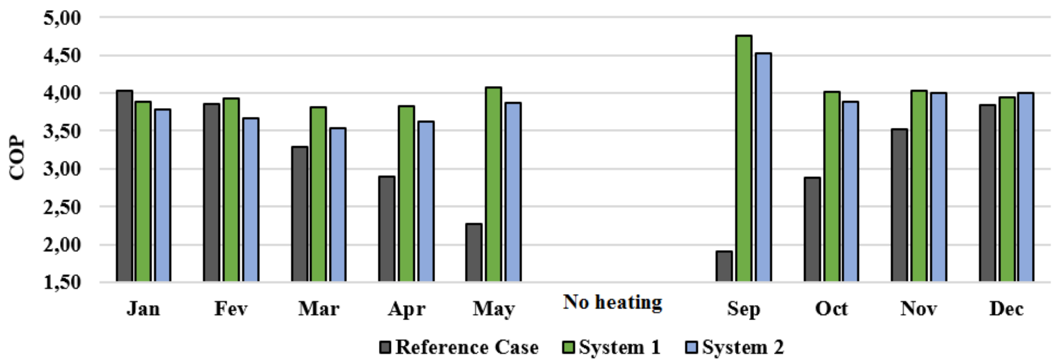

Figure 6.3 – Heat pump’s monthly averaged Coefficient of Performance, for reference case, system 1 and system 2.

In systems 1 and 2 however, short-term storage is implemented. In this case, during temperate months, heat pumps can continue operating at nominal conditions without loss of performance (COP), by storing the excess heat in the short-term storage, and using it during the following hours.

Nonetheless, the annual heat production (final energy) has increased in both, systems 1 (7 %) and 2 (7.4%), when compared to the annual heat production in the reference case (166 MWh). This essentially derives from having higher thermal losses, associated to the short-term storage and heat distribution, (see table 6.1). However, although more heat has to be produced by the heat pumps to compensate for thermal losses in systems 1 and 2, this is balanced by the higher COP values. In fact, when compared to the reference case, the electricity consumption was lower in both system 1 (-13.3%) and system 2 (-8.5%). In addition, it was noticed that electricity consumption in pumps, for water pumping purposes, is not a determining factor, at least for small and compact districts systems with short distribution pipes (300 to 400 m), as the one studied in this project.

Regarding the number of operating hours, this parameter is a measure of the heat pumps utilization efficiency. When short-term storage is available, it is beneficial to have the maximum number of operating hours possible, because this indicates that heat producing equipment is efficiently distributing the thermal demand over more hours. It is for this reason that the installed capacity can be defined based on the average load, instead of based on peak conditions. However, when globally analyzing this performance indicator, it possible to notice that the number of operating hours slightly decreases in both, system 1 (-4.4%) and system 2 (-0.9%), when compared to the 3323 hours of utilization, resulting from reference case.

This is contrary to expectations, but is related to the fact that the demand for heating in some of the buildings, presents a large discrepancy between average load and peak load. In these cases, the heat pumps could not be dimensioned based on the average load, otherwise the system would not be capable of supplying the heat requirements during the coldest hours. For this reason, the heat pumps' capacity stills higher than the heat requirements during many hours (average and below average demand periods). Since the heat pump control is implemented in order to prioritize operation at nominal conditions, the thermal output is generally superior to the required, and the excess heat at each hour is stored. This reduces energy production in the following hours, which ultimately results in lower global utilization hours. The reduction in the number of operating hours may also be related to the fact that the storage strategy implemented, prioritize heat production during nights and weekends, therefore limiting the operation hours to these periods.

From the Aquifer thermal energy storage system (ATES) point of view, all the three district configurations presented similar results (table 6.1). However, as in systems 1 and 2, the heat pump's coefficient of performance (COP) is higher, more heat is extracted from the ATES and used as source in the evaporator. Figure 6.4 provides the monthly heat exchanges with ATES, considering system 1 configuration. The negative direction represents the heat extraction, and the positive heat injected.

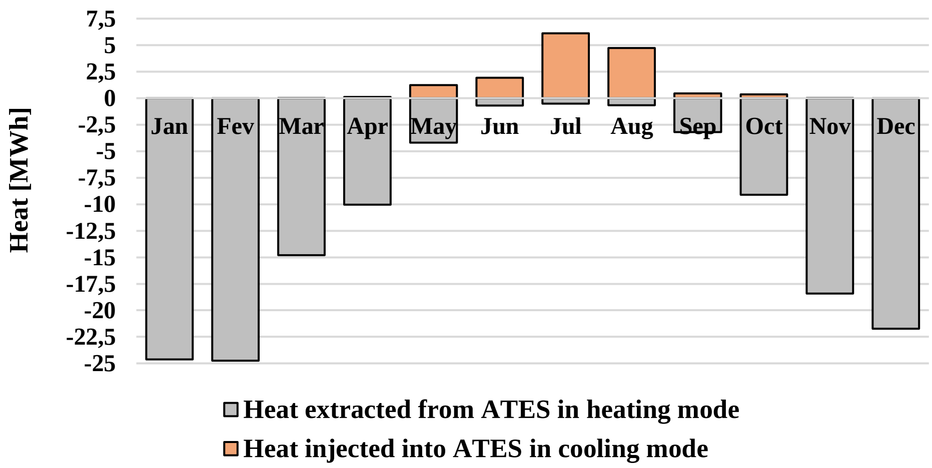

Figure 6.4 – Monthly heat exchange with ATES, system 1.

It is possible to observe that the amount of heat injected into ATES was largely inferior to the amount extracted. To be precise, the amount of heat injected is less than 13% the heat extracted. This occurs because the demand for heating is much higher than for cooling. If the seasonal thermal energy storage is unbalanced, the stored water temperature will drop, and consequently the heat pump's efficiency is reduced. In order to avoid this, other equipment is generally employed, as for instance heat exchangers may be used to transfer heat from the air to the ATES, during Summer. However, this was not considered in this project. It is likewise important to take into consideration that the diversity of the neighborhood buildings considered in this project is somewhat reduced. If, for instance, a shopping center was connected to this system, the unbalance between heat and cold could be reduced, since this type of buildings generally requires cooling during the entire year.

In what concerns the comparison between the two district configurations containing short-term storage, in a decentralized (system 1) and centralized manner (system 2), it is important to refer that for system 2 it was assumed 1% higher efficiency in all heat transfer processes, as well as 85%, instead of the 80% isentropic compressor efficiency defined for system 1. This was done in order to resemble the higher operation efficiency characteristic of centralized facilities.

Even so, the results obtained show that system 1 is slightly better from the energy performance point of view (table 6.1). Additionally, it was found that thermal losses regarding the underground water tank in system 2 were slightly lower (-2.21%) than the sum of thermal losses of all the individual water tanks, at each building. Furthermore, in system 2 the required storage capacity to shift the same amount of thermal load from peak periods to night and weekends, was 10% lower than in system 1. Basically, in system 2 the thermal losses related to distribution are higher than in system 1; however, this is compensated by the lower thermal losses of the buried water tank.

Table 6.1 – Summary of the most relevant annual results.

	Reference Case	System 1	System 2
Thermal Energy – Heating			
Annual Heat demand [MWh/year]	158.05	158.05	158.05
Annual Heat production [MWh/year]	166.02	177.62	178.24
Total thermal losses [MWh/year]	7.97	19.57	20.19
Annual average COP [-]	3.42	3.92	3.82
Heat extracted from ATES [MWh/year]	115.49	132.26	132.73
Total compressor Work [MWh/year]	50.52	45.36	46.56
Number of operating hours [h]	3323	3176	3294
Thermal Energy - Cooling			
Annual cooling load [MWh/year]	13.49	13.49	13.49
Heat injected in ATES [MWh/year]	16.94	16.94	16.94
Electricity Consumption			
Electricity consumption [MWh/year]	59.73	51.74	54.60
Water pumping parcel [MWh/year]	2.07	2.62	4.27
Electricity costs [€]	8968	5603	-
Short-term storage			
Total storage capacity [MWh]	0	1	0.9
Thermal losses Storage [MWh/year]	-	9.73	7.52

To conclude, the introduction of short-term energy storage into a District Energy containing seasonal thermal energy storage and heat pumps, proved to be beneficial. The results show that this led to a reduction in the electricity consumption, resulting essentially from the increase of the heat pumps' coefficient of performance (COP), as well as a significant reduction in the individual electricity costs for the heat consumers. On the other hand, when analyzing the results from the centralized operation (system 2), no significative differences were found, when comparing to the decentralized approach. This is probably due to the fact that the implemented MATLAB model has low resolution and is based on simple equations and methods, thus not taking into consideration a number of complex, but relevant aspects that differ in centralized operation. Therefore, the comparison between systems 1 and 2 is somewhat inconclusive.

Chapter 7 - Conclusions and Future Work

The main objective of this project was to study the influence of the introduction of short-term thermal storage in a district energy, where heat and cold requirements are supplied by a combination of seasonal TES and heat pumps. In order to do this, the first step was to do a bibliographic review, focusing on thermal energy storage and all the three main components of District Energy, which are the thermal energy production units, the distribution system and finally, the energy consumers. The most relevant information was compiled and is presented in the first part of this report.

After this, using space heating and cooling demand data determined in Dymola/Modelica, the energy performance of three district configurations was evaluated, through an analytical MATLAB model. The three configurations consist on the reference case, representing a conventional system without short-term storage, as well as systems 1 and 2 containing short-term storage in a distributed and centralized manner, respectively. Firstly, a global comparison between the systems with and without storage is made.

The results obtained show that the introduction of short-term storage using a partial demand limiting strategy, allowed to shift some of the heat load from peak to off-peak periods. This operation led to a significant reduction in the individual electricity costs for the heat consumers (11.5% to 37.5%), which were determined based on electricity prices from the Dutch EPEX day-ahead spot market. Furthermore, the heat pumps capacity (maximum heat output) could be reduced around 30%, supporting the hypothesis that by having short-term storage, the capacity of thermal producing equipment may be selected closer to the average load, rather than peak conditions. Regarding electricity consumption and total heat production, it was noticed that the introduction of short-term storage led to an increase in total heat output from heat pumps (~7%), mainly due to higher thermal losses. However, the global heat pumps efficiency is also significantly higher (~14%), which results in less electricity consumption (~-13%), despite of the higher heat production.

In what concerns system 2, it seems that the heat production based on the overall district demand does not result in any improvements, when comparing with system 1. In short, the heat pumps efficiency was slightly lower and it was noticed an increase in distribution thermal losses, that were compensated by lower storage related losses. However, the results obtained regarding this configuration are somewhat inconclusive, as an analysis regarding the centralized operation should not take into account technical characteristics only, but also economic, social and the local overall impact. Moreover, mainly due to lack of data, several aspects were not taken into account during implementation.

All things considered, it was very hard and time consuming to implement/represent an energy system at the district level using MATLAB. In addition, the time step of 1 hour is also a limiting factor, since it is too long for certain processes to be proper analyzed. For instance, it only allowed a very simple storage control to be implemented. Another limiting factor, was the fact that during literature review, very few reports describing methods to implement energy systems at the district scale, were found.

The choice of matlab was a last resort decision, and in future work a more appropriate software should be chosen when analyzing energy systems at the district level. The increasing complexity of the relations in energy systems, due to the emergence of new technologies such as the latest generation district energy systems, forces designers and researchers to use software with the capability for integrated and dynamic simulations. Only this way, it is possible to properly embrace the different components and complex interactions in an acceptable time.

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Appendix A - Neighborhood Case Study Related Inputs

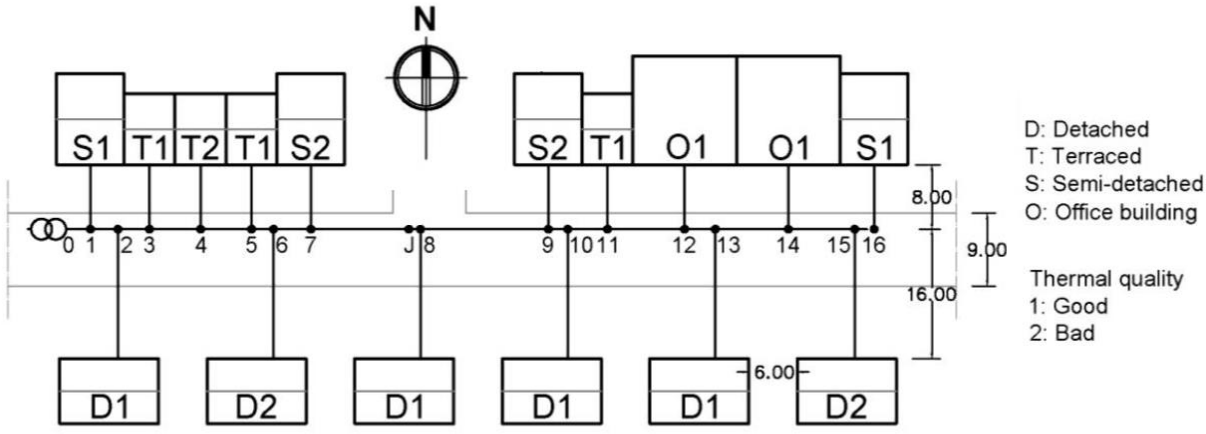

Figure A.1 - Layout of the different building typologies which form the neighborhood case study.

Table A.1 - Distance in meters between the connection nodes of the buildings.

Node interval	0-1	1-2	2-3	3-4	4-5	5-6	6-7	7-J	J-8	8-9	9-10
Length [m]	3.0	3.4	3.9	6.3	6.3	2.7	4.6	12.1	1.4	15.8	2.4

Node interval	10-11	11-12	12-13	13-14	14-15	15-16
Length [m]	4.9	9.5	3.8	9.0	8.2	2.4

Construction Elements for each thermal quality level

Table A.2 - Thermal insulation of the main construction elements of the building envelope, for thermal quality 1.

Construction	Elements	U-value [W/m^{2}K]
Façade	Cavity wall with 6 cm mineral wool wall insulation	0.4
Roof	10 cm mineral wool roof insulation between rafters or 8 cm XPS on concrete flat roof	0.3
Floor	5 cm XPS floor insulation below screed	0.4
Windows	Insulated profiles U_f =1.8 and high-performance glazing U_g = 1.1	1.7
Doors	Insulated door leafs	2.9

Table A.3 – Thermal insulation of the main construction elements of the building envelope, for thermal quality 2.

Construction	Elements	U-value [W/m^{2}K]
Façade	Outer brick leaf, air cavity 5 cm, inner brick leaf	1.7
Roof	Wooden roof construction with tiles	2.85
Floor	Concrete structural floor, floor screed and floor finishing	3.0
Windows	Wooden window profiles – single glazing	5.0
Doors	Un-insulated door leafs	4.0

Table A.4 – Infiltration rate for each thermal quality group.

Thermal quality	Infiltration rate
#1	0.1 ACH
#2	0.5 ACH

Occupancy Profiles and set-points per building typology

Table A.5 – Occupancy, heating schedules and temperature set-points applied in each building typology.

Building typology	Heating schedule	Occupancy schedule	Temperature set-point
D1	Week: 18-24h Weekend: 8-24h	2 people Week: 18-8h Weekend: 24/7	Heating: 21ºC Cooling: 23ºC
D2	8-24h	2 people 24/7	
T1	Week: 18-24h Weekend: 8-24h	4 people Week: 18-24h Weekend: 24/7	
T2	8-24h	1 person 24/7	
S1	24/7	2 people Week: 18-24h Weekend: 24/7	Heating/day: 21ºC Heating/night: 16ºC Cooling: 23ºC
S2	24/7	2 people 24/7	
O1a	24/7	Table A.6	
O1b	24/7	Table A.6	

Table A.6 – Detailed office buildings occupancy schedule.

Time Period	O1a [0-1]	O1b [0-1]
0-8 h	0	0
8-9 h	0.7	0.9 < Random < 1
9-12h	1	
12-13h	0.7	
13-18h	1	
18-19h	0.7	
19-24 h	0	0

Maximum_people = 0.1*Floor_area

People (h) = [0-1]*Maximum people

Table A.7 - Internal gains due to equipment. (Low = $2 W/m^2$, Medium = $4 W/m^2$; High = $6 W/m^2$)

<table>
<tr><th>Time Period</th><th>D1, T1, S1</th><th>D2, T2, S2</th><th>O1a</th><th>O1b</th></tr>
<tr><td>0-8 h</td><td rowspan="5">Low</td><td>Low</td><td>Low</td><td>Low</td></tr>
<tr><td>8-9 h</td><td rowspan="2">Medium</td><td>Medium</td><td rowspan="4">High</td></tr>
<tr><td>9-12h</td><td>High</td></tr>
<tr><td>12-13h</td><td>High</td><td>Medium</td></tr>
<tr><td>13-18h</td><td>Medium</td><td>High</td></tr>
<tr><td>18-19h</td><td>Medium</td><td>High</td><td>Medium</td><td>Medium</td></tr>
<tr><td>19-24 h</td><td>High</td><td>Medium</td><td>Low</td><td>low</td></tr>
</table>

Appendix B - MATLAB relevant code

I – Main functions

Heating

Reference Case

```
function [m_rad_flow, m_flow_STES, Energy_hot_extracted_STES, Energy_compressor_Work,…
Electricity_consumption_HP, Energy_produced, Energy_consumed, COP, COP_mean,…
Electricity_price, Annual_electricity_price, Total_Energy_produced, Energy_Losses,…
Energy_output_HP] = Space_Heating_Reference_Case (Heating_Load)

%% INPUTS & OTHERS
Time = (1:1:8760)'; %time step = 1 hour
cp_water_40 = cp_water (40); %Specific heat [J/KgC] at 40ºC
cp_water_12_225 = cp_water (12.225); %Specific heat [J/KgC] at 12.5ºC
Tw_Hot_STES = importdata ('STES_HOT.txt'); % Water temperature from cold reservoir
e_price = importdata ('Simple_contract.txt'); %input electricity price
[Heat_peak_value, ~] = max (Heating_Load);% Heating peak info

%% RADIATOR
%{ It calculates the inlet and outlet temperatures for a given water mass flow and heat
demand, based on the radiator European Norm EN 442 heat output calculation methodology. %}

T_air_nominal = 20.5; %Air temperature at nominal condition (System ON)
Tw_sup_nominal = 50; %Water inlet temperature at nominal/design condition
Tw_ret_nominal = 30; %water outlet temperature at nominal/design condition
n = 1.32; %Exponent for radiator heat transfer

% Nominal radiator inlet and outlet temperatue difference
deltaT_rad_nominal = Tw_sup_nominal - Tw_ret_nominal;

% Temperature difference between radiator's average temperature and surrounding air
deltaT_rad_design = ((Tw_sup_nominal + Tw_ret_nominal)/2) - T_air_nominal;
 m_rad_flow_nominal = Heat_peak_value / (cp_water_40 * deltaT_rad_nominal);

% Computes the required radiator inlet and outlet temperatures
delta_T = ((Heating_Load ./ Heat_peak_value).^(1/n)) .* deltaT_rad_design;
MWT = T_air_nominal + delta_T;
MWT_deviation_nominal = (Heating_Load ./ (2*cp_water_40*m_rad_flow_nominal));

Tin_rad_nominal = MWT + MWT_deviation_nominal;
Tout_rad_nominal = MWT - MWT_deviation_nominal;

%% WATER-TO-WATER HEAT PUMP
%% 1 - THERMODYNAMICS

T_source = Tw_Hot_STES;
T_condenser  = Tin_rad  + 10; %Hot side temperature
T_evaporator = T_source - 10; %Cold side temperature
compressor_eff = 0.8;

% H4 = Liquid Saturated at T_condenser temperature
% Temperature vs. Enthalpy Liquid Saturated R134-A R^2 = 0.9998 WEBOOK.nist.gov
H4 = 0.0027.*(T_condenser).^2 + 1.3237.*T_condenser + 51.267; %[KJ/Kg]
% S3s = S2 isentropic compressor
% S2 = Saturated vapor at T_evaporator temperature R134-A R^2 = 0.9996
S2 = -1E-07.*(T_evaporator).^3 + 9E-06.*(T_evaporator).^2 - 0.0006.*T_evaporator + 0.9313;
S3s=S2;
```

```
H2 = -3E-05.*(T_evaporator).^3 + 0.0001.*(T_evaporator).^2 + 0.5696.*(T_evaporator) +
250.39; %[KJ/Kg]  R^2 = 0.9999

% P3 = P3s = P4 = Psat @ T_condenser
P_sat_MPa = 9E-07.*(T_condenser).^3 + 0.0002.*(T_condenser).^2 + 0.0106.*(T_condenser) +
0.2722; % [MPa] R^2 = 0.9997

% ISOBARIC CURVES
H3s = zeros (8760, 1);

for t = Time(1):Time(end)

if P_sat_MPa (t) < 0.55
    H3s (t) = 58.578.*S3s(t).^2 + 223.68.*S3s(t) + 0.8035;

elseif P_sat_MPa(t) >= 0.55 && P_sat_MPa(t) < 0.65
    H3s(t) = 59.452.*S3s(t).^2 + 226.61.*S3s(t) + 1.2009;
```

… (similar code removed)

```
elseif P_sat_MPa(t) >= 3.35 && P_sat_MPa(t) < 3.45
    H3s(t) = 76.592.*S3s(t).^2 + 238.57.*S3s(t) + 2.724;

else
    H3s(t) = 77.432.*S3s(t).^2 + 238.67.*S3s(t) + 2.7569;

end
end

H3 = H2 + ((H3s - H2)./compressor_eff);

HE_eff = 0.95;
Q_HP = ((m_rad_flow .* cp_water_40 .* (Tin_rad - Tout_rad))./HE_eff)./1000; %[KW]

%% 2 - COMPUTES ELECTRICITY AND COMRPESSOR WORK

m_r143a = Q_HP ./ (H3 - H4); % Mass flow rate of the R-134a
W_comp_temp = (m_r143a .* (H3 - H2)).*1000; % Compressor work [W]
[max_W_comp_temp, ~] = max (W_comp_temp);

MIN_COMP(1:8760) = 0.33.*max_W_comp_temp; % Turn down ratio = 1:3
W_comp_min = MIN_COMP';

W_comp = zeros(8760, 1);
for t = Time(1):Time(end)
    if W_comp_temp(t) <= W_comp_min(t) && W_comp_temp(t) > 0
        W_comp(t) = W_comp_min(t);
    elseif W_comp_temp(t) == 0
        W_comp(t) = 0;
    elseif W_comp_temp(t) > W_comp_min(t)
        W_comp(t) = W_comp_temp (t);
    end
end

COP = (Q_HP.*1000)./ W_comp; % Coefficient of Performance HP

%{Added to the compressor power will be the circulation pump power.This can vary
considerably. It should be between 5 and 10% of the power input.%}

Electricity_input_HP = W_comp ./ 0.925; %Electricity input HP
LWT_source = 4; %Source leaving water temperature
delta_T_source = T_source - LWT_source;
Q_from_STES_hot = Q_HP.*1000 - W_comp;% Heat extracted from STES water
Q_from_STES_hot (Q_from_STES_hot<=0) = 0;
m_flow_STES_hot = Q_from_STES_hot ./ (cp_water_12_225 .* delta_T_source);
```

```
%% Results

m_rad_flow = m_rad_flow.*1;
m_flow_STES = m_flow_STES_hot;

COP = COP.*1;
COP_mean = nanmean (COP);

Energy_hot_extracted_STES = Q_from_STES_hot./1000; %[Kwh]
Energy_compressor_Work = W_comp./1000; %[kwh]
Electricity_consumption_HP = Electricity_input_HP./1000; %[Kwh]
Energy_produced = Energy_hot_extracted_STES + Energy_compressor_Work; %[Kwh]
Energy_output_HP = Q_HP; %[KWh]
Energy_consumed = Heating_Load./1000; %[Kwh]

Electricity_price = Electricity_consumption_HP .* e_price; %[€/h]
Annual_electricity_price = sum(Electricity_price);%[€]
Total_Energy_produced = sum (Energy_produced);%[kwh]
Energy_Losses = sum(Energy_produced) - sum(Energy_consumed);

End
```

System 1 (Short-term storage at each building)

```
function [m_rad_flow, m_flow_STES, m_flow_st, COP, COP_mean, Energy_hot_extracted_STES,…
Energy_compressor_Work, Electricity_consumption_HP, Energy_produced, Energy_consumed,…
Electricity_price, Annual_electricity_price, Energy_Losses, Energy_losses_storage,…
Total_Energy_produced] = Space_Heating_system1 (Heating_Load)

%% INPUTS
Time = (1:1:8760)'; %time step = 1 hour
cp_water_40 = cp_water (40); %Specific heat [J/KgC] at 40ºC
rho_water_40 = 992.2; % water density [Kg/m^3] at 40ºC
e_price = importdata ('double_contract.txt'); %input
Tw_Hot_STES = importdata ('STES_HOT.txt'); % Water temperature from hotreservoir
cp_water_12_225 = cp_water (12.225); %Specific heat [J/KgC] at 12.5ºC

%% RADIATOR
…(Removed code, same code as in “radiator” section of reference case)

eff_dist = 0.95;
Qload1_W = (m_rad_flow .* cp_water_40 .* (Tin_rad - Tout_rad))/eff_dist;

%% Consecutive 24h cumulative load
Time_horizon_24 = 24;
Q24 = zeros (Time(end)-(Time_horizon_24), 1);
for n =1:1:Time(end)-Time_horizon_24
    Q24(n)=sum(Qload_W(n:n+(Time_horizon_24-1)));
end
for z = 1:1:Time_horizon_24;
      Q24(end+1) = sum(Qload_W((Time(end)-Time_horizon_24)+z:end));
end

%% Consecutive 2h cumulative load
Time_horizon_2 = 2;
Q2 = zeros (Time(end)-(Time_horizon_2), 1);
for n =1:1:Time(end)-Time_horizon_2
    Q2(n)=sum(Qload_W(n:n+(Time_horizon_2-1)));
end
for z = 1:1:Time_horizon_2;
      Q2(end+1) = sum(Qload_W((Time(end)-Time_horizon_2)+z:end));
end

%% Water tank storage Sizing & nominal conditions
```

```
%{- The short term thermal energy storage is designed to supply 50% of the
 required load for the 24 consecutive hours in which highest demand occurs;
 - For design purposes, it is defined that the thermal energy stored in the
 tank is null when the average temperature of the stored water is equal to
 the average annual return temperature from the radiators; %}

[peak_load, ~] = max (Qload_W);% Heating peak info
[Q24_max, ~] = max (Q24);
f_part_st = 0.5;
Qst_design_wh = Q24_max * f_part_st; %[Wh]
Qst_design = (Qst_design_wh * 3600); %[J]
rad_ret = Tout_rad;
rad_ret(rad_ret==20.5)=NaN;
st_min_T_design = nanmean (rad_ret);
st_max_T_design = Tw_sup_nominal + 10; %60ºC
m_st = Qst_design/((st_max_T_design-st_min_T_design) * cp_water_40); %[Kg]
V_st = m_st/rho_water_40; %[m^3]

%Heat-pump
HP_dim_factor = 0.75;
HP_design_power = HP_dim_factor*peak_load; %[W]
delta_T_HP_nominal = 15;
m_HP_flow_nominal = HP_design_power./(cp_water_40.*delta_T_HP_nominal);
m_HP_max_fac = 1.25;
m_HP_flow_max = m_HP_max_fac*m_HP_flow_nominal;

%% storage loss factor [W/K] determination
%{Computes the storage loss factor [W/K] by the following steps:
1- Computes minimum surface area of a cylindric tank for a specific volume;
2- The defined R value is the R15 on the ASHRAE standardized insulation rating;
3 - The SI units R value = 0.1761101838 * ASHRAE R value (American to
European units);
4 - U = 1/R;%}

syms r
Asup_diff = 4*pi*r - 2*V_st*r^-2 == 0;
min_r_sym = solve(Asup_diff,r);
min_r = double(min_r_sym);
r_real = real(min_r);
r=r_real(r_real>0);
h_cil=(V_st)/(pi*r^2);
Asup_min = 2*pi*r^2 + 2*V_st/r;

R_ASHRAE = 15;
R_SI = R_ASHRAE.*0.1761101838;
U = 1./R_SI; %[W/m^2K]
Ut = U*Asup_min; %[W/K]

%% Storage related
Tamb = zeros (8760, 1);
for k = 1:1:Time(end)
    if k>= 3600 || k<= 5760
        Tamb(k) = 23;
    else
        Tamb (k) = 14;
    end
end
Qload_J = Qload_W*3600;

% Consecutive mean 24h electricity price
Time_horizon_24 = 24;
e_price24 = zeros (Time(end)-(Time_horizon_24), 1);
for n =1:1:Time(end)-Time_horizon_24
    e_price24(n)=mean(e_price(n:n+(Time_horizon_24-1)));
end
for z = 1:1:Time_horizon_24;
```

```
      e_price24(end+1) = mean(e_price((Time(end)-Time_horizon_24)+z:end));
end

Qst = zeros(8760, 1); Tst = zeros(8760, 1); Tcharge = zeros(8760, 1); Qproduced =
zeros(8760, 1); m_flow = zeros(8760, 1); Qloss = zeros(8760, 1);

 %initial conditions
Qst(1) = 0;
Tst(1) = st_min_T_design;
Tcharge(1) = Tst(1) + delta_T_HP_nominal;
Qproduced(1) = HP_design_power*3600;
m_flow(1) = Qproduced(1)/((Tcharge(1)-Tst(1))*cp_water_40);
Qloss(1) = (Ut.*(Tst(1)-Tamb(1))).*3600;

Q24_J = Q24 .*3600;
Q2_J = Q2 .* 3600;

T_charge_max =60;
E_st = 0.96; %Heat exchanger effectiveness (Constant)
max_fact_turn_off = 0.97;

for h=2:1:Time(end)-1
    Qst(h) = Qst(h-1) + Qproduced(h-1).*E_st - Qloss(h-1) - Qload_J(h-1);%[J] %Qstored for
hour h
    Tst(h) = ((Qst(h) - Qst(h-1))./(m_st*cp_water_40)) + Tst(h-1);%[ºC] %Average water
temperature for hour h
    if e_price(h) <= e_price24(h) %Check if electrity cost is below the average cost for
next 24h
        if Qst(h) < Q24_J(h) && Qst(h) < Qst_design*max_fact_turn_off %check if Energy
stored is less than the required energy for the next 24h
            Tcharge(h) = Tst(h) + delta_T_HP_nominal;
            if Tcharge(h) >= T_charge_max%max limit T_charge
                Tcharge(h) = T_charge_max;
            else
                Tcharge (h) = Tcharge (h);
            end
            m_flow(h) = (HP_design_power*3600)./(cp_water_40.*(Tcharge(h) - Tst(h)));
%[Kg/h]
            if m_flow(h) >= (m_HP_flow_max*3600)%max limit the m_flow
                m_flow(h) = (m_HP_flow_max*3600);
            else
                m_flow(h) = m_flow(h);
            en
            Qproduced(h) = m_flow(h) .* cp_water_40 .* (Tcharge(h)-Tst(h));
        else
            m_flow(h) = 0;
            Qproduced(h) = m_flow(h) .* cp_water_40 .* (Tcharge(h)-Tst(h));
        end
    else %if the cost is above the average then...
        if Qst(h) < Q2_J(h) && Qst(h) < Qst_design*max_fact_turn_off
            Tcharge(h) = Tst(h) + delta_T_HP_nominal
            if Tcharge(h) >= T_charge_max % max limit T_charge
                Tcharge(h) = T_charge_max;
            else
                Tcharge (h) = Tcharge (h);
            end
            m_flow(h) = (HP_design_power*3600)./(cp_water_40.*(Tcharge(h) - Tst(h)));
            if m_flow(h) >= m_HP_flow_max*3600 %max limit m_flow
                m_flow(h) = m_HP_flow_max*3600;
            else
                m_flow(h) = m_flow(h);
            end
            Qproduced(h) = m_flow(h) .* cp_water_40 .* (Tcharge(h)-Tst(h));
        else
            m_flow(h) = 0;
            Qproduced(h) = m_flow(h) .* cp_water_40 .* (Tcharge(h)-Tst(h));
```

```
        end  %Check if Energy stored is less than the required energy for the next 2h
    end
    Qloss(h) = (Ut.*(Tst(h)-Tamb(h))).*3600;  %thermal losses
end

%% WATER-TO-WATER HEAT PUMP
%% 1 - THERMODYNAMICS
```

…(Removed code, same code as in "radiator" section of reference case)

```
%% 2 - COMPUTES ELECTRICITY AND COMRPESSOR WORK
```

…(Removed code, same code as in "radiator" section of reference case)

```
%% RESULTS

m_flow_STES = m_flow_STES_hot;%[Kg/s]
m_flow_st = m_flow./3600;%[Kg/s]
m_rad_flow = m_rad_flow.*1;%[Kg/s]
m_flow_work_fluid = m_r143a;
COP = COP.*1;
Energy_hot_extracted_STES = Q_from_STES_hot./1000; %[Kwh]
Energy_compressor_Work = W_comp./1000; %[kwh]
Electricity_consumption_HP = Electricity_input_HP./1000; %[Kwh]
Energy_produced = Q_HP; %[Kwh]
Energy_consumed = Heating_Load./1000; %[Kwh]
Energy_stored = (Qst./3600)./1000; %[Kwh]
Energy_Losses = sum(Energy_produced) - sum(Energy_consumed);
Energy_losses_storage = (Qloss./3600)./1000; %[Kwh]
Electricity_price = Electricity_consumption_HP .* e_price; %[€/h]
Design_storage_capacity = (Qst_design./3600)./1000; %[kwh]
Filling_degree = Energy_stored./Design_storage_capacity; %[Kwh]
```

System 2 (Centralized heat production and storage)

```
Time = (1:1:8760)'; %time step = 1 hour
cp_water_40 = cp_water (40); %Specific heat [J/KgC] at 40ºC
rho_water_40 = 992.2; % water density [Kg/m^3] at 40ºC
Heating_Load = importdata ('District_Heating.txt');%hourly input data from dymola [W]or[Wh]
e_price = importdata ('Double_contract_MT.txt'); %input electricity tariff
Tw_Hot_STES = importdata ('STES_HOT.txt'); % Water temperature from hotreservoir
Tw_Cold_STES = importdata ('STES_COLD.txt'); % Water temperature from cold reservoir
cp_water_12_225 = cp_water (12.225); %Specific heat [J/KgC] at 12.5ºC
cp_water_45_5 = cp_water(45.5);
Out_T = importdata ('weather_T.txt'); %Outside temperature

%% Supply temperature curve
T_sup_max = 55; T_sup_min = 35; T_ret = 30;
T_supply = zeros (8760, 1);

for t=1:1:Time(end)
    if Out_T(t) <= -7
        T_supply(t) = T_sup_max;
    elseif Out_T(t) >= 15
        T_supply(t) = T_sup_min;
    else
        T_supply(t) = -0.9091.*Out_T(t) + 48.636;
    end
end

m_flow_dist = Heating_Load ./ (cp_water_40 .* (T_supply - T_ret));%[kg/s]
Qload_W = m_flow_dist .* cp_water_40 .* (T_supply - T_ret);%[W]
```

```
%% Consecutive 24h cumulative load
%% Consecutive 2h cumulative load
%% Water tank storage Sizing & nominal conditions
% Consecutive mean 24h electricity price
%% WATER-TO-WATER HEAT PUMP
%% 1 - THERMODYNAMICS

T_source = Tw_Hot_STES;
T_condenser  = Tcharge  + 10; %Hot side temperature
T_evaporator = T_source - 10; %Cold side temperature
compressor_eff = 0.85;

%% 2 - COMPUTES ELECTRICITY AND COMRPESSOR WORK

%%                             --- Hydraulic ---

[Velocity_STES_heating, Total_pressure_loss_STES_heating, Hidraulic_Power_STES_heating]  =
Pipe_Hidraulic_Power (m_flow_STES, 25, 20, 2.5, 0, 0, 0, 0, 0, 100000);

Electricity_consumption_pump_ATES_heating = Hidraulic_Power_STES_heating ./ 0.7;

[Velocity_supply, Total_pressure_loss_supply, Hidraulic_Power_supply] =
Pipe_Hidraulic_Power (m_flow_dist, 300, 0, 2.5, 0, 0, 0, 0, 0, 175000);

Electricity_consumption_pumping_supply = Hidraulic_Power_supply ./ 0.7;
zzzz = reshape (Electricity_consumption_pumping_supply, 730, []);
Monthly_Electricity_consumption_pump_supply = sum (zzzz);

%%                             --- Thermal losses ---

T_soil = Soil_T (1.5);

[~, ~, ~, Energy_Loss_distribution] = THERMAL_LOSSES_PIPE (m_flow_dist, 2.5, 150, T_supply,
T_soil);

Energy_Loss_distribution = Energy_Loss_distribution.*(-1);
Energy_Loss_distribution (isnan(Energy_Loss_distribution))=0;
g = reshape (Energy_Loss_distribution, 730, []);
Monthly_Energy_Loss_distribution = sum (g);
```

Cooling

```
function [m_Air_flow, m_flow_STES, Energy_transfered_coil, Energy_consumed] = Space_Cooling
(Cooling_file)

%Cooling_file = importdata ('D1_Cooling.txt'); %Data from dymola
cp_Air_dry = 1.005; %Specific heat [kJ/KgC] at 40ºC
cp_Water = cp_water (6);
Time = (1:8760)';
 Out_Air_Temperature = importdata ('weather_T.txt'); %Outside air temperature
phi = importdata ('phi.txt'); %Outside air relative humidity [%]
Tw_cold_STES = importdata ('STES_COLD.txt'); % Water temperature from cold reservoir

% barometric pressure from elevation in kPa
elevation = 13;
P = (29.921 .* ((1 - 6.87535e-06 .* elevation./0.3048).^5.256).*3.38650);

%% DESIGN CONDITIONS
Ta_nominal = 23.5; % Nominal room air temperature (System ON)
Ta_supply_nominal = 18; % Air inlet temperature at nominal/design condition
Ta_ret_nominal = Ta_nominal; % Air outlet temperature at nominal/design condition
Cooling_Load = -1 .* Cooling_file;
```

```
[~, h_sup_nom] = Enthalpy (Ta_supply_nominal, 55); %Enthalpy supply air
[~, h_ret_nom] = Enthalpy (Ta_ret_nominal, 55);%Enthaply return air
delta_h_nominal = h_ret_nom - h_sup_nom;
mAir_flow_rate = Cooling_Load ./ (delta_h_nominal*1000);

%% HEAT RECOVERY (HEAT EXCHANGER WITH CONSTANT EFFECTIVENESS - DRY)
E = 0.7; % Heat recovery effectiveness
[w_out_air, h_out] = Enthalpy (Out_Air_Temperature, phi); %Enthalpy supply air [KJ/KG]

%{Temperature on cooling surf is ALWAYS above or equal to the dew point temperature
of the surrounding air, the air will be cooled without any change in specific humidity.%}

Cmin = mAir_flow_rate .* cp_Air_dry; %Min transfer occurs when air is dry;
delta_T_recovery = Out_Air_Temperature - (Ta_ret_nominal + 0.25);
Q_HR = E .* Cmin .* delta_T_recovery;  % heat_recovered
delta_T_recovery = -Q_HR ./ (mAir_flow_rate.* cp_Air_dry);
T_air_after_recovery = Out_Air_Temperature + delta_T_recovery;

% phi calculation from Tdb and w
Pw=w_out_air.*P./(0.621945+w_out_air); %partial pressure of water vapor
T_db_absolute=T_air_after_recovery+273.15;
Pws_0=exp(-(5.8002206e3)./T_db_absolute+1.3914993+-(4.8640239e-
2).*T_db_absolute+(4.1764768e-5).*(T_db_absolute.^2)-(1.4452093e-
8).*(T_db_absolute.^3)+6.5459673.*log(T_db_absolute)); %in Pa valid for 0 to 200C
Pws=Pws_0./1000; % in kPa
phi_after_recovery=Pw./Pws.*100;

%% COOLING COIL

[~, h_inlet_coil] = Enthalpy (T_air_after_recovery, phi_after_recovery); %Enthalpy supply
air [KJ/KG]
[~, h_outlet_coil] = Enthalpy (Ta_supply_nominal, 55);

Q_coil = (mAir_flow_rate .* (h_inlet_coil - h_outlet_coil)).*1000; %[W]
Q_coil(isnan(Q_coil))=0;
[~, h_coil_surf] = Enthalpy (Tw_cold_STES, 95);
contact= (h_inlet_coil - h_outlet_coil) ./ (h_inlet_coil - h_coil_surf); % 0-1
%bypass = 1 - contact; %0 - 1;
T_water_hot_setpoint = 16;
delta_T_water_Coil = T_water_hot_setpoint - Tw_cold_STES;
mW_STES_flow_Cold = Q_coil ./ (cp_Water .* delta_T_water_Coil);


%% Results

m_flow_STES = mW_STES_flow_Cold; %[Kg/s]
m_Air_flow = mAir_flow_rate; %[Kg/s]
contact_factor = contact; %[0-1]
Tair_admited = T_air_after_recovery - delta_T_recovery;%[C]
Tair_recovery = T_air_after_recovery;%[C]
delta_T_supply = 16 - T_air_after_recovery;
Tair_supply = T_air_after_recovery + delta_T_supply;%[C]
delta_phi_recovery = phi - phi_after_recovery;
phi_admited = phi_after_recovery + delta_phi_recovery;%humidity ratio [%]
phi_recovery = phi_after_recovery;%humidity ratio [%]
delta_phi_supply = 55 - phi_after_recovery;
phi_supply = phi_after_recovery + delta_phi_supply;%humidity ratio [%]
h_recovery = h_inlet_coil; %enthalpy [KJ/Kg]
delta_h_recovery = h_inlet_coil - h_outlet_coil ;
h_supply = h_inlet_coil - delta_h_recovery;%enthalpy [KJ/Kg]
delta_T_h = h_out - h_inlet_coil;
h_admitted = h_inlet_coil + delta_T_h;%enthalpy [KJ/Kg]
Energy_transfered_coil = Q_coil./1000;%[Kwh]
Energy_consumed = Cooling_Load./1000;%[Kwh]

end
```

II – Secondary functions

Thermal losses Pipe

```
function [m_flow, Velocity, Tout, Toscilation, Energy_Loss] = THERMAL_LOSSES_PIPE
(m_flow_rate, nominal_velocity, length, T_in, T_se)

Time = (1:8760)';
m_flow_rate(m_flow_rate<=0) = 0;
Pr = (-6E-5.*(T_in.^3)) + (0.0085.*(T_in.^2)) - 0.48.*T_in + 13.617;
cp_water = 4179.6;%water specific heat at 30ºC
rho_water = 995.6; %water density at 30ºC
nu_water = 0.801e-6; %Kinematic viscosity [m^2/s]

%% Nominal conditions
V_flow_rate = m_flow_rate ./ rho_water; %volumetric flow rate (m^3/s)
[V_flow_rate_nominal, ~] = max (V_flow_rate); %Nominal volume flow rate
%Computes the diameter according to the nominal conditions
D = ((4 * V_flow_rate_nominal) / (pi * nominal_velocity))^0.5;
%Computes water velocity through pipe
velocity = (4.*V_flow_rate)./(pi*(D)^2);

R = D/2;
A0 = 2*pi * R*length;

R_ASHRAE = 18;
R_SI = R_ASHRAE.*0.1761101838;%[m^2K/W]
U = 1./R_SI;
Ut = U * A0;

R_Total = 1/Ut;

delta_T_in = T_se - T_in;
pow = -1./(m_flow_rate.*cp_water.*R_Total);
T_out = ((exp(pow).*delta_T_in)-T_se).*(-1);
T_oscilation = T_out - T_in ;
delta_T_out = T_se - T_out;
dT_ml = (delta_T_out - delta_T_in)./(log(delta_T_out./delta_T_in));
Q_loss = (dT_ml./(R_Total));

%Q_loss(isnan(Q_loss))=0;
%Annual_Q_loss = sum (Q_loss, 1);
%% Results

m_flow = m_flow_rate;
Velocity = velocity;

Tout = T_out;
Toscilation = T_oscilation;

Energy_Loss = Q_loss./1000; %[Kwh]

end
```

Hydraulic Power (pumping)

```
function [Velocity, Total_pressure_loss, Hidraulic_Power]  = Pipe_Hidraulic_Power
(m_flow_rate, length, z, nominal_velocity, nr_meter, nr_elbow_90, nr_tee, nr_valve,
nr_check_valve, others)

%% INPUTS
Time = (1:8760)';
g = 9.81; % standard gravity [m^2/s]
rho_water = 995.6; %water density at 30ºC
nu_water = 0.801e-6; %Kinematic viscosity [m^2/s] |

%% Nominal conditions
V_flow_rate = m_flow_rate ./ rho_water; %volumetric flow rate (m^3/s)
[V_flow_rate_nominal, ~] = max (V_flow_rate); %Nominal volume flow rate
%Computes the diameter according to the nominal conditions
D = ((4 * V_flow_rate_nominal) / (pi * nominal_velocity))^0.5;
%Computes water velocity through pipe
velocity = (4.*V_flow_rate)./(pi*(D)^2);

%% Reynold's number & friction factor (smooth pipe)
 Re = (D.*velocity)/nu_water; %Reynold's number
f = zeros(8760, 1);
for t=Time(1):Time(end)
    if Re(t) < 2300
        f(t) = 64/Re(t); % pipe friction factor if laminar
    else
        f(t) = (0.790.*log(Re(t))-1.64).^-2;% friction factor if turbulent
    end
    f(isinf(f))= 0;
end

%% PRESSURE LOSSES  @Piping Calculations (McGraw-Hill Calculations)
%% 1 - Due to elevation
    delta_P_elevation = zeros(8760, 1);
for t=Time(1):Time(end)
    delta_P_elevation(t) = rho_water*g*z; %total vertical head loss [Pa]
end

%% 2 - Due to friction
 delta_P_friction = f.* (length/D) * (rho_water/2) .* velocity.^2; %[Pa]

%% 3 - Due fittings & others
%3.1 Turbine-wheel water meter
%nr_meter = 1;
k_w_meter = 6; %K-value method
h_meter = ((k_w_meter.*velocity.^2)./(2*g)) .* nr_meter; %equivalent lenght
delta_P_meters = 9804.1 .* h_meter;

%3.2 90 degrees elbow curved
%nr_elbow_90 = 1;
k_elbow = (800./Re) + 0.2*(1 + (1/D));% 2K-value method (K1=800 & k_inf = 0.2)
h_elbow = ((k_elbow.*velocity.^2)./(2*g)) .* nr_elbow_90;
h_elbow(isnan(h_elbow))=0;
delta_P_elbows = 9804.1 .* h_elbow;

%3.3 Tee, Run Through (sub-in_type Branch)
%nr_tee = 1;
k_tee = (100./Re); % 2K-value method (K1=100 & k_inf = 0)
h_tee = ((k_tee.*velocity.^2)./(2*g)) .* nr_tee;
h_tee(isnan(h_tee))=0;
delta_P_tee = 9804.1 .* h_tee;

%3.4 Diaphragm Valve
```

```
%nr_valve = 1;
k_valve = (1000./Re) + 2*(1 + (1/D));% 2K-value method (K1=1000 & k_inf = 2)
h_valve = ((k_valve.*velocity.^2)/(2*g)) .* nr_valve;
h_valve(isnan(h_valve))=0;
delta_P_valve = 9804.1 .* h_valve;

%3.5 Check Valve (Swing)
%nr_check_valve = 1;
k_valve_check = (1000./Re) + 0.5*(1 + (1/D));% 2K-value method (K1=1000 & k_inf = 0.5)
h_valve_check = ((k_valve_check.*velocity.^2)/(2*g)) .* nr_check_valve;
h_valve_check(isnan(h_valve_check))=0;
delta_P_valve_check = 9804.1 .* h_valve_check;

% Total head loss and respective conversion to Pascal (1 meter = 9804.1 Pa)
delta_P_others = delta_P_meters + delta_P_elbows + delta_P_tee + delta_P_valve +
delta_P_valve_check;

%% P2 - P1 TOTAL
delta_P = (delta_P_elevation + delta_P_friction + delta_P_others + others); %Pressure Loss
(Pa)

Hidraulic_Power = delta_P .* V_flow_rate; %Hidraculic Power [W]

%% Results

m_flow = m_flow_rate;
Velocity = velocity.*1;
Reynolds = Re;
friction_factor = f;
Elevation_losses = delta_P_elevation;
Friction_losses = delta_P_friction;
Other_losses = delta_P_others + others;
Total_pressure_loss = delta_P;
Hidraulic_Power = Hidraulic_Power./1000;

end
```

Enthalpy

```
function [w, h] = Enthalpy (Tdb, phi)

% VARIABLES
% Tdb (dry bulb temperature)
% Tdp(dew point temperature) in C
% w (humidity ratio) in kg./kg of dry air
% phi (relative humidity) in %
% h (enthalpy) in J./kgK of dry air
% P (atmospheric pressure) in kPa
%c_air = 1006; %J./kg, value from ASHRAE 2013 Fundamentals eq. 32
hlg = 2501000; %J./kg, value from ASHRAE 2013 Fundamentals eq. 32
%cw  = 1860; %J./kg, value from ASHRAE 2013 Fundamentals eq. 32

%Tdb = 21; %
%phi = 60;

elevation = 13;
T_db_absolute = Tdb + 273.15;

%specific heat of dry air (Shapiro 2ed)
c_air = ((3.653 + -1.337e-3.*T_db_absolute + 3.294e-6.*T_db_absolute.^2 + -1.913e-
9.*T_db_absolute.^3 + 0.2763e-12.*T_db_absolute.^4 ) .* 0.287055).*1000;
```

```
%specific heat of water vapor (Shapiro 2ed)
cw = ((4.070 + -1.108e-3.*T_db_absolute + 4.152e-6.*T_db_absolute.^2 + -2.964e-
9.*T_db_absolute.^3 + 0.807e-12.*T_db_absolute.^4 ) .* 0.461520).*1000;

% barometric pressure from elevation in kPa
P = (29.921 .* ((1 - 6.87535e-06 .* elevation./0.3048).^5.256).*3.38650);

% w calculation from Tdb and phi
T_db_absolute=Tdb+273.15;
Pws_0=exp(-(5.8002206e3)./T_db_absolute+1.3914993+-(4.8640239e-
2).*T_db_absolute+(4.1764768e-5).*(T_db_absolute.^2)-(1.4452093e-
8).*(T_db_absolute.^3)+6.5459673.*log(T_db_absolute)); %in Pa valid for 0 to 200C
Pws=Pws_0./1000; % in kPa
Pw=phi./100.*Pws;
w=0.621945.*Pw./(P-Pw);

% h calculation from Tdb and w
h = (c_air.*Tdb+w.*(hlg+cw.*Tdb))./1000; %ASHRAE 2013 fundamentals eq. 32

end
```

Soil temperature (Kusuda model)

```
function T = Soil_T (D)

day = 1:365;
w = 5; % moisture content of soil - dry basis [%]ASHRAE
cp_wate = 4.182; %[kJ/kgK] %4.182ASHRAE
cp_soil = 0.73; %[kJ/kgK] %0.73ASHRAE
th_cond = 2.16; %[W/mK) %2.16ASHRAE
Density = 1600; %[Kg/m^3] %1600ASHRAE

th_diff = ((86.4*th_cond) / (Density*(cp_soil + cp_wate*(w/100))) ); %Thermal Diffusivity
[m^2/day]

tms = 12.225; %Mean Annual soil Surface Temp [ºC] %12.225
As = 15.5;  %Surface soil Temperature Amplitude [K] %15.5
step = 365; % year days
T_shift = 45; %phase lag of soil surface temperature [days] %45

%ASHRAE  chapter 33
T_depth = tms - As .* exp(-D.*(pi/step/th_diff)^0.5) * cos(((2*pi)/step)*(day-T_shift-
(D./2)*(step/pi/th_diff)^0.5));
S = kron(T_depth, [1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1,
1]); %day to hour conversion
T = S';

end
```

Appendix C - Electricity indexed tariffs

The electricity price at each hour is defined as equal to the price negotiated in Dutch EPEX spot market summed to the connection and transmission fees. Three price structures created were:

Reference Case – Simple tariff

Electricity price (h) = EPEX SPOT market NL price (h) + 0.0999 [€/kWh]

System 1 – Double tariff, with low power contract

- From 8:00h to 22:00h and weekends:

  Electricity price (h) = EPEX SPOT market NL price (h) + 0.1286 [€/kWh]

- From 22:00h to 8:00h:

  Electricity price (h) = EPEX SPOT market NL price (h) + 0.0409 [€/kWh]

System 2 – Double tariff, with medium power contract

- From 8:00h to 22:00h and weekends:

  Electricity price (h) = EPEX SPOT market NL price (h) + 0.0432 [€/kWh]

- From 22:00h to 8:00h:

  Electricity price (h) = EPEX SPOT market NL price (h) + 0.0216 [€/kWh]