Anabela Duarte Carvalho

HIGH EFFICIENCY GROUND SOURCE HEAT PUMP SYSTEMS FOR SUSTAINABLE BUILDING SPACE CONDITIONING

Tese de Doutoramento em Sistemas Sustentáveis de Energia, orientada pelo Professor Doutor Aníbal Traça de Almeida e apresentada à Faculdade de Ciências e Tecnologia da Universidade de Coimbra

Maio de 2015

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To my son

Gonçalo

ACKNOWLEDGMENTS

Undertaking this PhD thesis has been a truly life-changing experience for me and it would not have been possible to do without the support and guidance received from many people.

First and foremost, I would like to thank my supervisor, Professor Aníbal Traça de Almeida, for its guidance, encouragement, patience and wisdom during these past few years. Professor Traça supervision was outstanding throughout the PhD work, and for his methodological contributions, fruitful discussions, continuous motivation, friendship and for offering me all the necessary means to carry out this thesis, I am forever grateful. Professor Traça supervised me during my BSc, MSc and PhD studies. I owe him a great deal of my training as both student and researcher. It has indeed been a great privilege to work with such far-thinking and inspirational individual. All I have learned and accomplished during the course of this thesis would not have been possible without his exceptional dedication.

I also owe much to the Institute of Systems and Robotics (ISR) for having provided the excellent conditions and resources that have allowed me to accomplish my PhD work. I am also grateful to Instituto Superior de Engenharia de Coimbra (ISEC) for providing me with the conditions to conciliate, during the past five years, my lecturing activities with this PhD work. This work has been partially supported by the European project Ground-Med, from the FP7 under grant TREN/FP7EN/218895.

I am deeply thankful to Gilberto Vaz for his encouragement, fruitful discussions and invaluable advices on my research, which allowed me to move forward in some stages of the PhD study. I am also very grateful to André Quintino and Pedro Moura, who gave me a precious help.

I owe a great debt of gratitude to Pedro Miraldo and Carlos Rebelo for their friendship, support, continuous motivation and all the help they gave me in numerous ways.

I am very grateful to all my friends, colleagues and professors from DEEC and DEM of FCTUC, and I want to express a special gratitude for those with whom I have worked directly during my PhD work.

To my colleagues from DEM of ISEC whose help was decisive to conciliate the lecturing activities with the PhD work over the last few years. A big thank to Pedro Ferreira and Avelino Virgílio for their support and help on the mammoth task of doing the final formatting and printing of this thesis. To my special friend Ana Lopes for the support, encouragement and friendship during all these years. To Doctor José Manuel Pinto for the persistence and believe. My special thanks to Melandra's family for their friendship and all support with my son during this last year.

I also want to thank all users from 2nd floor of the ARH Centro (Administração da Região Hidrográfica do Centro) and CCDRC (Comissão de Coordenação Regional do Centro) for their sympathy and patience, in special to Eng. Emídio Cardoso for his indispensable assistance. I either thank Eng. Fernando Madeira for his collaboration and technical assistance whenever needed.

Finally, I would like to thank my family for their love, support and sacrifices, to whom I am deeply grateful; to my husband and my son, who endured my absence during my PhD work; to my father, who already has passed away, but left me a true example of life strength and courage, and to my mother, an example of willpower and humility, who gave me those values.

ABSTRACT

This thesis assesses the benefits of using ground source heat pump systems for space conditioning in buildings, in an integrated perspective, in terms of potential impacts such as increasing energy efficiency, increasing the use of renewable energy resources and reducing carbon emissions, as well as their use as a flexible load with thermal storage for balancing demand and supply, namely for the integration of intermittent renewable generation in the context of smart grids. Part of these impacts were evaluated based on the experimental results obtained in the ground source heat pump system installed in a public service building in Coimbra, under the European Project Ground-Med from the Seventh Research Framework Programme - FP7.

The high efficiency of an advanced ground source heat pump system, which integrates state of art components in an optimal manner, was assessed by calculating the seasonal performance factors based on the monitored data of the experimental ground source heat pump system. The results demonstrated the feasibility of high efficiency ground source heat pumps, reaching a seasonal performance of 5.4 for the heating season and 6.4 for the cooling season.

A control system was developed to allow programing the daily operation period of the overall system in order to guarantee temperature comfort when users arrive and to implement load control strategies. By the tests performed during summer and winter, it was concluded that the control strategies of adjusting the set point temperature of the ground source heat pump and the velocities of the circulation pumps, are very important to improve the system efficiency, namely during lower thermal needs periods.

The thermal response of the building was analyzed and the results showed that the building has a good thermal inertia that delays the building temperature variation, which allows the implementation of space heating load shifting strategies without creating major disturbances in the users thermal comfort.

Load shifting strategies by preheating the building and avoiding peak periods, to take advantage from the lower electricity prices and to minimize or even to avoid the energy consumption during the peak periods, were applied. The results showed that applying these strategies, profiting from the thermal mass of the building, significant costs savings can be achieved (about 34%).

Additionally, the impacts on the national electric diagram of applying the proposed load shifting strategies to other office buildings were also assessed, assuming that 25% of the total office buildings space heating load in Portugal can be controlled in a centralized way. It was concluded that preheating the buildings can contribute to a larger integration of the renewable electricity generation surplus. The implementation of Demand Response actions by switching off heating loads, during some periods of time, to compensate the variations and forecasting errors of wind

power, was also evaluated with positive impacts.

A fuel switching strategy for building space heating at the EU level, by a large scale penetration of high efficiency heat pumps in a scenario where natural gas boilers will be progressively phased out until 2050, was also proposed and assessed. It was found that in European Union in 2050, through the replacement of natural gas space heating by high efficiency heat pumps, around 60% of the primary energy required and 90% of the associated CO_2 emissions can be saved. Other positive impacts were achieved concerning decreasing the European natural gas dependency and increasing the renewable energy sources share in the total final energy consumption. A similar scenario was developed for Portugal and large positive impacts were also achieved.

In summary, high efficient heat pumps can have an important role towards exploiting the increasing penetration of renewable electricity generation, effectively contributing to the replacement of fossil fuels and to decrease the growing natural gas dependency of European Union from risky and unstable countries. Additionally, high efficient heat pumps, when combined with the building thermal storage capacity, can be used as a flexible load to balance supply and demand, employing load leveling and Demand Response strategies, and to significantly reduce operation costs at the consumer side.

Keywords: Ground source heat pump, energy efficiency, renewable energy, space conditioning, thermal storage, load management, fuel switching, CO_2 emissions

RESUMO

Esta tese avalia de uma forma integrada os vários benefícios da utilização de bombas de calor geotérmicas na climatização de edifícios. Diversos impactes foram avaliados em termos do seu potencial para o aumento da eficiência energética e da utilização das energias renováveis, para a redução das emissões de CO_2, bem como para a sua utilização como carga flexível, quando combinada com armazenamento de energia térmica, para melhorar o equilíbrio entre a oferta e a procura de energia, a partir da integração da geração intermitente de eletricidade renovável no contexto das redes inteligentes.

Parte desses impactes foram avaliados com base nos resultados obtidos numa instalação experimental de uma bomba de calor geotérmica que foi instalada num edifício de serviços, em Coimbra, no âmbito do Projeto Europeu Ground-Med do Sétimo Programa-Quadro de Investigação (7ºPQ).

A eficiência do sistema BCG, que integra componentes de última geração de uma forma otimizada, foi avaliada através do cálculo dos fatores de desempenho sazonal com base na monitorização dos dados do sistema BCG experimental instalado. Os resultados comprovam a alta eficiência destes sistemas BCG, atingindo um desempenho sazonal de 5.4 para a estação de aquecimento e de 6.4 para a de arrefecimento.

Foi desenvolvido um sistema de controlo que permite programar o período de funcionamento diário, para garantir o conforto térmico à chegada dos utilizadores ao edifício e que permite implementar estratégias de gestão de carga. Os testes realizados nas estações de aquecimento e de arrefecimento permitem concluir que estratégias de ajuste da temperatura de funcionamento da bomba de calor e da variação de velocidade das suas bombas de circulação são fundamentais para a melhoria da eficiência do sistema, particularmente nos períodos de menor necessidade térmica.

A resposta térmica do edifício foi analisada e os resultados evidenciam que o edifício tem uma boa inércia térmica, que atrasa a variação da temperatura interna em relação às condições ambientais, mesmo depois de desligado o sistema de climatização. Esta característica do edifício permite a implementação de estratégias de desvio de consumo associadas à sua climatização, sem perturbar significativamente o conforto térmico dos utilizadores.

Para minimizar ou mesmo evitar o consumo de energia elétrica durante as horas de ponta do diagrama de carga elétrico, foram aplicadas estratégias de desvio de consumo, pré-aquecendo o edifício ainda durante as horas de vazio para aproveitar os preços mais baixos da eletricidade. Os resultados mostraram que a aplicação destas estratégias, tirando partido da massa térmica do edifício, permite alcançar poupanças significativas (cerca de 34%).

Adicionalmente foi analisado o impacte no diagrama elétrico nacional da aplicação destas estratégias de desvio de consumo ao nível dos edifícios de escritórios portugueses, assumindo que em 25% destes edifícios os sistemas de aquecimento podem ser controlados de forma centralizada. Concluiu-se que o pré-aquecimento destes edifícios pode contribuir para uma maior integração do excesso de eletricidade renovável produzida durante a noite. Foi também avaliada a implementação de medidas de resposta da procura, através do desligar das cargas de aquecimento, durante alguns períodos de tempo, para compensar as variações e erros de previsão da energia eólica, tendo-se registado resultados positivos.

Foi proposto e avaliado o potencial da substituição progressiva em larga escala de caldeiras a gás natural por bombas de calor na climatização de edifícios, num horizonte até 2050 a nível de toda a União Europeia. Verificou-se que existe um potencial de poupança de energia primária de cerca de 60% e uma redução de cerca de 90% nas emissões de CO_2 associadas. A aplicação desta medida contribui ainda para uma menor dependência do gás natural importado e para o aumento da contribuição das energias renováveis no consumo final de energia a nível da União Europeia. Foi aplicado um cenário similar para o caso particular de Portugal e também se obtiveram resultados muito positivos.

Em resumo, as bombas de calor de alta eficiência podem ter um papel muito importante no aumento da penetração de energias de origem renovável, podendo ter um contribuindo efetivo na substituição dos combustíveis fosseis e na diminuição da importação do gás natural proveniente de países politicamente instáveis a nível da UE. Adicionalmente, as bombas de calor de alta eficiência, quando combinadas com a massa térmica de edifícios com elevada capacidade térmica, podem ser usadas como uma carga flexível na gestão do diagrama de carga elétrico, a partir de estratégias de gestão de cargas e de resposta da procura, e também para reduzir significativamente os custos de operação do aquecimento para o consumidor final.

Palavras chave: Bombas de calor geotérmicas, eficiência energética, energias renováveis, climatização, armazenamento de energia térmica, gestão do diagrama de carga, substituição de combustível, emissões de CO_2

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List of Acronyms

ASHP	Air Source Heat Pump
BHE	Borehole Heat Exchanger
COP	Coefficient Of Performance
CP	Circulation Pump
CPI	Current Policy Initiatives
DAQ	Data Acquisition
DPF	Daily Performance Factor
DR	Demand Response
DSC	Differential Scanning Calorimeter
DSM	Demand Side Management
EC	European Commission
EE	Energy Efficiency
EER	Energy Efficiency Ratio
EHPA	European Heat Pump Association
EPBD	Energy Performance of Buildings Directive
EU	European Union
FC	Fancoil
GHE	Ground Heat Exchanger
GHG	Greenhouse Gases
GSHP	Ground Source Heat Pump
HP	Heat Pump
HTF	Heat Transfer Fluid
HVAC	Heating, Ventilation and Air Conditioning
HW	Hot Water
LHS	Latent Heat Storage
LPG	Liquefied Propane Gas
NEEAPs	National Energy Efficiency Action Plans

NG	Natural Gas
PCM	Phase Change Material
PPGS	Portuguese Platform for Shallow Geothermal
RES	Renewable Energy Sources
SCOP	Seasonal Coefficient Of Performance
SEER	Seasonal Energy Efficiency Ratio
SH	Space Heating
SHS	Sensible Heat Storage
SPF	Seasonal Performance Factor
TCM	Thermochemical Material
TES	Thermal Energy Storage
UK	United Kingdom
USA	United States of America
VRF	Variable Refrigerant Flow
WSHP	Water Source Heat Pump

LIST OF SYMBOLS

C	Heat capacity [kJ/°C]
c	Specific heat [J/kg°C))
CO_2eq	CO_2 equivalent
deltaT	Temperature difference
E_{RES}	Supplied renewable energy
m	Mass [kg]
P	Electrical power
Q	Thermal power supplied to the building
Q_H	Heat released at high temperature
Q_L	Heat absorbed at low temperature
Q_p	Rate of heat input (plant thermal power) [kW]
Q_{usable}	Total usable heat delivered by heat pumps
S_{th}	Steady state heat loss from the building if the plant is kept on overnight
T	Temperature [°C]
t	Time
Tavg	Average temperature
T_H	High temperature
T_i	Indoor temperature [°C]
T_{i2}	Theoretical building temperature [°C]
T_L	Low temperature
Tmax	Maximum temperature
Tmin	Minimum temperature
T_o	Outdoor temperature [°C]
t_{unocc}	Building unoccupied period
U'	Building heat loss coefficient [kW/°C]
v	Volume [m^3]
$W_{net,in}$	Net work input

η	Electricity generation efficiency
η_s	Seasonal space heating energy efficiency factor
ρ	Density [kg/m^3]
τ	Time constant for the cooling phase

CHAPTER 1

INTRODUCTION

The fossil fuels dependency from external countries and the negative environmental impacts from fossil fuel combustion have been during the last decades a growing concern of the European Union (EU). Several targets have been defined at EU level to reduce these problems, both on the supply and demand side, with the purpose to significantly increase the Renewable Energy Sources (RES) contribution and the energy efficiency of the economy. In addition to the targets defined for the year 2020, known as the 20-20-20 targets, the EU has already established new targets to be achieved by 2030, namely: a 40% reduction in Greenhouse Gases (GHG) emissions compared to 1990, a RES share of 27% in the final energy consumption, and a minimum energy efficiency increase of 27%, to be reviewed by 2020. These intermediate targets for 2030 were defined in a way to be possible to decrease the carbon emissions by 80% in relation to 1990 levels until 2050 (European Commission, 2015a).

Concerning the 20-20-20 targets, at time, the climate and energy package 2009 did not address directly the Energy Efficiency (EE) target, which is a non-binding target, contrary to the RES share and GHG emissions reduction targets (EEA, 2014). For the two last ones, mandatory targets for each Member State were defined, but no national targets were defined for the EE target, leading to a slowly improvement of the energy efficiency, mainly at the buildings level.

Energy efficiency is one of the most cost effective ways to enhance security of energy supply, and to reduce emissions of GHG and it can be seen as the Europe's biggest energy resource. However, in 2011, the European Commission (EC) forecasts suggested that, with the energy efficiency measures in place, the EU was on course to only achieve half of the 20% energy savings target for 2020, despite the large savings potential, mainly in the buildings sector as presented in Figure 1.1 (European Commission, 2011a).

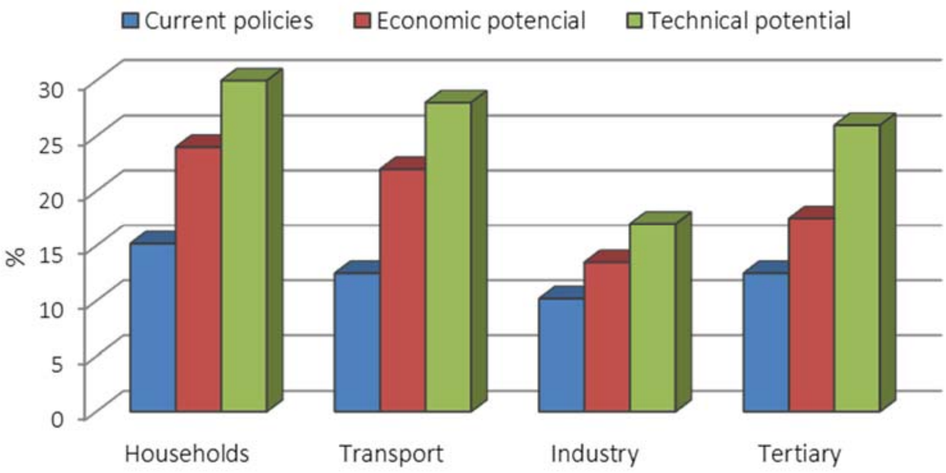


Figure 1.1: Final energy savings potential in EU 27 in 2020, as percentage of the projections done in 2007 (European Commission, 2011a).

To reduce this gap by exploiting the considerable potential for higher energy savings in buildings and transportation, the EC reviewed the Energy Efficiency Plan in 2011. Key energy efficiency legislation was also reviewed such as, the Ecodesign Directive (EU, 2009a) (which state minimum energy efficiency standards for all energy-related products), the Energy Labelling Directive (EU, 2010a) (concerning standard information about the energy consumption of the energy-related products) and the Energy Performance of Buildings Directive known as the EPBD Directive (EU, 2010b) (which defines mandatory energy performance certificates for the sale or rental of buildings, minimum energy performance requirements for new buildings, inspection schemes for heating and air conditioning systems, etc.).

Another important development was the adoption of the Energy Efficiency Directive (EU, 2012) in 2012. Under the Energy Efficiency Directive, the Member States must set indicative national targets of primary energy savings for 2020, defined in the National Energy Efficiency Action Plans (NEEAPs) every three years, and provide annual progress reports, as well as implement a set of mandatory requirements (being the most significant the energy obligation schemes for utilities - 1.5% of energy savings by year) (EEA, 2014). Concerning buildings, the Energy Efficiency Directive imposes that, EU countries must:

- establish long-term national building renovation strategies for public and private buildings;
- make renovations to at least 3% of buildings owned and occupied by central government (double of the current rate);
- governments should only purchase buildings which are highly energy efficient.

Currently, and according to the Energy Efficiency Communication of July 2014 (European Commission, 2014a), the EU can reach energy savings target of 20% in 2020, if the EU countries implement all the existing legislation on energy efficiency. Concerning the RES targets and GHG emissions targets, projections indicates that they will be over-achieved by 2020. Emissions are projected to be 21% lower in 2020 than in 1990 (European Commission, 2014b) and the RES share is estimated to be 20.6% in 2020, if the National Renewable Energy Action Plans (NREAPs) of 2010 are fulfilled (EEA, 2014).

For the global RES share, in 2012, in the EU, the main renewable technologies contribution was solid biomass mainly used for heating applications (43%), followed by hydro power (18%) and wind power (10%) used in electricity generation (Figure 1.2). For 2020, based on the NREAPs, the three main renewable technologies will be the same, although with a reduced individual contribution for solid biomass (33%), mainly for heating and hydro power (13%), but with a higher contribution for wind power (17%) (EEA, 2014).

In 2009, the Directive 2009/28/EC of the European Parliament and of the Council, on the promotion of the use of energy from renewable sources (EU, 2009b), has officially recognized that Heat Pumps (HPs) use renewable energy sources and that its contribution should be also considered in the RES share. As it can be noticed in Figure 1.2, in 2012, heat pumps represented 4.3% of the total renewable energy consumed in EU.

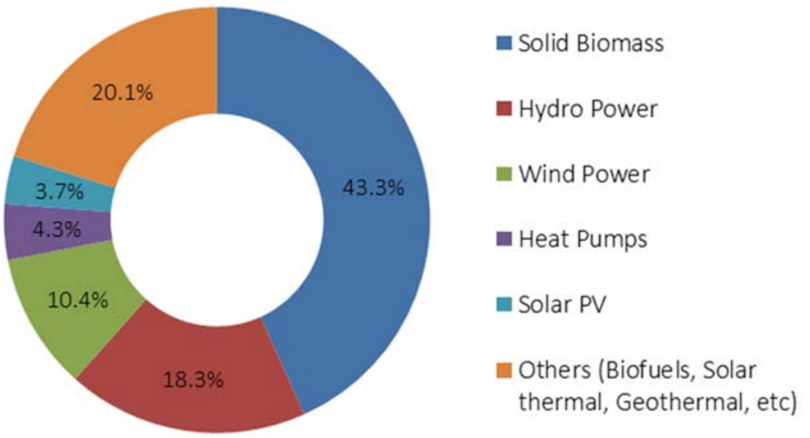


Figure 1.2: Breakdown by RES technologies for total RES in EU-28, 2012 (EEA, 2014).

The increasing penetration of renewable energy in the electricity sector has had a major contribution towards the achievement of EU targets, and it is expected to increase much more until 2050. According to the European Commission, in order to be possible to meet the long-term goal of reducing Europe's GHG emissions by 80% to 95% below 1990 levels by 2050, the EU's energy sector will also need to undergo a rapid decarbonization. In the energy sector the global share of renewables must reach between 55% and 75% by 2050, with a specific contribution of 86% in the power generation sector (European Commission, 2011b).

One consequence of the large scale use of renewable generation is the high penetration levels of intermittent energy resources. Most of the energy generation by renewable resources (wind and solar) is dependent on weather conditions and presents a high variability, being its generation profile very different from the energy demand profile (Moura and De Almeida, 2010a). This poses a significant challenge associated with balancing of demand and supply. In 2012, at EU level, wind power represented 25% of the total renewable electricity generation and it is expected to increase to 40% in 2020 (Eurostat, 2015). The future of intermittent energy resources rests on successfully increasing energy system flexibility. The modernization of power grids, the development of cost-effective load balancing and energy storage options and the control of the demand through Demand Response (DR) programs have been on the top of the agenda of the European research with a large number of Smart Grid Projects (Covrig et al., 2014).

1.1 - MOTIVATION

Member states have to deal with several challenges associated to energy and climate issues. The buildings sector, because it represents about 40% of the final energy consumption and 36% of the total GHG emissions in EU (European Commission, 2015b), as well as due to its high savings potential, is considered one of the key sectors to reach the EU targets. Besides the need to significantly improve the building envelope thermal efficiency (both for new and existing buildings), it is also necessary to increase the use of renewable energies and to invest in the use of high efficiency final appliances. Furthermore, there is a need to increase the electric grid flexibility which can be ensured for instance, by introducing decentralized energy storage systems and by increasing demand elasticity (shifting or curtailment of loads) using DR.

Within the buildings sector, at the EU level, space heating is the end use that presents the highest energy consumption. Moreover, in Europe, space heating accounted for 50% of the final energy consumption in total buildings (residential + services) in 2010 (Connolly et al., 2013). Two-thirds of the supplied heat is generated with fossil fuels and is mostly dominated by natural gas (NG) boilers (Connolly et al., 2013). These circumstances create a huge opportunity for replacing fossil fuels and conventional space heating technologies by renewable energy resources and high efficient technologies.

With the electricity generation becoming less and less polluting, the current situation, in which there is a major reason for a strong investment in the energy efficiency level in buildings, including in their energy end-use equipment, and to decrease significantly the use of fossil fuels, is a good occasion for promoting high efficient technologies which also use RES for space conditioning, like electric heat pumps.

Several studies have presented heat pumps as a very promising technology to contribute to the building sector decarbonization and to decrease the primary energy consumption for the heat demand in the EU countries (Morrone et al., 2014; Sarbu and Sebarchievici, 2014; Self et al., 2013; Ecofys, 2013a; Bayer et al., 2012). Simultaneously, electricity generation is becoming a cleaner form of energy, due to the massive increase of renewable energies observed in EU in the last decade, leading to lower fossil fuels consumption in electricity generation and to progressively lower carbon content in electricity (Eurelectric, 2012).

Besides demonstrating the potential of heat pumps, namely of Ground Source Heat Pumps (GSHPs), for primary energy and carbon savings, it is also necessary to demonstrate their potential as flexible loads to balancing demand and supply and for the integration of intermittent renewable electricity, when combined with thermal storage.

Although GSHP technology and market are developed in countries of Central Europe, the corresponding market in South Europe is at early developing stages. Some European Projects, such as Ground-Reach, Groundhit and more recently the Ground-Med Project (Ground-Med, 2014), have as main purpose to demonstrate the feasibility of this technology with high performance levels so as to disseminate its implementation in Southern European countries. For example, in Portugal, this technology is almost inexistent and almost unknown. Only recently was created the Portuguese Platform for Shallow Geothermal (PPGS), in order to increase the knowledge in the shallow geothermal area and to promote its proper dissemination and use.

Portugal is even more dependent on imported fossil fuels than the EU and there are sustainable renewable energy resources widely available. Portugal energy dependence has been decreasing along the years, reaching 75% in 2012, due to the increasing share of RES (between 2004 and 2012 the RES share increased from 20% to 25%) (Eurostat, 2015). The main contribution has been from the electricity generation. In the last 3 years, due to very raining winters, about 50% of the electricity was generated by RES. In 2020 it is expected to achieve a RES share of 31% in final energy consumption and a RES share of 60% in electricity generation with a major contribution from wind and hydro energy. Wind power already represents 26% of the total installed power and ensured 24% of the generation in 2014 (REN, 2015). During the night, in rainy (and also windy) winters, the electricity generation from renewable sources quite often exceeds the electricity demand in Portugal. There is a need to study new solutions for space conditioning in buildings that, besides being more efficient and less pollutant, also helps to balance demand and supply and to integrate the intermittent power generation.

In the particular case of Portugal, due to its moderate climate conditions, space heating is not so significant, but together with space cooling, they already represent one of the most significant end-uses energy consumption. Currently and in the coming years a deep intervention in building sector is required, which is a great opportunity for the installation of high efficiency space conditioning systems, like GSHPs. Additionally, HPs or GSHPs when combined with thermal storage can be a sustainable answer to integrate energy generation.

1.2 - RESEARCH FRAMEWORK

Part of the research work of this thesis was developed based on the GSHP prototype that was installed in a Pilot Building in the city of Coimbra (Portugal) under the European Project Ground-Med (Advanced ground source heat pump systems for heating and cooling in Mediterranean climate) (Ground-Med, 2014) from the Seventh Research Framework Programme - FP7. This project aims to demonstrate a seasonal performance factor greater than five for GSHPs installed in eight pilot buildings in Southern Europe. The eight GSHP demonstration systems are located in (Figure 1.3):

- France - CIAT subsidiary office building in Septemes les Vallons.
- Romania - University of Oradea, Department of Visual Arts.
- Portugal - Public service building in Coimbra.
- Slovenia - Municipal hall in Benedikt.
- Spain - University Polytechnic of Valencia.
- Spain - Information center for renewable energies in Barcelona.
- Italy - HIREF factory building in Padova.
- Greece - Head office building of EDRASIS in Athens.

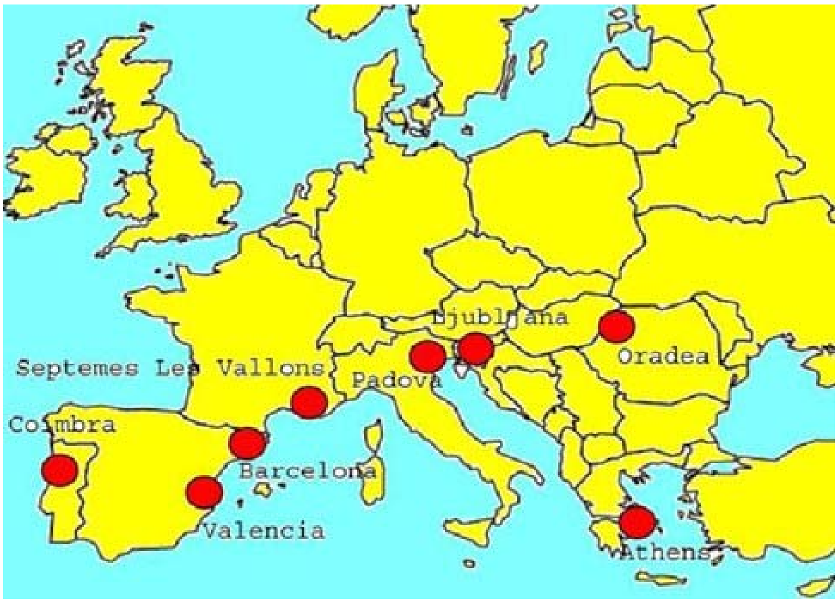


Figure 1.3: Localization of the 8 demo sites of the Ground-Med Project.

The Ground-Med consortium consists of 24 European organizations including a wide diversity of GSHP actors, such as research institutes, technical universities, heat pump manufacturers, national and European industrial associations, energy consultants, works contractors and an information centre. The Project started in January 2009 and finished in December 2014 (duration of 6 years). More information about the project and the demo sites is available at the Ground-Med website (http://www.groundmed.eu).

1.3 - MAIN GOALS

The main goals pursued by the work developed in the scope of this thesis are:

- to assess and optimize the performance of a GSHP pilot system installed in a pilot building in Coimbra, in winter and summer seasons, taking into account the main parameters that influence its performance;
- to develop and assess control strategies applied to the GSHP system in order to save electricity costs at the consumer side using the thermal mass of the building as a storage medium;
- to evaluate the potential of those control strategies for better management of the load profile and for integration of intermittent RES if applied in large scale in national office buildings;

- to assess the impacts of the large scale penetration of high efficiency HPs in a scenario where NG space heating will be progressively phased out in EU and in Portugal until 2050, in the following aspects: decrease in carbon emissions, decrease in primary energy consumption, decrease of NG dependency and increase of the RES share contribution.

Initially it was planned to couple a thermal storage tank to the GSHP system based on Phase Change Materials (PCMs), but due to external reasons it was not installed. Therefore, part of this thesis work aimed also to evaluate and select the most suitable PCM technology to be applied in the test installation.

1.4 - MAIN CONTRIBUTIONS

The main contribution of this work is the assessment and optimization of GSHP systems to provide sustainable space conditioning in buildings. The benefits of GSHP systems were evaluated in terms of potential impacts such as increasing energy efficiency in Buildings, increasing the use of RES, reducing carbon emissions, as well as their use as a flexible load with thermal storage for balancing demand and supply and for integration of intermittent renewable energies.

Aiming to address current and future energy challenges by assessing the different impacts of the GSHP technology, the specific contributions developed in this thesis are the following:

- Assessment and optimization of the high Seasonal Performance Factor (SPF) of the GSHP and of the overall system installed in a public service building in Coimbra, with 30 users.

  Results of SPFs for two winter seasons and one summer season are presented. They were calculated based on the electric energy and thermal energy consumption monitored during the entire seasons, after guarantying that the system was working properly. Indoor temperatures of five different rooms were monitored in order to adapt the GSHP system operation to improve user's satisfaction.
- Development of the GSHP control system with the support of an advanced monitoring system.

  The main purpose of this system was to program the period operation of the GSHP system for each day of the week and to vary GSHP set points and circulation pumps velocities remotely to perform the experimental tests.
- Evaluation of key parameters for the improvement of the overall system efficiency,

like the circulation pumps velocities and the water temperature supplied to the building.
For each season the best combination of the circulation pumps velocities (at the condenser side and the evaporator side) was determined, which maximizes the GSHP system performance.

- Analysis of the building thermal response in order to be possible to implement space heating shifting strategies without creating major disturbances in user´s thermal comfort.
  To determine the necessary preheating time, guarantying that at a desired time the building temperature was already stabilized, a model based on the lumped capacitance method was used (the building is considered as an equivalent homogeneous body). The key parameters of the building (time constant of the building cooling phase; building heat loss coefficient; and building thermal capacity) were determined to be used in the model. These parameters were calculated based on monitored data of the outdoor temperature, indoor temperature and thermal power consumption. This is a simple method that can be easily implemented in other buildings.
- Analysis of load shifting strategies for space heating and assessment in terms of the savings costs at the consumer side and in terms of integration of intermittent renewable energies.
  These strategies consist in preheating the building during lower electricity rates periods and shutting down the GSHP system during the peak periods prices, using the building thermal mass as a storage medium. One of the main purposes of applying these strategies was to enhance the use of an existent thermal storage medium in buildings with medium and high thermal inertia. Although active Thermal Energy Storage (TES) systems can increase the space conditioning flexibility, the existent thermal capacity of the building can also be used, without significant investment. Based in the results achieved in this particular building, an extrapolation of the application of these strategies to the office buildings at national level was made to assess its impact on the electrical grid. The impact of preheating the office buildings as a way to integrate the renewable generation surplus was studied, as well as the use of office buildings space heating as a flexible load to reduce the mismatch between the forecasted and real wind power generation.
- Assessment of the impacts of a large scale penetration in EU of high efficiency HPs in a scenario where NG boilers (main space heating technology) will be progressively phased out until 2050.
  Besides the assessment of the primary energy and carbon emissions savings (which are more common in other studies) the contribution to increase of RES share and to decrease NG dependency in 2050 was also carried out.

A similar scenario was developed for Portugal and the same impacts were also assessed. In this case the full replacement of the oil products and NG until 2050 was considered. The results achieved in the experimental GSHP installation were used and a smaller carbon emission factor was considered.

- Analysis of thermal storage technologies to be coupled with GSHP systems.
  The most suitable PCM technology to be used in a thermal storage tank coupled to the GSHP test installation was selected, based in the main criteria factors, and a possible thermal storage capacity was also estimated which could be used as reference for future installations.

1.5 - THESIS OUTLINE

The remainder of this thesis is organized as follows:

Chapter 2 - State-of-the-art on heat pump systems and thermal energy storage applications

This chapter provides a background on the role of HPs for sustainable building space conditioning presented in other studies, namely for carbon and energy savings, as well as for improve grid management. Studies made so far do not address the cross benefits of HPs in an integrate perspective. The HP market status in Europe is also presented as well as the typical efficiencies found in literature and in recent commercial products available by leading companies. Finally, a background on TES technologies and main applications is also addressed.

Chapter 3 - Experimental GSHP installation in a service building

A description of the pilot building and of the overall GSHP system is presented. The main components of the hydraulic system are described (borehole heat exchangers, circulation pumps, fancoils and heat pump). The operating principle of all the system is presented as well as the advanced monitoring system and the control system. A possible TES tank to be installed is also presented.

Chapter 4 - Experimental results and performance analysis of the GSHP system

The GSHP system experimental results are presented in this chapter. The seasonal performance factors are calculated based on the thermal energy consumption and electric energy consumption. Typical daily loads profile of the GSHP and other components, and temperatures evolution are presented. The results achieved in the improvement of the global efficiency with the adjustment of the circulation pumps velocity and of the water supply temperature are also presented.

Chapter 5 - GSHPs combined with building thermal mass for electric load management

An integrated approach using load shifting strategies applied to the GSHP system is presented and assessed, in terms of costs reduction to the user and in terms of the improvement on the electrical grid management, through the contribution to the integration of intermittent renewable energy generation.

Part of this chapter is presented in the paper entitled "Ground source heat pumps as high efficient solutions for building space conditioning and for integration in smart grids", which was submitted to the Energy Conversion and Management journal (Carvalho et al., 2015a).

Chapter 6 - GSHP carbon emissions and primary energy reduction potential for buildings heating in EU and in Portugal

The assessment of impacts of the large scale penetration of high efficiency HPs to replace NG boilers, used in space heating in EU, and the corresponding implementation scenario until 2050 are presented in this chapter. The impacts are evaluated in terms of: primary energy and CO_2 emissions savings, decrease of NG dependency and increase of the RES share contribution. The methodology used in the analysis is described along this chapter.

Part of this chapter is presented in the paper "Ground source heat pump carbon emissions and primary energy reduction potential for heating in buildings in Europe – results of a case study in Portugal", published online in the Renewable and Sustainable Energy Reviews journal (Carvalho et al., 2015b).

Chapter 7 – Conclusions and recommendations for future work

This chapter summarizes the main achievements of the thesis and gives some suggestions for future research work.

CHAPTER 2

STATE-OF-THE-ART ON HEAT PUMP SYSTEMS AND THERMAL ENERGY STORAGE APPLICATIONS

This chapter starts by presenting some introductory concepts and definitions referred along this thesis. A review of the state-of-the-art is done about heat pumps and thermal storage technologies and its main applications in buildings. Heat Pumps (HPs) are classified according the heat source used. An overview about the heat pump market at EU level is presented, as well as the typical efficiencies and the new standards for heat pumps performance and energy labelling. The method to calculate the Renewable Energy Sources (RES) contribution of heat pumps is also presented. Concerning Thermal Energy Storage (TES), different methods and materials are considered. More emphasis is given to the Phase Change Materials (PCMs), since, at the beginning of this work, it was planned to install a storage tank in the building where the experimental study was carried out. Some studies related to primary energy and emissions savings by using HPs instead of conventional technologies for space heating are presented, as well as case studies where HPs are combined with TES to reduce energy bills by load shifting and to integrate intermittent renewable energy.

2.1 - INTRODUCTORY CONCEPTS

Heat Pump (HP)

A heat pump is a device that absorbs heat from a relatively low temperature heat source and releases it to a warmer medium. Heat pumps are designed to move thermal energy in the opposite direction of spontaneous heat flow by transferring heat from a low temperature to a higher temperature medium. The external power (e.g. electric) used by the heat pump is to accomplish the work of transferring energy from the cold medium to the warm medium and not to generate directly heating or cooling. The heat pumps classification depends on the used heat source. The most common are those that use air as a heat source, called Air Source Heat Pumps (ASHPs), and those that use the soil or ground as the heat source, called Ground Source Heat Pumps (GSHPs).

Thermal Energy Storage (TES)

Thermal energy storage consists in storing thermal energy in some materials during a certain period, which can be hours, days, weeks, months or seasons, to be released when necessary. Thermal energy can be stored as a change in internal energy of a material as sensible heat, latent heat, thermochemical reaction, or a combination of these (Sharma et al., 2009).

Sensible Heat Storage (SHS)

Thermal energy is stored by raising the temperature of a solid or liquid. The sensible heat storage system utilizes the heat capacity and the change in temperature of the material during the process of charging and discharging. The amount of heat stored depends on the specific heat of the medium, the temperature change and the amount of storage material (Sharma et al., 2009).

Latent Heat Storage (LHS)

Latent heat storage is based on the heat absorbed or released when a material undergoes a phase change from one physical state to another. The phase change can occur in the following forms: solid–solid, solid–liquid, solid–gas, liquid–gas and vice-versa. The materials used in this type of storage are called phase change materials. Solid–liquid PCMs are more common and have benefited from many developments during the past two decades, although the amount of heat involved during their phase change is smaller than that of solid–gas or liquid–gas PCMs

(Tatsidjodoung et al., 2013).

Thermochemical storage

Thermochemical storage consists of using thermal energy to excite a reversible chemical reaction, which can be recovered by reversing the reaction. The materials used in this type of reaction are known as Thermochemical Materials (TCMs) (Tatsidjodoung et al., 2013).

Passive heat storage

Passive storage can be described as storage with heat transfer on the storage surface by free convection and radiation with no actively moved heat transfer fluid. The charging and discharging of passive storage systems depends only on the temperature difference between the storage medium and the surroundings (Mehling and Cabeza, 2008). In order to clarify this concept, it is considered that even if an active system is used to increase the indoor air temperature, the heat stored in walls by the heat transferred from indoor air to the walls is a passive storage.

Active heat storage

In opposite, an active storage system exists when the charging and/or discharging occurs with active help from pumps or fans to move the heat transfer fluid and the charging and discharging can be controlled. In this case, the storage in building walls is considered active if there is some heat transfer fluid (air or water) circulating inside the walls through some kind of pipes or cavities, purposely.

Thermal mass or heat capacity

Thermal mass is equivalent to thermal capacitance or heat capacity, which is the ability of a body to store thermal energy. Thermal mass is a property of the mass of a building which enables it to store heat, providing "inertia" against temperature fluctuations. Quite often is referred as the thermal inertia of building.

Demand Side Management (DSM)

“Demand side management (DSM) is 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, i.e., changes in the time pattern and magnitude of a utility’s load” (Gellings, 1985). Examples of DSM strategies are presented in Figure 2.1.

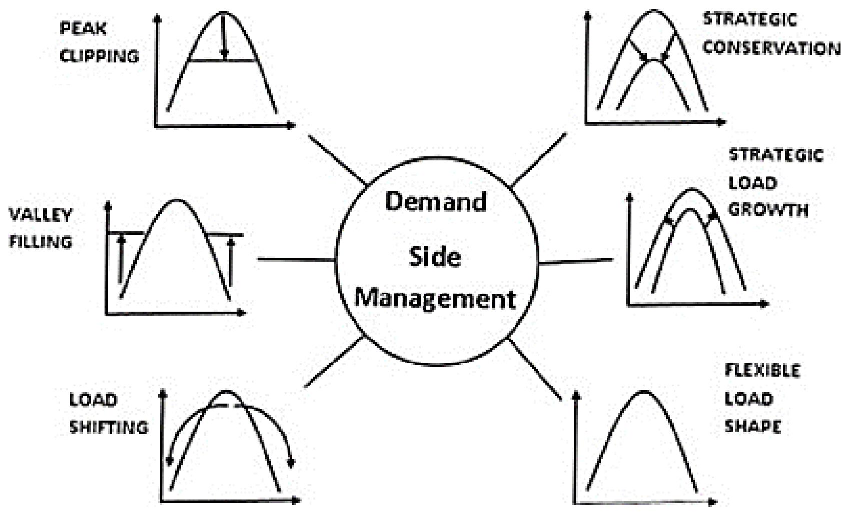


Figure 2.1: Demand side management strategies (Gellings, 1985).

Load shifting is a major DSM strategy that consists in shift the loads from periods with higher energy prices to other periods where energy costs are lower. Electric heating and cooling systems are typically used to charge thermal storage mediums during lower electricity prices periods, and during peak prices periods the heating or cooling demand are totally or partially satisfied by the thermal storage systems.

Demand Response (DR)

According to the Federal Energy Regulation Commission (FERC, 2014), Demand Response is defined as "Changes in electric usage by demand-side resources from their normal consumption patterns in response to changes in the price of electricity over time, or to incentive payments designed to induce lower electricity use at times of high wholesale market prices or when system reliability is jeopardized". In a more simple definition, the Demand Response concept refers to changes in electricity consumption in response to supply conditions (Balijepalli et al., 2011).

Smart grid concept

The Smart Grids European Technology Platform has defined smart grid as "an electricity network that can intelligently integrate the actions of all users connected to it – generators, consumers and those that do both – in order to efficiently deliver sustainable, economic and secure electricity supplies". Integrating smart grid technologies such as advanced information, sensing, communications, control and energy technologies, and systems, can significantly improve electricity network reliability while at the same time enable demand side management (IEA, 2013a).

2.2 - HEAT PUMPS TECHNOLOGIES

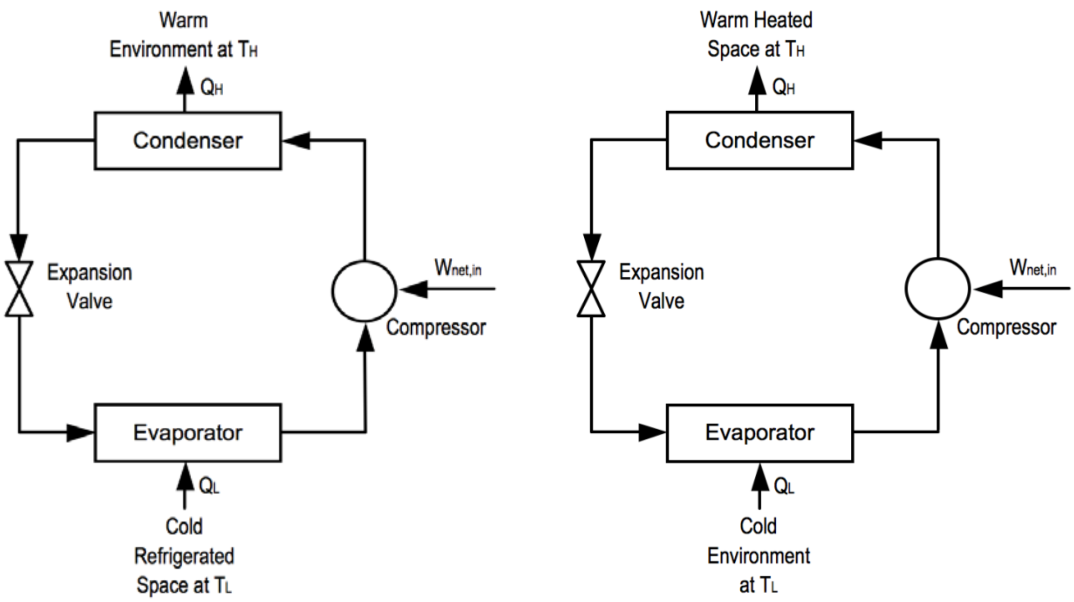
2.2.1 - HEAT PUMPS OPERATING PRINCIPLE

This subsection includes a brief thermodynamic review of the concepts of refrigerators and heat pumps (Cengel and Boles, 2004).

In nature, heat transfer occurs from high-temperature mediums to low temperature ones without requiring any device. The inverse, which is transfer heat from a low temperature medium to a high temperature medium, requires special devices operating in a thermodynamic cycle called refrigeration. To accomplish this energy transfer, the refrigerator receives external energy in the form of work or heat from the surroundings. The refrigeration cycle most frequently used is the vapor-compression refrigeration cycle (Figure 2.5) which involves four main components: a compressor, a condenser, an expansion valve and an evaporator.

In the vapor-compression refrigeration cycle, the refrigerant enters in the compressor as a gas and is compressed to the condenser pressure. It leaves the compressor at a relatively high temperature and cools down and condenses as it flows through the coils of the condenser by rejecting heat to the surrounding medium. Then the refrigerant enters into a capillary tube or an expansion valve where its pressure and temperature drop drastically due to the throttling effect. The low-temperature refrigerant then enters into the evaporator, where it evaporates by absorbing heat from the refrigerated space. The cycle is completed as the refrigerant leaves the evaporator and reenters in the compressor.

Refrigerators and heat pumps operate on the same cycle, but differ in their objectives. The purpose of a refrigerator is to maintain the refrigerated space at a low temperature by removing heat from it. Discharging this heat to a higher-temperature medium is merely a necessary part of the operation, not the purpose. In opposite, the objective of a heat pump is to maintain a heated space at a high temperature. This is accomplished by absorbing heat from a low-temperature source, such as water, soil, or cold outside air in winter, and supplying this heat to the high-temperature medium such as a building. Heat pumps, if reversible, can also work as a refrigerator or as an air conditioner, removing the heat from a room or a building during the summer season and release it to the outside environment. Figure 2.5 illustrates a refrigerator (a) and a heat pump (b) operating in a thermodynamic cycle.



a) b)

Figure 2.2: Vapor-compression refrigeration cycle: a) Refrigerator, b) Heat pump.

The performance of a refrigerator or heat pump is expressed in terms of the Coefficient Of Performance (COP), which is the ratio of the desired result to the required input. For the refrigerator, the desired result is the heat supplied (Q_L) at low temperature (T_L) and the input is the net work ($W_{net,in}$) into the device to make the cycle operate. The COP is obtained by the equation:

$$COP_R = \frac{Q_L}{W_{net,in}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{\frac{Q_H}{Q_L} - 1} \tag{2.1}$$

where Q_L is the heat absorbed at low temperature (T_L) and Q_H is the heat released at high temperature (T_H). In cooling mode, the heat is absorbed at low temperature (T_L) from inside rooms and released at high temperature (T_H) to the outside. In the heating mode the opposite happens: the heat is absorbed at low temperature (T_L) from outside air and released at high temperature (T_H) into rooms.

For the device acting like a heat pump, the primary function of the device is to transfer heat (Q_H) to the high temperature medium (T_H). The coefficient of performance for a heat pump is given by Equation (2.2).

$$COP_{HP} = \frac{Q_H}{W_{net,in}} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - \frac{Q_L}{Q_H}} \tag{2.2}$$

Under the same operating conditions the COP_{HP} and COP_R are related by Equation (2.3).

$$COP_{HP} = COP_R + 1 \tag{2.3}$$

It is important to note that the most efficient cycles are reversible, that is, cycles that consist entirely of reversible processes. A reversible process is defined as a process that can be reversed without leaving any trace on the surroundings. Reversible cycles cannot be achieved in practice because the irreversibilities associated to each process cannot be eliminated, but they provide upper limits on the performance of real cycles. The best known reversible cycle is the Carnot cycle. This ideal cycle is composed of four reversible processes, two isothermal and two adiabatic. An ideal refrigerator or a heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator, or a Carnot heat pump.

According to the second law of thermodynamics, for ideal reversible refrigerators or heat pumps, such as a Carnot refrigerator or heat pump, the heat transfer ratios in Equations (2.1) and (2.2) can be replaced by the respective ratio of the absolute temperatures of the two reservoirs. Therefore, the $COP_{R,rev}$ and $COP_{HP,rev}$ can be calculated directly by the temperatures of the reservoirs, as represented in Equations (2.4) and (2.5), respectively:

$$COP_{R,rev} = \frac{1}{\frac{T_H}{T_L} - 1} \tag{2.4}$$

$$COP_{HP,rev} = \frac{1}{1 - \frac{T_L}{T_H}} \tag{2.5}$$

These are the maximum possible COPs for a refrigerator or a heat pump operating between the temperature limits of T_H and T_L. Actual refrigerators or heat pumps may approach these values as their designs are improved, but they can never reach them. From Equations (2.4) and (2.5) it is clear that the COPs of both the refrigerators and the heat pumps decrease as T_L decreases. Also, the COPs decrease as the reservoirs temperature difference increases.

Finally, the performance of refrigerators and air conditioners is typically expressed by the Energy Efficiency Ratio (EER), which is equal to the COP_R. In some countries like in United States, the EER is calculated by the ratio of the amount of heat removed from the cooled space in British Thermal Units (BTU) over the electricity consumed in Watt-hour (Wh). In this case, the relation between EER and COP is given by Equation (2.6) (1 Wh = 3600 J and 1 BTU = 1055.0558526 J).

$$EER = 3.412\,COP_R \tag{2.6}$$

2.2.2 - CLASSIFICATION OF HEAT PUMPS

Electrically driven heat pumps move renewable thermal energy from a low temperature medium, such as the air, ground or water, to a higher temperature medium, which can be used for space or water heating. Heat pump systems can be classified according to the heat source and the Heat Transfer Fluid (HTF) used for energy distribution, as presented below:

- Air Source Heat Pumps (ASHP), when the heat source is the outside air. ASHP are divided in air-to-air HPs and air-to-water HPs, according the heat transfer fluid used for energy distribution (air or water). Air-to-air heat pumps are the most common and are particularly suitable for factory-built unitary heat pumps (mono-split).
- Water Source Heat Pumps (WSHP), when heat pumps rely on water as the heat source. The water can be ground-water extracted from wells, lakes, ponds or stream, or water from solar collectors. When they use ground-water they are also classified as ground source heat pumps. They are also divided in two groups according to the HTF used for energy distribution. Water-to-air HPs use air to transmit heat to or from the conditioned space. Water-to-water HPs use water as the heat source and sink for heating and cooling. Heating/cooling changeover can be done in the refrigerant circuit or in the water circuits.
- Ground Source Heat Pumps (GSHP), when heat pumps use the ground as a heat source and sink. However, the classification GSHP is typically used when the HTF fluid circulates in pipes embedded in the ground, called Ground Heat Exchangers (GHE) or Borehole Heat Exchangers (BHE). The most common system is the refrigerant-to-water heat exchangers system, where water or an antifreeze solution is pumped through horizontal, vertical, or coiled heat exchangers. The typical HTF used for energy distribution in GSHP systems is also water. Whenever the term GSHP is used in this thesis, it refers to this last concept described.

Although ASHP are the most common systems in mild climates, they have some limitations since, as the outdoor air temperature (T_L) decreases, their efficiency decreases, as referred before for the ideal heat pumps, and their heating capacity diminishes. When outdoor air temperature drops below 0 °C, the evaporator freezes and the HP cannot work properly. This is the reason why ASHPs are not commonly installed in cold climates. In turn, GSHPs can theoretically be installed everywhere, but they have been mostly applied in cold climates.

Using the ground as a heat source or sink in space conditioning systems is attractive from a thermodynamic point of view, as the ground temperature is nearly constant and generally much closer to room conditions than the outdoor dry-bulb or wet-bulb temperatures over the whole year. Ground source heat pump systems commonly consist of water-to-water heat pumps coupled to closed loop vertical borehole heat exchangers. Closed loop ground heat exchangers of this type consist of a borehole (of 75-150 mm diameter) into which is inserted one or more loops of high density polyethylene pipe. The borehole is then either backfilled or, more commonly, grouted over its full depth (e.g. filled with graded sand). The depth of the borehole typically varies between 30 and 120 m (Fisher et al., 2006).

The financial viability of a heat pump system depends on a number of variables. First of all, the installation of a ground source heating and cooling system is more complex and expensive than the installation of an ASHP system. It involves drilling, digging and laying pipes, all of which are expensive and time consuming. On the other hand, the benefits are greater. The type of system that can be installed depends on local conditions, such as the hydrological, geological, spatial characteristics of the land and how much space is available around the building (Smart Cities Stakeholder Platform, 2013).

For ground coupling, the cost varies between different technologies and different geological settings. Today, a borehole heat exchanger costs between 30 and 60 €/m, with the lower prices prevalent in Scandinavia and higher prices in Austria and Germany. In the particular case of the GSHP system installed in Coimbra, the price of the BHE was 37 €/m, including drilling, digging, laying pipes and grout material.

Concerning drilling costs and BHE efficiency for the GSHP systems, the European Technology Platform on Renewable Heating and Cooling recommends that further R&D is needed in order to optimize the ground-coupling technology (RHC-Platform, 2013). Targets of reducing the installation cost by at least 25% in 2020 and 50% in a longer term, and of increasing the borehole heat exchanger efficiency by 25% until 2020, were defined.

2.2.3 - HEAT PUMPS EFFICIENCY

Typically, the HPs efficiency is given by the COP for heating and by the EER for cooling, by manufacturers. More recently the seasonal performance factor is being used, which corresponds to the system efficiency during a complete heating season and it is calculated by the ratio between the total thermal energy supplied (output) and the total electric energy consumed (input, including auxiliary devices) during the entire season. SPF is equal to the average COP of the whole system during a heating (or cooling) season.

The seasonal performance factor of a HP system can vary significantly depending on a variety of parameters such as the local climate, the operation mode, the operation period, the heat demand of the building and the building heating system, e.g. radiators, radiant floor heating or fancoils (Bayer et al., 2012). In the case of a GSHP system, additional parameters associated to the boreholes influence its performance like the soil type, depth of borehole, velocity in the pipe, thermal conductivity of grout, the thermal resistance and heat exchange rate, etc. (Chua et al., 2010; Luo et al., 2014). The most critical factor is the temperature difference between the heat source and the heat sink (Huchtemann et al., 2012). A higher temperature differential corresponds to a lower efficiency of the system.

Since GSHPs rely on extracting thermal energy from the ground, which behaves like an approximately constant temperature source, they are more efficient than air source heat pumps, in which the heat exchange is done with the outside air (Chua et al., 2010; Chiasson and Yavuzturk, 2009; Urchueguía et al., 2008). A recent study made in the ASHRAE headquarters building, in Atlanta, compared the energy consumption of the GSHP system and the Variable Refrigerant Flow (VRF) system (considered one of the most efficient technologies within HVAC applications), each one installed in two different floors of the same building. The study concluded that the GSHP system exhibits a substantially higher energy performance (65% in heating mode and 70% in cooling mode) than the VRF system (Spitler and Southard, 2014).

A summary review of the performance factors supplied by equipment manufacturers, and the values published in the USA Energy Star web site for the most efficient GSHPs in 2014 (Energy Star, 2014), are presented in Table 2.1.

As it can be observed, there are very different data for heat pump efficiencies and, in some cases, they are not comparable, since they correspond to different reference conditions. In order to easily compare the efficiency of different technologies for the same application, a more detailed and standard methodology for the efficiency calculations is foreseen in the EU legislation.

Table 2.1: Heat pumps efficiencies used in literature and given by manufacturers.

Heat pump	Country/Manuf.	COP/SPF	Source
GSHP	-	3 - 5	(Self, 2013)
GSHP	-	3.5	(Sarbu and Sebarchievici, 2014)
GSHP	-	3 - 4	(Mustafa Omer, 2008)
Air to water	-	4	(RHC-Platform, 2013)
GSHP	-	4.3	(RHC-Platform, 2013)
Brine/Water HP	BE, FR,UK	4.78 / 5.46	(Ecofys, 2013b)
Brine/Water HP	IT, ES	5.17 / 5.71	(Ecofys, 2013b)
Brine/Water HP	AT, DE	4.66 / 5.33	(Ecofys, 2013b)
Brine/Water HP	SE	4.34 / -	(Ecofys, 2013b)
Air to water		2.5 – 4.4	(IEA, 2011)
GSHP		2.8 - 5	(IEA, 2011)
GSHP	Central Europe	3.6	(EHPA, 2009)
Water to water	Daikin EWWD-J-SS	4 - 4.3	(Daikin, 2014)
GSHP	Daikin RWEYQ-T	5.2 – 5.9	(Daikin, 2014)
GSHP	Lenox MWC	3.9 - 4	(Lennox, 2014)
GSHP-water-air	Bosch-Geo 6000	4.3 -4.4	(Energy Star, 2014)
GSHP-water-air	Bosch Gr. Source	4.0 – 4.5	(Energy star, 2014)
GSHP with HW	BRYANT	3.6 – 3.8	(Energy star, 2014)
GSHP with HW	CARRIER	3.6 – 3.8	(Energy star, 2014)
GSHP with HW	ClimateMaster	3.8 – 4.6	(Energy star, 2014)
GSHP	GeoComfor	4.0 - 4.5	(Energy star, 2014)

2.2.3.1 - HEAT PUMPS ENERGY LABELING AND ECODESIGN

Ecodesign work is the basis for the development of energy labelling and ecodesign requirements for a number of energy-using appliances, including heat pumps. Ecodesign work is divided into several lots, each covering different areas that are progressing in different stages towards formulation and adoption of legislation for the area. Different types of heat pumps are covered by different lots, as illustrated in Figure 2.3.

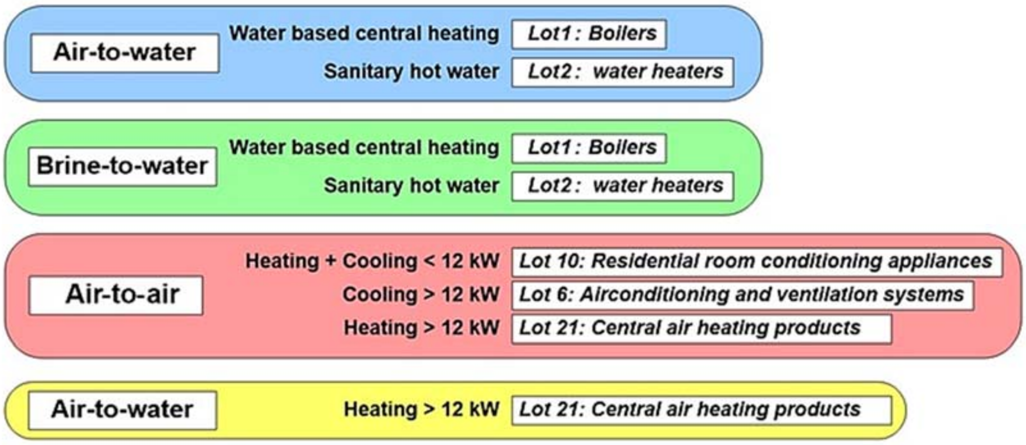
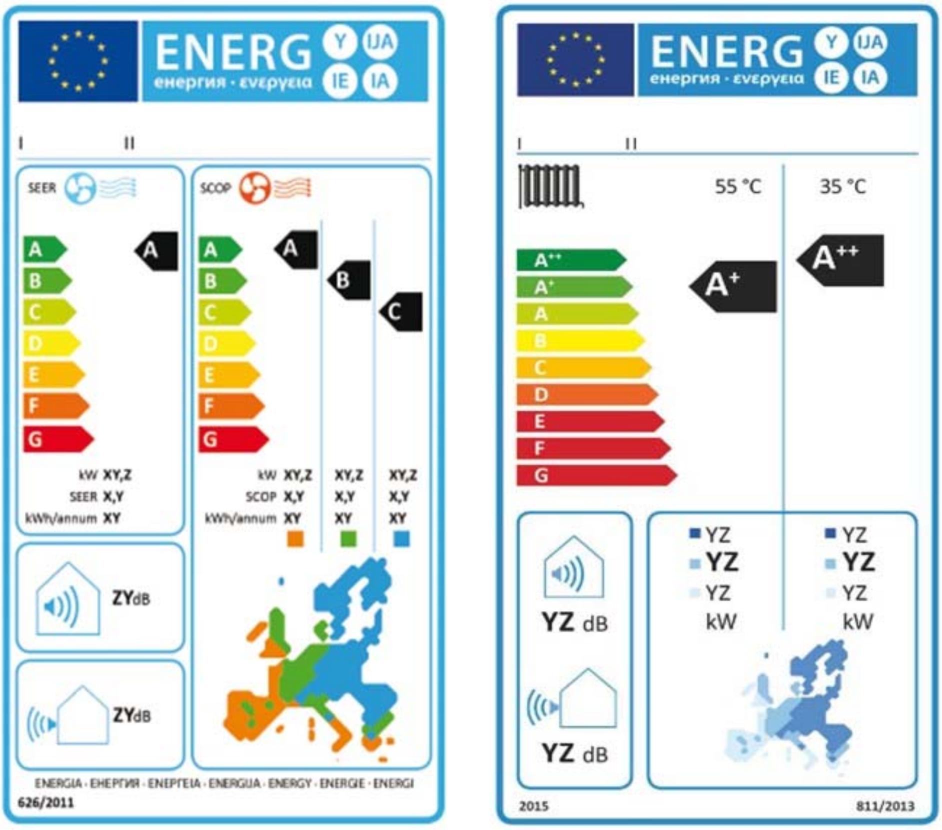


Figure 2.3: Lots of the ecodesign work where HPs are included (Rasmussen, 2011).

With the application of the Ecodesign Directive (EU, 2009a) and the Energy Labelling Directive (EU, 2010b) in the EU to the space and water heating technologies (European Commission, 2013a; European Commission, 2013b), including HPs, the efficiency will be classified by the Seasonal Coefficient Of Performance, that will be designated by SCOP for heating, and, if reversible, by Seasonal Energy Efficiency Ratio (SEER) for cooling, instead of SPF. An example of the energy label for air-to-air heat pumps (Lot 10), mandatory from January 1, 2015, is presented in Figure 2.4 a). In the case of water based central heating systems (Lot 1 – Boilers), where air-to-water HPs and GSHPs are included, the efficiency of the system will be classified by the seasonal space heating energy efficiency factor (ηs) which takes into account the necessary primary energy consumption instead the final energy consumption. For example, in the case of electric heat pumps the ηs is calculated by the ratio between SCOP and the primary energy factor for electricity generation added of some correction factors depending on the technology (European Commission, 2013b; Rasmussen, 2011). For water based central heating, two different efficiency classes will appear in the energy label (Figure 2.4b)), one for radiator heating (55 °C) and another for underfloor heating (35 °C) (European Commission, 2013b). The main purpose of this energy efficiency factor (ηs) is to allow the comparison of all the water based central heating systems available in the market, including combustion boilers.

Energy label is a benchmark for the end consumer to see how economical, environmentally friendly and/or energy saving the product is. The main purpose is to help end consumer to choose products which save energy and thus money. Taking all the performance indicators into consideration, the product will be put in a category ranging from A (best) to G (worst), with the option of adding classes A+, A++ and A+++ to accommodate for technical progress. Heat pumps are in the top classifications of the label and only heat pumps with an efficiency class of A+ and better will be allowed in the market (EHPA, 2015).



a) b)

Figure 2.4: Energy labels for HPs: a) air-to-air reversible HPs (European Commission, 2011c); b) HPs for space heating (water based) (European Commission, 2013b).

For Lot 1 (Boilers), minimum energy efficiency requirements are mandatory since January 2015. The minimum efficiency values for heat pumps used in water based central heating systems (European Commission, 2013a) are presented in Table 2.2.

Table 2.2: Heat pump minimum energy efficiency requirements for water based central heating.

Lot1 - Water based CH	ηs (%) 2015	SCOP	ηs (%) 2017	SCOP
Air to Water (55°C)	100	2.58	110	2.83
Water/brine to water (55°C)	100	2.7	110	2.97
Air to Water (35°C)	115	2.96	125	3.22
Water/brine to water (35°C)	115	3.1	125	3.38

Both legislative acts are expected to have positive impact on the heat pump technology market, since they will promote the most efficient products and will give to end users the chance to consider system efficiency to select the appropriate product for their specific needs. The Ecodesign Directive (EU, 2009a) will have far reaching implications for manufacturers, importers, consumers, contractors, consultants and architects, and will promote innovation in design and marketing of heating technologies (Ecofys, 2013a). Minimum Energy Performance Standards will remove from the market the units with lower performance.

2.2.4 - HEAT PUMPS AND RES

Heat pumps have been presented in several studies as a very promising technology to answer to the EU climatic and energetic targets. Heat pumps output can have a renewable energy contribution from 66% to over 80% (RHC-Platform, 2013). Moreover, since they are high efficient systems with improvements perspectives, they can effectively contribute to an energy efficiency increase of the heat production in EU, particularly in buildings through individual, central, collective and district heating systems, leading to a primary energy consumption reduction and, consequently, to a carbon emissions reduction and to a decreasing energy dependency of the EU.

On the other hand, electrical heat pumps will profit from increasingly cleaner electricity and can play an important role in the power grid management since they can store the electricity excess from intermittent renewables in the form of thermal energy, either in a dedicated heat store or in the building mass (Smart Cities Stakeholder Platform, 2013).

From the Member States National Renewable Energy Action Plans, heat pumps will have an aggregated target for renewable share of 5% of the gross final energy for 2020 at EU level, related to the EU RES Directive (EU, 2009b).

The EU RES Directive imposes that the RES contribution of HPs is only taken into account if the SPF is greater than 1.15 × 1/η, where η represents the electricity generation efficiency. According to guidelines for calculating renewable energy from heat pumps (European Commission, 2013c), an electricity generation efficiency of 45.5% must be used leading to a minimum SPF of 2.5. According to the European Commission Decision 2013/114/EU (European Commission, 2013c), that defines the methodology to calculate the supplied renewable energy (E_{RES}) from HPs, which is calculated by the equation:

$$E_{RES} = Q_{usable} \times \left(1 - 1/SPF\right) \tag{2.7}$$

where the Q_{usable} is the total usable heat delivered by heat pumps, which corresponds to the thermal energy they supply.

2.2.5 - HEAT PUMPS MARKET

The total heat pumps installed within Europe exceeded 6 million units until 2012, including ASHPs (air-to-air and air-to-water), GSHPs and reversible heat pumps (EHPA, 2014a). The air-to-air systems with cooling as main function are not included in the analysis. The thermal capacity (kW_{th}) of individual units ranges typically from about 3 kW_{th} for residential use to 150 kW_{th} for non-residential building installations (EurObserv'ER, 2013). In Figure 2.5, the total units of HP sold between 2005 and 2012 per 10000 households in some EU countries are shown. It can be seen that Sweden, Finland and Estonia have the highest HP penetration. Sweden is the country with the higher density of HPs in EU and it is considered a very mature and an almost saturated market.

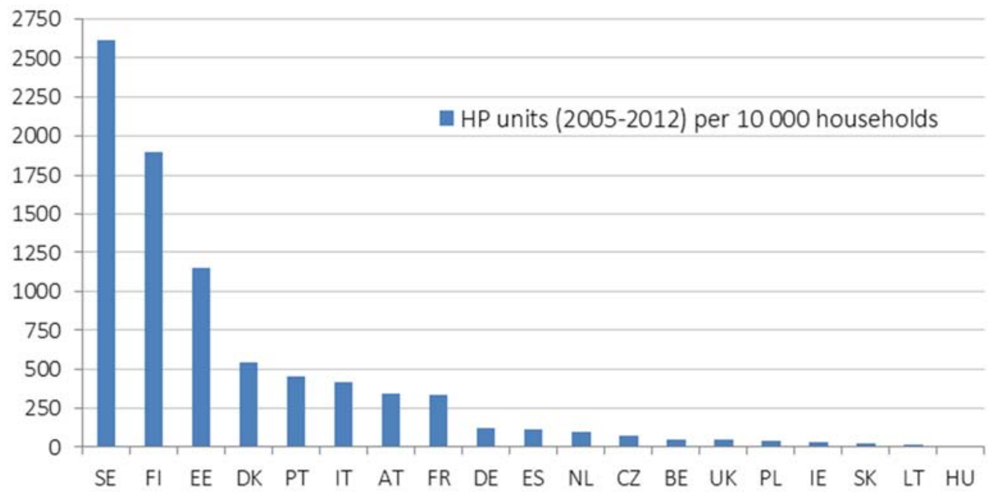


Figure 2.5: Heat pump density per households by EU country based on heat pump sales from 2005 to 2012 (adapted from Nowak (2013)).

Between 2005 and 2012, 5.4 Million of heat pumps were sold and contributed to a RES integration of 41 TWh, to Greenhouse Gases (GHG) emissions savings of 11 Mt of CO_2eq, and to primary energy savings of 29.5 TWh (EHPA, 2014b).

As depicted in Figure 2.6, after 2008 the EU heat pump market did not follow the increasing tendency observed between 2005 and 2008. It dropped due to the financial crisis in 2009 and, although it has slightly recovered in 2010 and in 2011, the heat pump sales dropped again in 2012 to values below the 2008 sales. This last drop can be explained by the construction market crisis in some countries, by changes in the electricity prices and in by political measures, among others (EurObserv'ER, 2013).

As a consequence, and according to the EHPA projections, the heat pump market needs to grow substantially until 2020, so that the RES integration target reaches 5% (141 TWh), as expected in the EU RES Directive 2009/28/EC (EU, 2009b). EHPA has forecasted that until 2020 around 14.5 Million of HP units will be sold, which can lead to a reduction of 34.5 Mt of CO_2eq, primary energy savings of 80 TWh and a RES integration of 131 TWh, which means that a higher HP penetration was expected in the RES target (EHPA, 2014b).

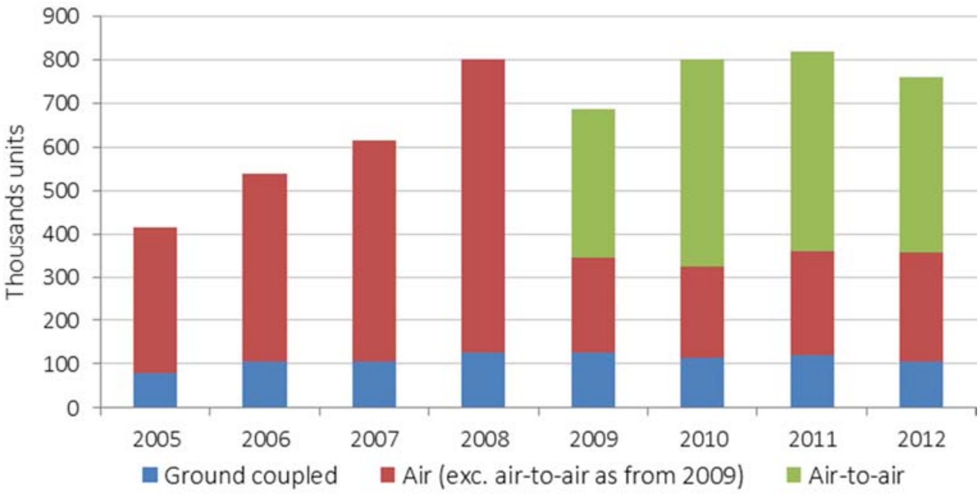


Figure 2.6: Heat pump sales in Europe between 2005 and 2012 by energy source (adapted from Nowak (2013)).

After 2009, the air-to-air and air-to-water heat pumps are presented separately, since they belong to different products groups with the application of the Ecodesign Directive 2009/125/EC (EU, 2009a), for air conditioners (European Commission, 2012) and for space heaters (water based central heating) (European Commission, 2013a), respectively.

Figure 2.6 also shows that air source heat pumps have been the dominant technology in the HP market. Even in the segment of hydronic distribution, the air-to-water HPs represented around 65% of the market in 2012 against the 35% share of GSHPs (EurObserv'ER, 2013). This can be explained due to a higher complexity and to higher costs of the GSHPs installation, associated to the drilling of the boreholes. Consequently, the economic advantage of GSHPs only comes after a longer term, particularly in countries with mild winters (EurObserv'ER, 2013). Improved and innovated drilling methods of the boreholes are needed to reduce the installation costs and to reduce the impact on the surroundings, in way that a higher GSHPs penetration can be achieved in next years (RHC-Platform, 2013).

The majority of GSHP systems are located in Northern and Central Europe. The highest numbers can be found in Sweden, Germany and France, which correspond to the countries with highest capacity installed (Figure 2.7).

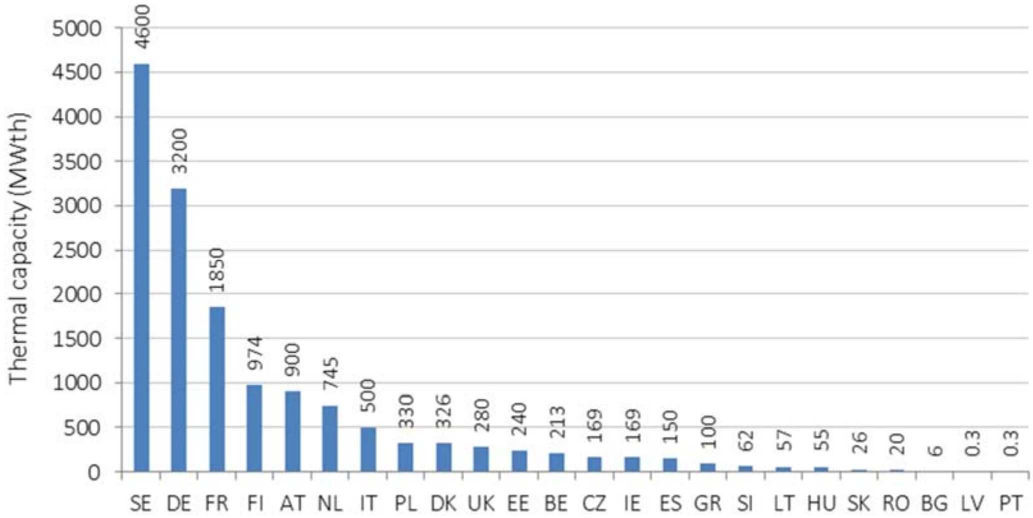


Figure 2.7: GSHP thermal capacity Installed in 2013 per country in EU (Dumas, 2013).

The market penetration (installations per capita) is highest in Sweden, Finland and Austria. GSHPs are already used in Southern Europe, but at a very small scale. Further R&D and practical experience is crucial to fully exploit the advantages of geothermal heat pumps in warmer climates for both cooling and heating appliances. With this purpose some European R&D Projects such as, Ground-Reach, Groundhit and more recently Ground-Med (Ground-Med, 2014) have been developed for South European countries.

2.3 - HEAT PUMPS FOR SUSTAINABLE SPACE HEATING

In the European Union, a great focus has been given to the heating sector with several studies to assess the potential for energy and emissions savings and renewable penetration in the heating sector. Some examples are the Heat Roadmap Europe 2050 studies (Connolly et al., 2012; Connolly et al., 2013), the Common Vision for the Renewable Heating and Cooling sector in Europe (RHC-Platform, 2011), the Strategic Research and Innovation Agenda for Renewable Heating and Cooling (RHC-Platform, 2013), among others. At national level, as a result of the Energy Efficiency Directive 2012/27/EU (EU, 2012), EU member states are required to submit national heating and cooling roadmaps until the end of 2015. A European project co-funded by the Intelligent Energy Europe Programme, the Stratego project (Stratego, 2015), was created to help national and local authorities in the development of national heating and cooling roadmaps.

More specific to the buildings sector, several studies have presented heat pumps as a very promising technology to contribute to the building sector decarbonization and to decrease the primary energy consumption for the heat demand in the EU, like the studies which are presented in next paragraphs.

De Almeida et al. (2004) have developed a case study in Portugal to evaluate the fuel switching opportunities for the residential sector and concluded that the use of electric air-to-air and air-to-water HPs for space heating and water heating were an attractive option, in terms of the lowest energy consumption and of the lowest carbon emissions.

Nishio and Hoshino (2010) estimated a 52% reduction potential of the CO_2 emissions from residential space heating and water heating if existent technologies (gas and oil boilers, as well as electric appliances with low efficiency like resistance heating) were replaced by heat pump technologies. This study was carried out for the seven countries responsible for one-third of the total amount of CO_2 emitted worldwide (United States, Japan, Canada, Germany, United Kingdom, Italy and France).

Bayer et al. (2012) made a comprehensive study about GHG emissions savings of the already installed GSHP systems for space heating in Europe. They estimated that 3.8 Mt of GHG emissions (0.74% from total) were saved in residential space heating, in nineteen European Countries, by the GSHP systems, when compared with conventional technologies. They also estimated that if all fossil fuels, electricity and wood fired boilers would be replaced by GSHPs with a SPF of 3.5, an average saving of up to 30% of CO_2 emissions is possible in space heating. This impact can be increased in the future with higher performance factors of GSHP and with the increasing RES share in electricity generation, leading to lower carbon content.

A study performed by Ecofys (Ecofys, 2013a) assessed the impact of different heat pump penetration rates in the building sector of eight European countries until 2030 (Austria, Belgium, Germany, Spain, France, Italy, Sweden and the United Kingdom). Based on three different scenarios of HPs penetration rates, they determined CO_2eq emissions and primary energy savings for heating, cooling and hot water till 2030, and the RES-share contribution from heat pumps for 2020. In the optimistic scenario with the highest heat pump implementation shares (100% for new buildings and 50% in retrofits buildings), CO_2eq emissions dropped by 46% (less 296 Mt) and primary energy dropped by 39% (1260 TWh) in 2030, compared to 2012 values, for the above mentioned end-uses. They also concluded that, in this scenario, the RES share from heat pumps could reach 9.8% of the gross final energy.

Other similar studies (Sarbu and Sebarchievici, 2014; Self et al., 2013; Blum et al., 2010), also conclude that heat pumps are a powerful technology for sustainable space heating.

The analysis of the previous studies showed that high efficient heat pumps have, in most cases, very positive impacts on the emissions and primary energy savings. However, the results can vary substantially depending on the considered key factors in the analysis, such as: specific country energy consumption for space heating by fuel type, typical space heating technology by fuel type, heating technology efficiency, heat pump efficiency and emissions factor for electricity generation.

One of the main differences found is the emissions factors that can represent the total GHG emissions (CO_2eq). Some authors use the life cycle assessment, which takes into consideration the overall life cycle of the energy carrier and technology (Sarbu and Sebarchievici, 2014; Self et al., 2013; Ecofys, 2013a; Bayer et al., 2012; Blum et al., 2010; Saner et al., 2010), and others use the direct GHG emissions (Self et al., 2013) or only the direct CO_2 emissions (Blum et al., 2010). In the first case, the emissions factor is always higher, which means that for the same situation the emissions savings will be lower. Another significant difference is due to the SPF that is used – the higher is the SPF used, the higher will be the primary energy and emissions savings. In this respect, HPs technology made significant progresses in the last 20 years, with further potential progress still available.

2.4 - THERMAL ENERGY STORAGE

2.4.1 - THERMAL ENERGY STORAGE IN BUILDINGS

Thermal energy storage has proven to be a technology that can have positive effects on the energy management of a building by contributing to an increased share of renewable energy and/or reduction in energy demand or peak loads for both heating and cooling (Heier et al., 2015). Thermal energy storage technologies operate with a goal of storing energy when it is available or in excess (for example solar energy, wind energy, run river hydro energy) for later use as heating or cooling capacity. Thermal storage technologies are well suited for an array of applications, including seasonal storage on the supply-side and demand management services on the demand-side of the energy system (IEA, 2014).

Seasonal storage is long term storage and usually refers to storage between seasons. Application of seasonal thermal energy storage with heat pumps for heating and cooling buildings has received much consideration in recent decades, as it can help to cover gaps between energy availability and demand (Heier et al., 2015). For example, during summer, the solar energy can be stored, normally in underground systems (boreholes, aquifers or tanks) (Tatsidjodoung et al., 2013) to be used later in winter to satisfy heat demand. In general low temperatures are used to reduce heat losses, which imply the use of auxiliary systems like heat pumps to elevate temperature to the demand desired levels. It works as a preheating of the heat source used by the heat pump leading to a higher efficiency (Hesaraki et al., 2015). This has the potential to reduce the large proportion of energy consumed by buildings, especially in colder climate countries. Underground TES for low temperature applications (at less than 40 °C) has been demonstrated and is now available in some European markets, particularly in the Netherlands and Sweden (Xu et al., 2014). Several review works and original studies on seasonal TES have been presented by Hesaraki et al. (2015), Xu et al. (2014), Pinel et al. (2011) and Qi et al. (2008), among others.

One of the main applications of thermal storage has been, for several years, its use as a strategy to improve the power demand profile leading to a reduction of the peak demand and of grid operation costs, to a higher reliability of the electrical grid and to a reduction of the consumers electricity bill (Kennedy et al., 2009; Qureshi et al., 2008; Stadler, 2008; Omagari et al., 2008). The application of TES in buildings has the following advantages (Zhang et al., 2007):

- The ability to narrow the gap between the peak and off-peak loads of electricity demand, achieving a higher load factor. This allows to avoid the use of expensive and pollutant power plants (natural gas and oils power plants – used for peak periods) leading to some primary energy and emissions savings.
- The ability to save operative costs by shifting the electrical consumption from peak periods to off-peak periods since the cost of electricity at night is much lower than during the day.
- The ability to use intermittent renewable energy sources, storing thermal energy when there is large renewable generation, and releasing the thermal energy when necessary, particularly for space conditioning.

This type of strategy where the demand profile is changed in order to reduce electricity bills, when electricity costs are time-dependent and peak power demand has high charges, are classified as demand side management strategies. Thermal energy storage has been used for long time as a load shifting strategy to decrease power demand and electricity consumption during peak periods, due to the high rates that are applied during these periods. Three different load shifting strategies can be associated with thermal storage as presented in Figure 2.8.

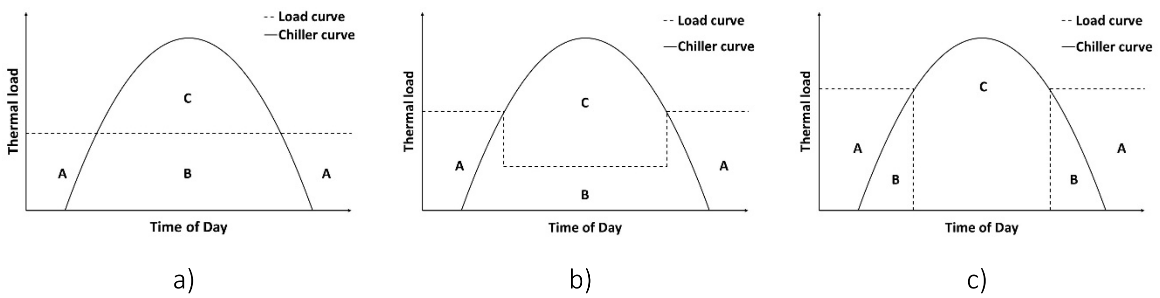


Figure 2.8: Load shifting strategies: a) load levelling, b) demand limiting (both partial storage), c) full storage (adapted from Dincer (2002)).

Full storage is used when the total replacement of the building cooling or heating equipment during day is desired (Figure 2.8c). In this situation the chiller capacity remains the same and larger storage capacity is needed requiring higher initial investment and higher space for the storage system installation (Dincer, 2002).

Partial storage is used when the main purpose is to partially reduce the operation of the cooling or heating equipment during peak demand and peak prices periods. During these periods thermal storage is used in addition to the heating or cooling system, which allows to install lower capacity equipment and to reduce the operation costs of the space conditioning system. Partial storage allows the application of load levelling and demand limiting strategies, as represented in Figure 2.8a) and Figure 2.8b).

Although these strategies can be used for both heating and cooling, they are more often discussed and applied for cooling of buildings. Cold storage has been used for decades for peak demand management as a load shifting strategy, in facilities where there is high cooling demand, such as in office buildings, schools and hospitals (Sun et al., 2013). The main technology used for cooling purposes in large buildings is centralized chillers. During off peak period the cooling energy produced by chiller is directly stored in TES systems. The most common media of cooling storage is ice, since it provides large energy density and water is easily available, chemically stable and cheap. In scientific literature, there are hundreds of studies about cold storage systems, much more than for heating purposes. In last years, several authors published papers reviewing used materials and applications of cold storage, like for example: Sun et al. (2013), Zhai et al. (2013), Osterman et al. (2012), Yau and Rismanchi (2012), Ban et al. (2012), among others. Parameshwaran et al. (2012), in its review of sustainable thermal energy storage technologies for buildings, also concluded that many research studies have been focused on cold thermal energy storage technologies for buildings cooling and air conditioning purposes, than for heating applications.

Heier et al. (2015) made a recent review of TES applications in buildings in Europe and concluded that for services buildings, especially in office buildings, TES has been used to reduce peak energy production and that seasonal storage can be a viable option if space is available. They refer the need of more studies on real buildings where TES is used in order to disseminate TES advantages and that TES should be a priority in new and renovated building projects.

Nowadays, TES assumes a role even more important with the fast increase of intermittent renewable generation from various forms of distributed energy sources. Thermal energy storage systems can store intermittent electric energy when produced in the form of thermal energy to be used when necessary in cooling and heating applications. On the other hand, it contributes to increase the flexibility of electric systems for cooling and heating, allowing their use as flexible loads for Demand Response strategies in the smart grids context. This type of strategy requires more modern power grids with two-way flow of information and communication, load control by electronic metering and automatic meter management systems, which has been on the top of the agenda of the European research with a large number of smart grids projects (Covrig et al., 2014).

Space heating is a suited load for DR application at buildings level, since space heating constitutes the major share of buildings load in EU (Odyssee, 2015) and, if thermostatic controlled, it can be curtailed, reduced or postponed as long as the user comfort is maintained (Ali et al., 2014). This will contribute to higher short term TES research and commercial applications for space heating in buildings than those verified till now. For example, the Cold Climate Housing Research Center, in its Thermal Storage Technology Assessment Report (Stevens et al., 2013), referred that, although TES systems present high potential for space heating, there are limited educational materials and literary articles on thermal storage systems in cold climates and on how thermal storage can be best integrated into a heating system.

Although TES is a very promising strategy that can be used in several applications, some barriers to its massive implementation still exist and must be overcome (Stevens et al., 2013), such as:

- The lack of information on thermal storage;
- The lack of commercial options for installing a thermal storage system;
- The capital cost of these type of systems;
- The infrastructure constraints (such as limited space in buildings);
- There is no measurement standards dedicated to energy storage materials (namely PCMs);
- There is no industry standard for determine the efficiency of a thermal storage system.

2.4.2 - TES TYPES AND MATERIALS

Thermodynamics dictates that there are three types of thermal storage: chemical, latent and sensible heat storage. Chemical energy storage refers to heat stored in reversible chemical reactions. The other two types of storage, sensible and latent, refer to heat that can be stored in a material. As heat energy is added to a solid, for example, two processes happen. Firstly, the temperature of the solid goes up and the heat added that causes this temperature change is called sensible heat. Secondly, when the temperature of the solid reaches the melting point, the solid undergoes a phase change to a liquid. During the phase change, heat is added to the material, but the temperature remains constant. The heat needed for the phase change is referred to as latent heat (Stevens et al., 2013).

2.4.2.1 - SENSIBLE HEAT STORAGE

Sensible heat storage is the most common method of heat storage. A good storage medium for sensible heat is water, be it in tanks, pits, aquifers (groundwater), caverns, abandoned mines, or others. Thermal oil is also used as a liquid storage material. Also solid matter can be used, like soil, rocks, bricks, concrete, iron, dry and wet earth, and many others. It is considered a comparatively mature technology that has been implemented and evaluated in many buildings. Water, rock-sort material and ground/soil are frequently used as storage materials for underground TES at the supply side (Xu et al., 2014).

For sensible heat storage systems it is important to use materials that have high specific heat (c [J/(kg.°C)]) and good thermal conductivity (k [W/(m.K)]). For building applications, materials with high density (ρ [kg/m^3]) are also important (Tatsidjodoung et al., 2013). Within sensible materials, water has the higher specific heat (4.19 kJ/(kg.°C)) but one of the lowest densities (1000 kg/m^3). Sensible heat storage systems have relatively low energy density (space limitation is usually a critical aspect, especially for building applications) and have high heat losses, being necessary to invest in strong insulation. These characteristics present the main disadvantages of this type of TES. In the case of storage water tanks, in addition, there is the problem associated to the stratification (higher water temperatures are found in the top and lower water temperature in the bottom of the tank). Some research has been done and some recommendations were presented in the "Strategic Research and Innovation Agenda for Renewable Heating and Cooling" study, developed by RHC-platform (2013), to improve the efficiency of these systems, such as: the replacement of metals by polymer casings in the containment materials, the use of high efficient heat exchangers, the use of high performance insulation materials, like vacuum insulation, and the use of stratification devices to reduce disturbances associated to the temperature stratification.

2.4.2.2 - LATENT HEAT STORAGE

Latent heat storage is associated with the stored heat in the material while changing phase. The main characteristic of this technology is that during the phase change the materials remain, theoretically, at constant temperature (real systems show temperature stabilization around the melting temperature). As already stated, the materials used in latent heat storage are known as phase change materials. The best known and most used PCM is water (used as ice in cold storage applications), but salt solutions (for low temperature applications), paraffins, salt hydrates, fatty acids, sugar alcohols (between 0 °C and 130 °C), and inorganic materials and salts (for temperatures above 150 °C) are also used.

Energy storage in PCMs has a lot of advantages over sensible systems because they lead to a lower mass and volume of the system, the energy is stored at a relatively constant temperature and energy losses to the surroundings are lower. Rousse et al. (2009) present a comparison of different thermal storage materials, as shown in Table 2.3.

Table 2.3: Comparison between common heat storage materials (adapted from Rousse et al. (2009)).

Property	Rock bed	Water	Organic PCM	Inorganic PCM
Density [kg/m^3]	2240	1000	800	1600
Specific heat [kJ/(kg.K)]	1.0	4.2	2.0	2.0
Latent heat [kJ/kg]	–	–	190	230
Latent heat [kJ/m^3]	–	–	152	368
Relative storage mass*	15	4	1.25	1.0
Relative storage volume*	11	6	2.5	1.0

** with respect to inorganic PCMs*

Depending on the PCM used, the storage volume can decrease significantly when compared with water storage tanks, which can overcome one of the big barriers of thermal storage that is the space available for its installation.

2.4.2.3 - THERMOCHEMICAL STORAGE

Chemical reactions are characterized by a change in the molecular configuration of the compound involved during the reactions. Thermal energy is absorbed when compounds are dissociated and released when recombined. In thermochemical storage, the reaction usually involves a solid and a gas that can react. In building applications, the gas involved is usually water vapor. Thermochemical Materials (TCMs) have the highest energy storage density and are seen as key materials for achieving loss-free thermal storages, also for long-term compact storage. However, the literature concerning this heat storage technique and materials is marginal compared to sensible and latent storage materials, since the research in that field is still in an early stage (Tatsidjodoung et al., 2013).

Figure 2.9 show the different volumes needed to full cover the annual storage need of a house by each technology.

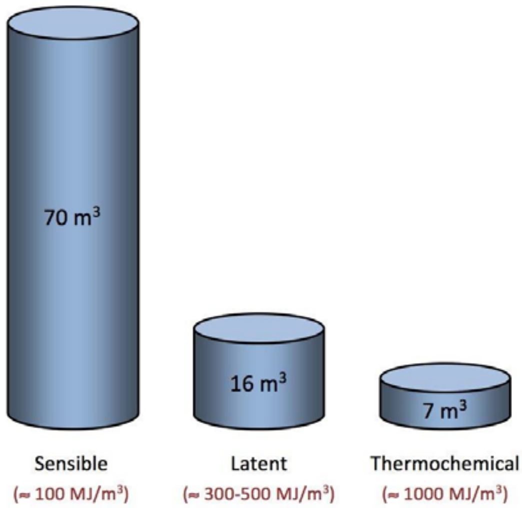


Figure 2.9: Volume needed to full cover the annual storage need of an energy efficient passive house (6480 MJ) by the three technologies (adapted from Tatsidjodoung et al. (2013)).

According to the Technology Roadmap for Energy Storage (IEA, 2014), while some energy storage technologies are mature or near maturity (e.g., residential water heaters with storage, ice storage, cold water storage and underground TES are already in a commercialization phase), PCMs are in a phase of demonstration and deployment, and other ones are still in the early stages of research and development like the thermochemical storage.

2.4.3 - TES MAIN CATEGORIES: PASSIVE AND ACTIVE STORAGE

TES in buildings can be divided into two major categories: passive and active storage. For a passive storage, the driving force for charging and discharging the stored energy is only the temperature difference between the store and the surroundings. In the case of an active storage, the charging and/or discharging occurs with active help from pumps or fans. There are also combinations of the two where the charging is active and the discharging is passive or vice versa. This can be the case with thermally active building structural elements, for example active ceilings, where charging with cold during the night might be active, while discharge during daytime is passive. Most of the techniques used for TES in buildings are active and only two are passive techniques such as sensible storage in building elements with high thermal mass and latent storage using PCMs in the building walls, ceiling or floor.

2.4.3.1 - PASSIVE STORAGE TECHNIQUES

There are two different passive techniques:

- Sensible storage in building elements with high thermal mass.
- Latent storage using PCM in the building walls, ceiling or floor.

Passive storage can have different purposes, the most known is to increase thermal comfort by smoothing the fluctuations in the indoor temperature. Another one, is using the thermal mass of the building in order that heat supply can be postponed from the heat demand for a certain period, depending on the characteristics of the building (Ellerbrok, 2014).

Sensible storage in building thermal mass

Depending on the type of materials used in buildings structure and of the thickness of walls, thermal mass of the building can have more or less capacity to store energy. The thermal mass (or thermal inertia) is physically defined by the building heat capacity value (C [kJ/°C]), which is directly proportional to the specific heat (c [J/(kg.°C)]) and the mass (m [kg]) of the materials. The theoretical total heat capacity of a building is the sum of the heat capacities of all structure elements (walls, ceilings and floors). Each element (e.g., wall) may be constituted by several layers, each one with different heat capacity, depending on the material used. The theoretical total heat capacity can be calculated by Equation (2.8).

$$C = \sum m_i c_i = \sum (\rho_i v_i)\ c_i \tag{2.8}$$

C – Building heat capacity [kJ/°C]

m – Building element mass [kg]

c –Specific heat of the material [J/(kg.°C)]

ρ – Density of the material [kg/m^3]

v – Building element volume [m^3]

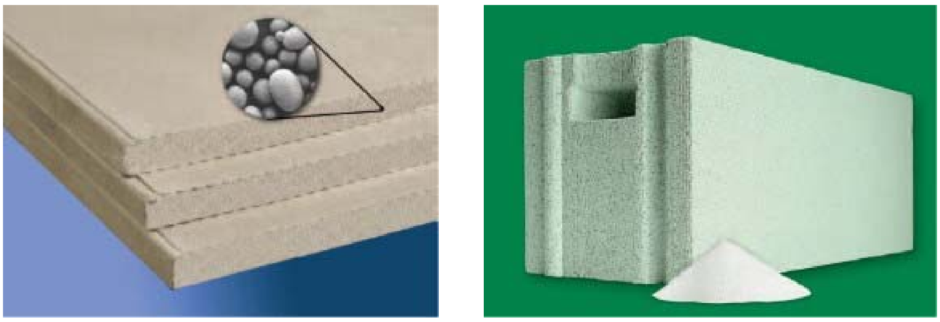
It must be noticed that the sum of the heat capacities of the individual building elements is only of very limited use, since the mass outside insulation (e.g., brick and veneer), mass in thick walls (e.g., Trombe walls) and distributed mass (e.g., drywall) have very different roles in determining building performance. Therefore, to assess the dynamic response of a building it is necessary to determine the building effective heat capacity, as referred by Subbarao et al. (1985).

It is common to characterize buildings as heavy or massive buildings when they have high thermal inertia and as light buildings when they have low thermal inertia. In light buildings, outdoor conditions (temperature, wind, radiation) have much more influence in indoor temperatures, leading to higher temperature oscillations than in heavy buildings.

Ferrari (2007) highlighted the importance of re-discovering thermal mass for energy reduction demand, not only for cooling (which was more usual) but also for heating, and concluded in his study that savings are higher in heating than in cooling, by comparing the test results obtained in a light building and in a heavy building during winter and summer. He also demonstrated that, beyond the ongoing trend in considering the insulation as a key issue for building energy conservation, thermal mass can also play an important role.

Latent storage using PCM in the building elements

Recently, much attention and research has been done to improve buildings thermal mass by the incorporation of PCMs in building elements such as in walls (gypsum plasterboards, façade panels), ceilings (ceiling panels), floor component (floor panels), brick (cylindrical holes filled with PCM). Typically, in this situation the PCM is microencapsulated in a thin, sealed and high molecular weight polymeric film, maintaining the shape and preventing PCM from leakage during the phase change process (Zhou et al., 2012), and mixed with buildings materials. An example of commercial products which can be incorporated into building elements is the Micronal PCM. In Figure 2.10 are shown two examples: a gypsum wallboard ("PCM SmartBoard") and concrete block ("CelBloc Plus").



a) b)

Figure 2.10: Two commercial examples of PCM incorporation into buildings elements: a) Wall board – "PCM SmartBoard"; b) Concrete block – "CelBloc Plus" (BASF, 2015)

This is especially beneficial in lightweight constructions, which suffer from low thermal inertia. For example, in cold climates where buildings have been built according to passive house standards, often involving large amounts of insulation to reduce heating loads in the winter, large temperature fluctuations in the summer happen due to excessive overheating caused by a lack of thermal mass (Kalnæs and Jelle, 2015).

Over the past few years several reviews have been written on the use of PCMs in buildings for passive thermal energy storage and indoor climate comfort purposes clearly showing that the interest for PCMs is increasing for passive storage systems (Zhu et al., 2009; Agyenim et et al., 2010; Cabeza et al., 2011; Pomianowski et al., 2013; Soares et al.,2013).

According to Kalnæs and Jelle (2015) and Heier et al. (2015), although many laboratory and full scale experiments have shown positive results on energy savings with the application of PCM in buildings elements, there is a scarcity of data published on actual performance in real life applications so far.

2.4.3.2 - ACTIVE STORAGE TECHNIQUES

According to Heier et al. (2015) the active storage can be divided in three groups:

- Active storage in the HVAC system includes storage both for heating and cooling purposes, more or less integrated with the heating, ventilation or air conditioning system of the building. Some of the most widely used storage techniques today are water tanks (sensible or latent) combined with solar collectors or with other space conditioning systems.
- Active storage in the building structure includes storage in building elements, such as buildings walls, floors or ceilings, designed to actively charge and/or actively discharge heat from these building components. They are known as thermally activated building systems. For heating, the most common is the floor heating (electric or thermal, with water pipes) and for cooling are the active ceiling panels. In these situations, the charge and/or discharge of the storage medium can be done by using air or water as heat transfer fluid.
- Storage in the direct vicinity of buildings includes aquifer, borehole and snow storage, as well as storage in pits or buried tanks. These thermal energy storage systems are located outside the building and are often designed for larger systems or large groups of buildings and for seasonal storage.

Active storage systems allow a better control, namely when and how to charge or discharge the stored energy. Some specific examples of active TES applications for space heating combined with HPs are presented in section 2.5.

2.4.4 - LATENT HEAT STORAGE – PHASE CHANGE MATERIALS

There are several materials that can be used as phase change materials. PCMs can be classified as organic, inorganic or eutectic (Tyagi and Buddhi, 2007), as presented in Figure 2.11. Organic materials include paraffins and non-paraffins (fatty acids, esters and alcohols), and inorganic materials include metals and salt hydrates. An eutectic compound is a minimum-melting composition of two or more substances. Eutectics almost always melt and freeze without separating because they freeze into a crystalline structure and then melt simultaneously.

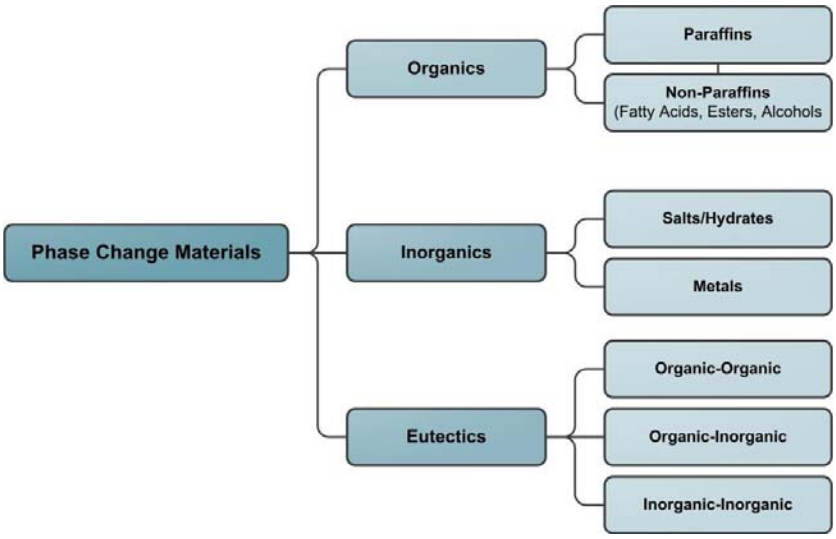


Figure 2.11: Classification of phase change materials.

The challenge of finding a material to use as a PCM is to find one with as many optimal characteristics as possible (Stevens et al., 2013). Phase change materials should present, in general, the following desirable thermophysical, kinetics, and chemical properties (Rousse et al., 2009):

- Appropriate phase change temperature (according to the operating temperature of the system);
- High heat storage capacity, known as latent heat of fusion (to minimize the size/volume of the heat store);
- High thermal conductivity (influences the heat transfer rate and thus the time of charging and discharging);
- Long-term chemical stability (to allow large number of charging and discharging cycles without storage capacity decrease due to phase separation);

- Compatibility with container materials, small volume change and low vapor pressure (to avoid problems with containers);
- No toxicity and no fire hazard (safety reasons);
- High crystallization rate;
- Ideally no supercooling. Supercooling is a phenomena that occurs when the PCM starts crystallization (solidification) only at temperatures well above the melting temperature, which limits the heat released at the desired temperature (Mehling and Cabeza, 2008);
- Low cost and availability (economic reasons).

The type and the behavior of the selected phase change material have a great influence on the thermal storage performance of the unit. One of the most important selection criteria for a PCM is the phase change temperature (Mehling and Cabeza, 2008). Other main criteria for the selection of PCMs are, high latent heat of fusion per unit mass, high thermal conductivity, little or no supercooling during freezing and chemical stability.

Higher latent heat of fusion per unit mass means a lower storage system size, which is very important in terms of space availability (one of the main barriers for TES installation). A low thermal conductivity of the PCMs affects the performance of the storage unit, since the heat transfer rate in the LHS unit and thus the performance of the system during charging (melting) and discharging (solidification) mainly depends on the difference between the heat transfer fluid temperature and the melting point of the PCM. Many promising PCMs have low thermal conductivity (around 0.15-0.2 W/(m.K)) for organic PCMs and around 0.5 W/(m.K) for inorganic salts). Low thermal conductivity reduces the rate of heat absorption or heat release throughout the PCM and may lead to not fully use the latent heat stored in the PCMs (Kalnæs and Jelle, 2015). Little or no supercooling during freezing and chemical stability are essential properties to have high number of charging and discharging cycles (long lifetime).

The most common PCMs used are paraffin waxes (organic) and hydrated salts (inorganic), which are solid-liquid PCMs. Paraffin waxes are cheap and have moderate thermal energy storage density but low thermal conductivity and, hence, require a large surface area for heat transfer. Hydrated salts have a larger energy storage density and a higher thermal conductivity (Demirbas, 2006). Salt hydrates are popular phase change materials for thermal energy storage because of their high latent heat. However, many of these substances are prone to supercooling, which is problematic for normal applications, as it prevents the release of the stored latent heat. The greater the degree of supercooling, the higher energy will be released during the initial supercooling and less energy will be available at the time of crystallization, i.e., less energy will be used at the phase change temperature (Sandnes and Rekstad, 2006).

A recent review of the type of PCMs used in cooling and heating application studies was done by Tatsidjodoung et al. (2013) and some of them are presented in Table 2.4, Table 2.5 and Table 2.6.

Table 2.4: Organic paraffins with potential use as PCM for heating and cooling applications.

Application	Compound	Melting temperature [°C]	Heat of fusion [kJ/kg]	c * [kJ/(kg.K)]	k * [W/(m.K)]	ρ * [kg/m³]
COOLING	Paraffin C15-C16	8	153	2.2 (s)	–	–
	Polyglycol E400	8	99.6	–	0.187 (liq, 38.6 °C) 0.185 (liq, 69.9 °C)	1125 (liq, 25 °C) 1228 (s, 3 °C)
	Dimethyl-sulfoxide (DMS)	16.5	85.7	–	–	1009 (s/liq)
	Paraffin C17-C18	20-22	152	2.2 (s)	–	–
	Polyglycol E600	22	127.2	–	0.189 (liq, 38.6 °C) 0.187 (liq, 67.0 °C)	1126 (liq, 25 °C) 1232 (s, 4 °C)
	Paraffin C13-C24	22-24	189	2.1 (s)	0.21 (s)	760 (liq, 70 °C) 900 (s, 20 °C)
HEATING	Paraffin C20-C33	48-50	189	–	0.21 (s)	769 (liq, 70 °C) 912 (s, 20 °C)
	Paraffin C22-C45	58-60	189	2.4 (s)	0.21 (s)	795 (liq, 70 °C) 920 (s, 20 °C)
	1-Tetradecanol	38	205	–	–	–
	Paraffin C16-C28	48-50	189	–	0.21 (s)	765 (liq, 70 °C) 910 (s, 20 °C)

(–): not available
** s = solid; liq = liquid*

Table 2.5: Eutectics PCMs suitable for heating and cooling applications.

Group	Application	Compound	Composition (%)	Melting temperature [°C]	Heat of fusion [kJ/kg]	k * [W/(m.K)]	ρ * [kg/m³]
INORGANICS	COOLING	$CaCl_2 \cdot 6H_2O + CaBr_2 \cdot 6H_2O$	45+55	14.7	140	–	–
	HEATING	$Mg(NO_3)_3 \cdot 6H_2O +$ NH_4NO_3	61.5+38.5	52	125.5	0.494 (liq, 65.0 °C) 0.515 (liq, 88.0 °C)	1515 (liq, 65 °C) 1596 (s, 20 °C)
		$Mg(NO_3)_3 \cdot 6H_2O +$ $MgCl_2 \cdot 6H_2O$	58.7+41.3	59	132.2	0.510 (liq, 65.0 °C) 0.565 (liq, 88.0 °C)	1550 (liq, 50 °C) 1630 (s, 24 °C)
		$Mg(NO_3)_3 \cdot 6H_2O +$ $MgCl_2 \cdot 6H_2O$	50+50	59.1	144	–	–
ORGANICS	COOLING	Triethylolethane+ water+urea	38.5+ 31.5+30	13.4	160	–	–
	HEATING	$NH_2CONH_2 + NH_4NO_3$	53+47	46	95	–	–

(–): not available
*** s = solid; liq = liquid*

Table 2.6: Inorganic substances with potential use as PCM for heating and cooling applications.

Application	Compound	Type of melting*	Melting temperature [°C]	Heat of fusion [kJ/kg]	c ** [kJ/(kg.K)]	k ** [W/(m.K)]	ρ ** [kg/m³]
COOLING	$LiClO_3 \cdot 3H_2O$	c	8.1	253	1.35 (s)	–	1720
	$ZnCl_2 \cdot 3H_2O$	–	10	–	–	–	–
	$K_2HPO_4 \cdot 6H_2O$	–	13	–	–	–	–
	$NaOH \cdot 7/2CO_2$	–	15	–	–	–	–
	$Na_2CrO_4 \cdot 10H_2O$	–	18	231	1.31 (s)	–	–
	$KF \cdot 4H_2O$	c	18.5	125.9	1.84 (s)	–	1447 (liq, 20 °C) 1455 (s, 18 °C)
HEATING	$K(CH_3COO) \cdot 3/2H_2O$	–	42	–	–	–	–
	$K_3PO_4 \cdot 7H_2O$	–	45	–	–	–	–
	$Ca(NO_3)_2 \cdot 4H_2O$	i	42.7	–	–	–	–
	$Na_2HPO_4 \cdot 7H_2O$	i	48	281	1.70 (s) 1.95 (liq)	0.514 (32 °C) 0.476 (49 °C)	1520 (s) 1442 (liq)
	$Na_2S_2O_3 \cdot 5H_2O$	i	48-49 48	209.3 201-206	3.83 (liq)	–	1666
	$Zn(NO_3)_2 \cdot 2H_2O$	c	54	–	–	–	–
	$NaOH \cdot H_2O$	c	58	–	2.18 (s)	–	–
	$Na(CH_3COO) \cdot 3H_2O$	–	58	226 267	– 2.79-4.57	– 0.63 (s)	1450 1280
	$Cd(NO_3)_2 \cdot 4H_2O$	c	59.5	–	–	–	–
	$Fe(NO_3)_2 \cdot 6H_2O$	–	60	–	–	–	–

(–): not available
** c = congruent; i = incongruent*
*** s = solid; liq = liquid*

2.4.4.1 - MAIN TOPICS OF RESEARCH ABOUT SOLID-LIQUID PCMS

Analyzing the vast scientific literature about PCMs research, it can be concluded that the main topics about PCMs suitable for cooling and heating applications are the following:

- Thermo-physical properties of PCMs and methods of analyzes;
- Methods to enhance thermal conductivity and heat transfer rate to and from PCMs;
- Methods to overcome the problems associated to PCMs such as phase separation and supercooling (salt hydrated PCMs);
- Methods of PCMs incorporation into construction materials (direct incorporation, immersion and encapsulation);

- Type of PCMs encapsulation, material and geometry of the PCMs containers and heat transfer phenomena, according to the type of geometry used;
- Heat transfer phenomena and thermodynamic properties which characterize the three different phases of phase change problems (melting, solidification and mushly phase - transient phase);
- Formulation of the phase change problem and limitations of the different models used (analytical and numerical models and use of computational tools);
- Methods to evaluate the performance of TES systems using PCMs, among others.

Zalba et al. (2003) presented results of a critical review of 237 sources in the area of thermal energy storage using PCMs. As in the work of Kenisarin et al. (2007), the main attention was also on thermophysical properties of PCMs which could be used in solar applications. One of the conclusions was that PCM thermophysical properties had not been studied sufficiently in order that clear recommendations could be made for the design process of commercial heat storage units. Even now there is no comprehensive database on thermophysical properties available. According to Kenisarin et al. (2007) there are significant discrepancies in the data published on the properties of PCMs, which demonstrate that the accuracy of the used methods is not sufficiently good.

A recent review on the investigations done over the past few decades about the thermal stability of PCMs has recently been presented by Rathod et al. (2013). For fatty acids, the purity plays an important role. Most studies have shown that the thermal stability of salt hydrates is poor due to phase separation and supercooling. However, the thermal stability may be improved to a certain extent by introducing gelled or thickened mixtures and suitable nucleating materials.

Several studies have been conducted about heat transfer enhancement techniques in phase change materials and inclusion of finned tubes of different configurations: bubble agitation, insertion of a metal matrix, using PCM dispersed with high conductivity particles, micro-encapsulation of the PCM or shell and tube (multitubes) (Agyenim et al., 2010).

Therefore, the development of a latent heat thermal energy storage system involves the understanding of heat transfers/exchanges in the PCMs when they undergo solid-to-liquid phase transition in the required operating temperature range, the design of the container for holding the PCM and the formulation of the phase change problem (Agyenim et al., 2010).

Some studies were carried out to analyze the behavior and the heat transfer performance in the charging and discharging process of some materials. In several studies it was demonstrated that the dominant heat transfer mechanism which occurs between the HTF and the PCM (melting) is natural convection and between the PCM and the HTF (discharging) is conduction, independently of the encapsulation geometry (rectangular, cylindrical or spherical) (Jegadheeswaran and Pohekar, 2009).

Other important factors to consider in TES with PCMs are the geometry of the PCM container and the thermal and geometric parameters of the container required for a given amount of PCM. Each of these factors has a direct influence on the heat transfer characteristics in the PCM and ultimately affects the melting time and the performance of the PCM storage unit. A survey of previously published papers dealing with LHS reveals that two geometries commonly employed for PCM containers are the rectangular and cylindrical containers (Agyenim et al., 2010).

Barba and Spiga (2003) studied analytically the discharge process of the phase change material and evaluated its effectiveness, for constant surface temperature conditions, in three different geometrical configurations, i.e., considering the PCM encapsulated in slab, cylindrical or spherical polyethylene containers in hot water tanks. Among different geometrical configurations of the PCM container, they found that the shortest time for complete solidification was matched for small spherical capsules.

Results from the reported use of numerical methods appear to show that they offer a good approach to solving the phase change problem, although most of the available solutions to phase change problems apply one or two-dimensional systems due to the complexity of the equations involved in the phase change. The latent heat phase change problem has been assumed by some authors to be controlled conduction, whereas natural convection in the liquid phase has been included in the analysis by other authors, owing to the complexity of the phase change problem (Agyenim et al., 2010).

According to Mehling and Cabeza (2008), analytical solutions can give important insights when, for example, the main purpose is to analyze the time development of the heat flux in geometric encapsulations applications, but have a lot of restrictions, which can lead to significant errors. Poor results can be achieved when using analytical models to get quantitative information, since do not include sensible heat, do not include convection, assume that PCM is at the phase change temperature and that boundary temperatures are constant. Numerical models give more detailed and realistic analyses of real problems, since the physical problem is defined by differential equations. Numerical models are more flexible with respect to thermal effects and to the geometry that can be treated.

PCMs are a very promising technology but still have a number of disadvantages, which must be overcome, in terms of: higher costs, lack of reliable information, lack of commercial options, low thermal conductivity, material stability over several cycles, phase segregation and supercooling (Stevens et al., 2013). Several authors referred, in their study reviews on PCM applications in buildings, that more research about TES with PCM is needed to overcome these limitations and improve their viability (Tatsidjodoung et al., 2013; Parameshwaran et al., 2012; Zhou et al., 2012).

2.4.4.2 - COMMERCIAL PCMS

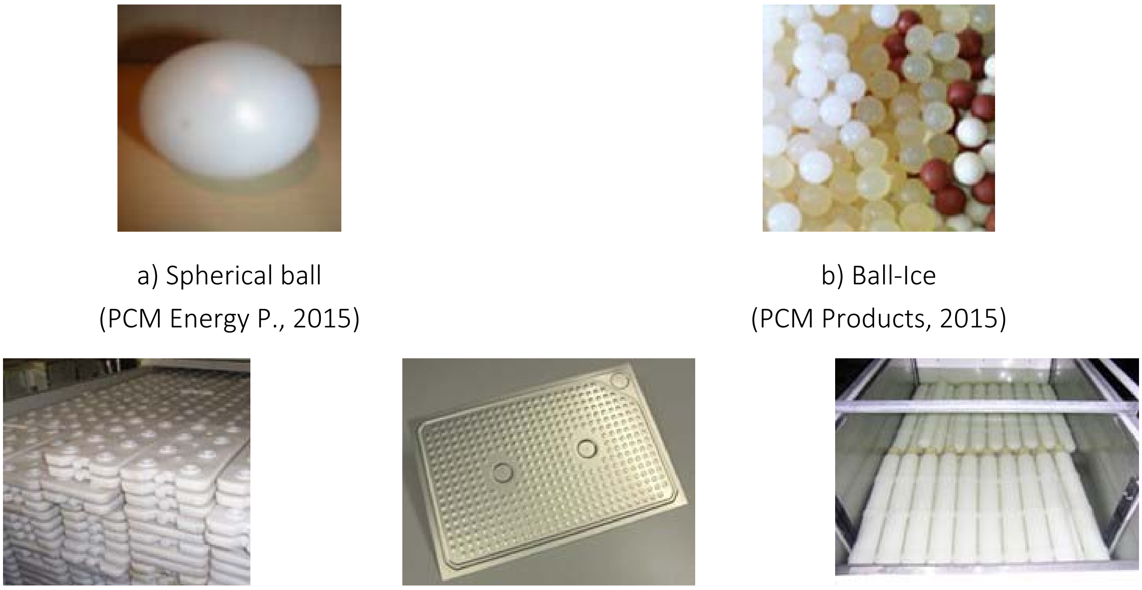
Besides the scientific literature review, some research was done about the PCMs available in the market for space heating applications. In this way a research of the PCMs manufacturers in the World and of their products was carried out. A list of PCMs available in the market within the melting point range of 30 to 50 °C is presented in Table 2.7.

Table 2.7: Characteristics given by manufacturers for PCMs available in the market.

PCM name	Type of Product	Melting point [°C]	Heat of Fusion [kJ/kg]	Specific Heat [kJ/(kg.K)]	Thermal Cond. [W/(m.K)]	Encapsulation	Supplier
Latest™34S	Salt hydrate	33-34	220	2	0.6	Panels / balls	PCM Energy P. Ltd (India)
HS34	Salt hydrate	32-35	150	-	-	Panels / balls	PLUSS (India)
S34	Salt hydrate	34	115	2,1	0,52	Tube-Ice (Cylindrical) or Flat-Ice (Rectangular)	PCM products Ltd (UK)
Latest™36S	Salt hydrate	34-36	230	2	0.6	Panels / balls	PCM Energy P. Ltd (India)
A39	Organic	39	190	2,2 - 2,22	0,21 - 0,2	Ball-Ice (Spherical)	PCM products Ltd (UK)
PK	n.a. (New product)	up to 40 °C	-	-	-	Not contained in a supporting structure	Rubitherm (Germany)
GR 42	SiO_2, Paraffin	41 (38 – 43)	57	1,5	0,2	Granulate substance	Rubitherm (Germany)
PX 42	SiO_2, Paraffin	41 (38 – 43)	114	1,6	0,1	Powder substance	Rubitherm (Germany)
RT42	Paraffin (C_nH_{2n+2})	41	174	2,1	0,2	CSM Panel (Rectangular)	Rubitherm (Germany)
A42	Organic	42	230	2,2 - 2,22	0,21 - 0,2	Ball-Ice (Spherical)	PCM products Ltd (UK)
S44	Salt hydrate	44	105	1,61	0,43	Tube-Ice (Cylindrical)	PCM products Ltd (UK)
S46	Salt hydrate	46	190	2,41	0,45	Tube-Ice (Cylindrical)	PCM products Ltd (UK)
Climsel C48	Sodium Sulphate, Water and Additives	48	226,8	3,6	0,5-0,7	-	Climator (Sweden)
Latest™48S	Salt hydrate	48	>230	2	0.6	Panels / balls	PCM Energy P. Ltd (India)

- Not available / Not defined

The different types of encapsulation developed by PCMs manufacturers are presented in Figure 2.12. The most common encapsulation geometries for air conditioning applications are spherical, cylindrical and rectangular. The geometry used depends on the PCM substance type (organic, inorganic or eutectic). The typical geometry found for paraffins is spherical.



a) Spherical ball (PCM Energy P., 2015)

b) Ball-Ice (PCM Products, 2015)

c) Flat-Ice (PCM Products, 2015)

d) CSM Panel (Rubitherm, 2015)

e) Tube-Ice (PCM Products, 2015)

Figure 2.12: Commercial encapsulation geometries: a) and b) – Spherical encapsulation; c) and d) - Rectangular encapsulation; e) - Cylindrical encapsulation.

Some real applications of TES systems with PCMs combined with GSHPs in buildings are presented in manufacturers literature. An application of a GSHP and TES systems in a large glass façade building of 5000 m2 in Turin, Italy, is described by Ure (2008). The main purpose of these systems is to guarantee thermal comfort for the whole year at maximum energy efficiency level. The heat storage tank was filled with 5600 Flat-Ice modules (Figure 2.12c), which correspond to a total volume of 25 m^3 and to 1200 kWh of heat storage capacity. The chilled water storage tank was filled with 3840 Flat-Ice modules, which correspond to a total volume of 20 m^3 and to 900 kWh of heat storage capacity. The storage underground tanks were designed to operate with a temperature of 46 °C (PCM melting point) during winter and of 13 °C (PCM melting point) during summer, to store the energy equivalent to three days usage at average load.

Figure 2.13: GSHP combined with TES installed in Turin, Italy (PCM Products, 2015).

Ure (2008) referred that the average annual energy savings of the GSHP combined with TES compared to a traditional best practice energy saving technology was estimated to be around 46%. In Italy, this means a pay-back period, without any regional utility incentives, of 8.6 years, or, with regional utility incentives, of just over 4 years. According to Ure (2008), this energy savings can be achieved due to the lower temperatures operation used in winter (water supply/water return = 40 °C / 36.5 °C) and the higher temperatures operation used in summer (17 °C / 19.5 °C), when compared with operating temperatures of other conventional systems (typically 7 °C / 12 °C for cooling and 70 °C / 50 °C for heating).

Other two examples were found (PCM Products, 2015), but no data about the systems performance and the impacts of the use of these systems is available. In the Burnley College Lancs in UK (Figure 2.14), a solar heat pump was installed combined with two thermal storage cylindrical tanks. The heat storage tank was filled with 33000 paraffin balls, which correspond to a total volume of 6 m^3 and to 300 kWh of heat storage. The PCM has a melting point of 45 °C and its encapsulation material is a plastic mixture (Figure 2.12b). The chilled water storage tank has the same storage capacity and volume, but was filled with paraffin balls with a melting point of 8 °C. The paraffin balls are lighter than water and when the tank is filled with water they tend to float and fill the whole volume of the tank, contributing to a fast heat exchange when water flows through the balls.

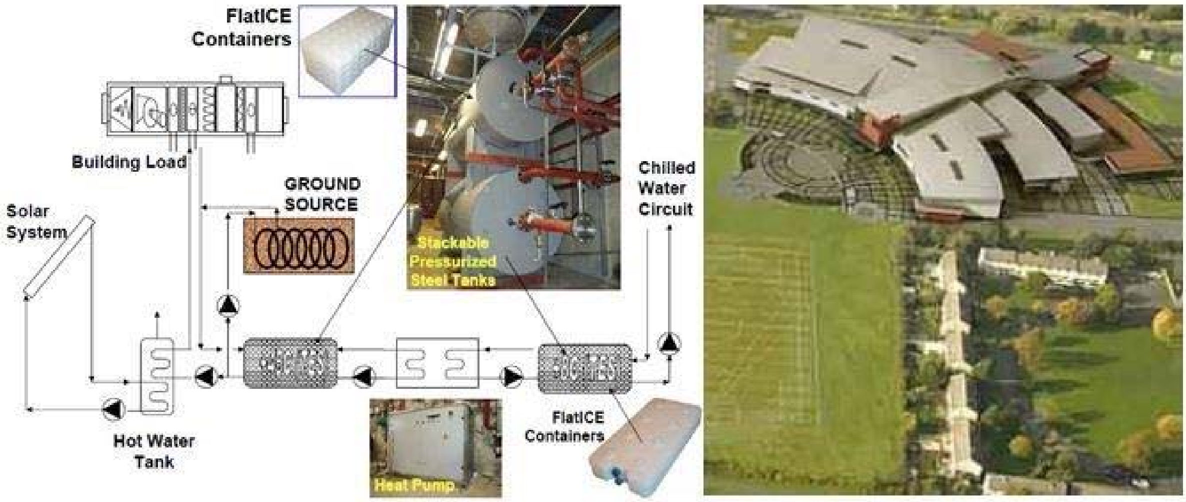


Figure 2.14: GSHP combined with TES installed in Burnley College Lancs in UK (PCM Products, 2015).

In the case of the Pendle Vale College Lancs in UK, a similar system was installed, but the PCM is inorganic and has a rectangular encapsulation (Flat-Ice modules). The heat storage tank was filled with 400 Flat-Ice modules, which correspond to a total volume of 3 m^3 and to 150 kWh of heat storage. The PCM has a melting point of 45 °C. The chilled water storage tank has the same storage capacity and volume, but was filled with Flat-Ice containers with a melting point of 8 °C.

2.5 - HPS AND TES APPLICATIONS FOR SPACE HEATING IN BUILDINGS

This section presents some case studies of the application of TES systems for buildings space heating, giving preference to situations where they are combined with HPs, namely GSHPs.

Tyagi and Buddhi (2007) have done a comprehensive review of the thermal performance of various possible methods for heating and cooling storage systems in buildings using PCMs. The storage systems found are mainly passive. The active storage systems such as the floor heating or the ceiling boards are more usual in the literature than the storage tanks units.

Storage tanks units with PCMs are mostly associated with solar-aided heat pumps for buildings space heating (seasonal and others), but most of the studies of latent heat storage are theoretical or simulation studies. Qi et al. (2008) enhanced that, using a storage tank with PCM instead of a water storage tank, solar collectors efficiency improves and the energy losses to the surroundings are lower for seasonal storage systems coupled to heat pumps. Hamada and

Fukay (2005) presented a study of an air source heat pump combined with two water tanks filled with PCMs for heating purposes installed in a building. The main focus of this study was to assess the effect of the carbon fiber brushes on the thermal outputs in terms of heat rate transfer improvement, having achieved positive results in their numeric simulations. The performance of the overall system and the energy savings when compared with conventional technologies were not a focus of this study.

Moreno et al. (2014) made a review of PCM thermal energy storage systems incorporated in air source heat pumps for space heating and cooling. Most of the studies presented in their paper are experimental research in laboratory scale. They concluded that for space heating, TES systems can be incorporated within the heat pump cycle for load shifting and as a defrosting system of the outdoor unit, being the PCMs most used the calcium chloride hexahydrate ($CaCl_2$ $6H_2O$) and paraffin, with phase change temperature within the range 22.5 - 58 °C.

The main application of GSHPs for space heating is mostly combined with solar energy through solar collectors, and they are commonly called solar aided GSHPs. GSHPs are mainly used in heating dominated climates and along the years it can cause a thermal heat depletion of the ground. To avoid this temperature decrease of the soil, thermal energy from solar collectors is injected in the borehole/ground heat exchangers. The soil and the heat exchangers are the thermal storage medium and are used quite often as seasonal storage, storing solar energy during summer to be used during winter.

Wang et al. (2009) carried out an experimental study on a solar-assisted GSHP with solar seasonal TES installed in a building. In summer, the soil was used as the heat sink to cool the building directly. In winter, the solar energy was used as a priority, and the building was heated by a GSHP and solar collectors alternately. The results show that the heat directly supplied by solar collectors accounted for 49.7% of the total heating output, and the average COP of the heat pump and the system were 4.29 and 6.55, respectively. After a year of operation, the heat extracted from the soil by the heat pump accounted for 75.5% of the heat stored by solar seasonal thermal storage.

Another experimental research on solar assisted GSHP and seasonal TES was developed by Kanhong et al. (2010), but in this case it is also coupled to a latent heat storage tank. The PCM used was paraffin with a melting point of 20 °C. Solar energy is stored in the PCM tank to be used as a heat source by the HP during night and rainy/cloudy days. In this case, the HP uses three different heat sources from ground, solar collectors and a PCM tank.

Zhu et al. (2014) made a recent research review of the applications of GSHP systems coupled with TES. As concluded in the research done so far, within this thesis work, Zhu et al. (2014) also concluded that the major application of GSHPs is a combination with solar collectors. They also indicate that there are not enough studies about GSHP combined with other storage systems like water tanks and PCM and that they are mainly simulation studies. They also concluded that solar systems increases the overall system performance and that the introduction of TES systems decreases the overall performance, since for the same heat supply more electricity is consumed in auxiliary equipment (circulation pumps). According to Zhu et al. (2014), although the overall performance decreases with TES application, significant savings can be achieved, since part of the stored energy is solar or heat recovered and if not stored would be wasted.

According to Zhao et al. (2014), a number of studies have been carried out by various researchers in order to analyze the performance of solar assisted GSHP systems. Most studies are performed by theoretical or simulation means and only a few are experimental. They also concluded that even for solar assisted ground source heat pump systems there is a lack of experimental data, which leads to weak evaluation of operating performance.

2.6 - CONCLUDING REMARKS ABOUT THE STATE OF THE ART

As a conclusion, heat pumps for space heating and cooling and thermal energy storage have been referred as key technologies to address the future energy and climate challenges in several important studies of the European Commission and other entities (RHC-Platform, 2011; Connolly et al., 2012; Connolly et al., 2013; RHC-Platform, 2013; IEA, 2014). Heat pumps are a well-known technology and already proven to be one of the most efficient. They have the advantage that can be reversible and used for cooling purposes, avoiding the investment in another technology. Ground source heat pumps are even more efficient than air source HPs, but in moderate climates are less usual than in cold climates, due to the higher payback.

Thermal energy storage technologies for cooling are much more common than for space heating. The main reasons for a higher development of TES cooling technologies, already with proven and commercial technologies available (e.g. ice storage and chilled water), are the availability, chemical stability and lower cost of the material used, which is water. Since space heating is the leading energy consumer in the EU buildings, further developments for local space heating storage are needed, in order to increase heat demand flexibility and apply DSM strategies for helping in grid management and in integration of intermittent renewable resources. In Portugal, space heating is not so representative as in EU due to its mild climate, but tends to grow due to the increasing of the comfort requirements from users.

Concerning recent and emerging TES technologies with high potential of application in buildings, further research and demonstration are needed, as referred by several authors (Heier et al., 2015; Tatsidjodoung et al., 2013; Parameshwaran et al., 2012; Zhou et al., 2012)). In the meantime, readily available and well-known technologies should be used, like for example the building thermal mass in medium and heavy buildings, since it can represent significant thermal storage capacity, as referred by Ellerbrok (2014), Reynders et al. (2013), Arteconi et al. (2013) and Ferrari (2007).

CHAPTER 3

EXPERIMENTAL GSHP INSTALLATION IN A SERVICE BUILDING

A high efficiency Ground Source Heat Pump (GSHP) system was installed in a pilot building in Coimbra (Portugal) under the European Project Ground-Med, as in other seven Southern Countries demo sites (Ground-Med, 2014). The main purpose of this project was to demonstrate the feasibility of this technology with a high seasonal performance factor (greater than five). For this purpose, the installation of high efficiency components was required, namely the Heat Pump (HP), the fancoil units and the circulation pumps.

In this chapter, firstly, a description of the building and its 2nd floor, which is space conditioned by the GSHP system, is presented, followed by the description of the main components of the hydraulic circuit and its characteristics. Furthermore, the monitoring system, as well as the developed control system, and finally, the thermal storage system based on Phase Change Materials (PCMs) that was intended to be installed, are also presented.

3.1 - COIMBRA PILOT BUILDING

The pilot building in Coimbra, where the GSHP system was installed and the experimental tests were carried out, is a tertiary building which houses two public services: the CCRC (Comissão de Coordenação Regional do Centro) and the ARH Centro (Administração da Região Hidrográfica do Centro). This building has four floors with a total area of 3250 m^2 and it is located next to the Mondego River (Figure 3.1).

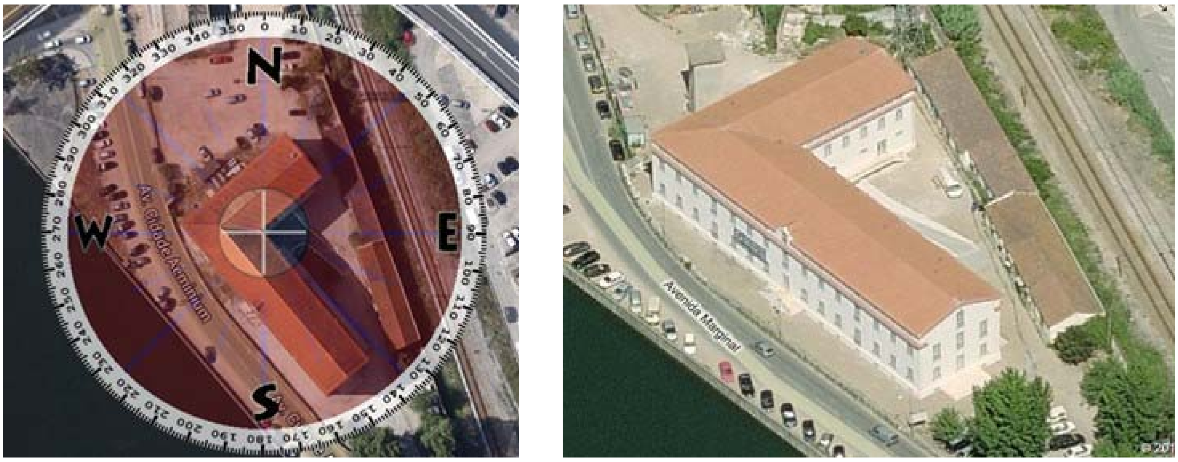


Figure 3.1: The pilot building in Coimbra where the GSHP system was installed.

It is an old building (built around 1900) with thick walls made of limestone masonry (density of 2100 kg/m^3 and specific heat of 0.81 kJ/(kg.°C)) with thickness that varies from 110 cm, in the basement, to 70 cm in the 2nd floor. The external walls are not thermal insulated and the building has a significant glazed area with poor thermal insulation.

The building is open during week days from 8 a.m. to 8 p.m. The main occupancy period is from 9 a.m. to 6:30 p.m., as it can be observed by the electrical load profile of the building presented in Figure 3.2 (although some people arrive before 9 a.m. and others leave after 6:30 p.m.). The peak electricity demand of the total building occurs in the morning, when workers turn on all the electric equipment (including the space heating system), which is coincident with the national morning peak load demand period.

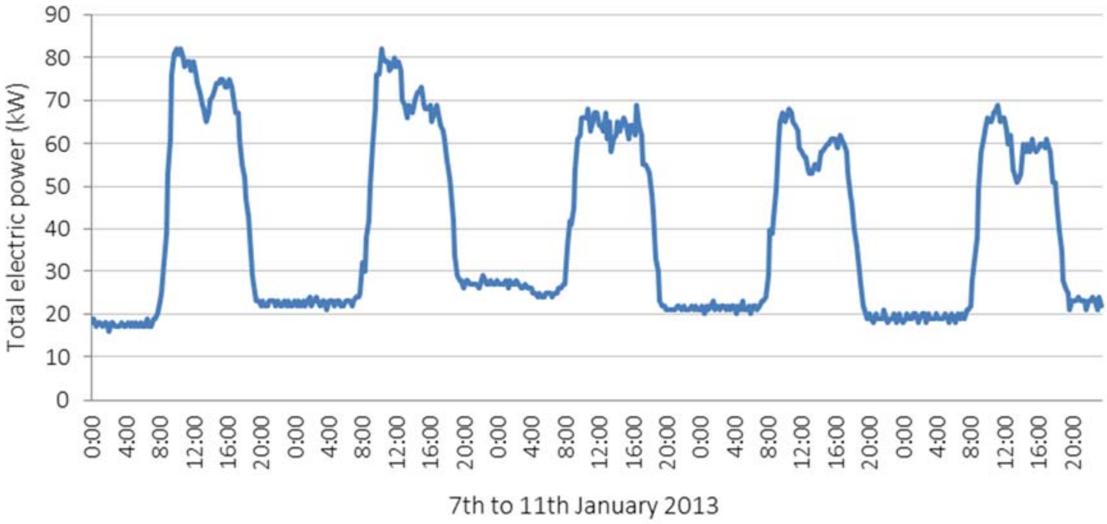


Figure 3.2: Electrical load profile of the total building in a winter week (telemetering data).

The GSHP system was designed and installed to satisfy the thermal needs of the 2nd floor. This is the floor which presents the highest thermal gains and losses due to its connection to the roof and due to the large glass area. This floor has 22 offices and has a space conditioned area of arround 600 m^2. The main façade of the building, with more offices and higher glass area, has an orientation very close to southwest (SW) (Figure 3.3).

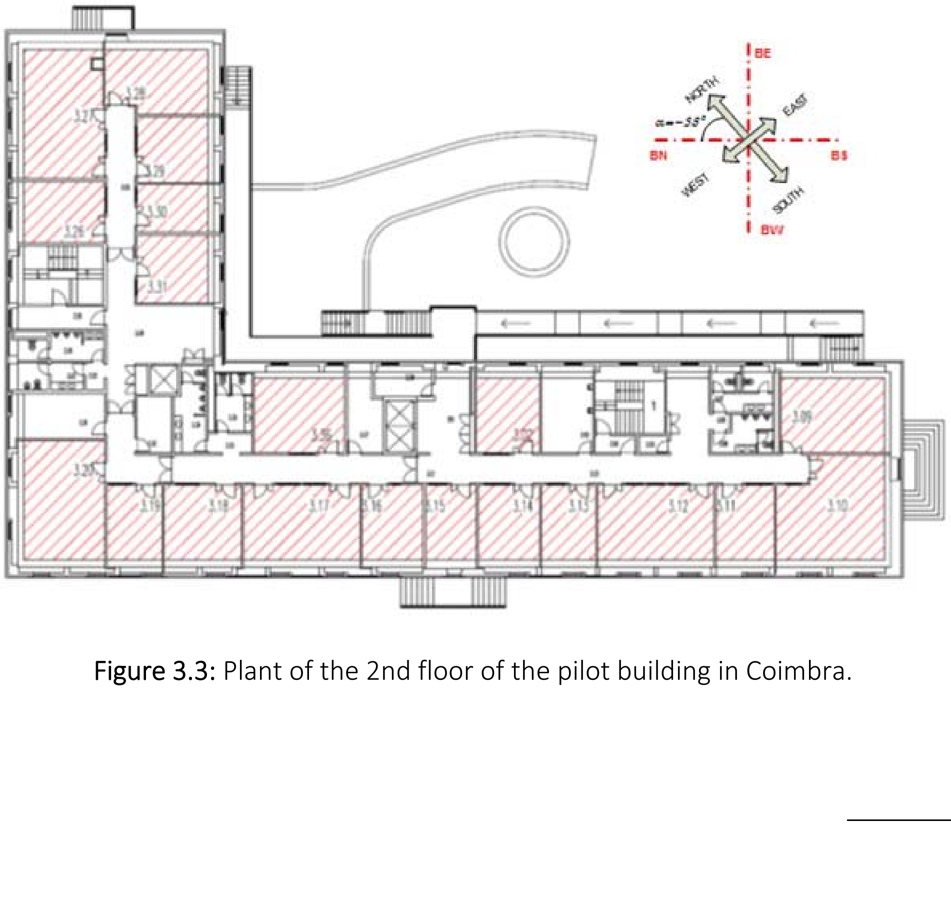


Figure 3.3: Plant of the 2nd floor of the pilot building in Coimbra.

The glazed area represents 24% of the total external walls area of the 2nd floor. There are two different types of glazed area, as presented in Figure 3.4, one is an open window with colorless glasses and the other is a fixed glass with blue tone, being both double glazed. However, the aluminum frames do not have any thermal insulation layer.

Figure 3.4: Picture of the southeast façade of the 2nd floor.

The 2nd floor is directly connected to the building under roof (ample space with a volume of about 1110 m^3), separated by a false ceiling made of gypsum board 13 mm thick with no insulation in the offices, and by removable steel panels in the corridors and halls, also without insulation.

The thermal mass of the external walls is considerable. About two thirds of the external walls are made of limestone masonry. The remaining is concrete, double brick walls or both (Table 3.1). The glazed area is interleaved with this last wall (Ext-wall 3A and Ext-wall 3B). The different types of external walls of the 2nd floor are presented in Table 3.1 and in Figure 3.5.

Table 3.1: Types of external walls of the second floor.

External wall types	Height [m]	Thickness [m]	Thermal capacity per wall area [kJ/(°C.m^2)]	Main composition
Ext. wall 1	0.66	1.0	1500	Limestone masonry wall
Ext. wall 2	0.83	0.7	1140	Limestone masonry wall
Ext. wall 3A	0.75	0.4	275	Double brick cavity wall (between windows)
Ext. wall 3B	0.75	0.2	390	Concrete pillars (between windows)
Ext. wall 4	0.20	0.4	818	Concrete lintels over windows

Figure 3.5: Office (left) and corridor (right) pictures of the 2nd floor.

The main data about each of the 22 offices of the 2nd floor are presented in Table 3.2, such as floor area, office volume, thermal power installed by floor area and by office volume, and the main equipment that contributes to the internal thermal gains.

Table 3.2: Main data about offices of the 2nd floor.

Conditioned offices	Orientation	Volume [m^3]	Floor area [m^2]	Thermal power [W_{th}/m^2]	Thermal power [W_{th}/m^3]	Persons [Nº]	Lights [36W each]	Computers [Nº]
Office 3.02	Northeast (NE)	58.2	19.1	109.9	36.1	1	8	1
Office 3.03	Northeast (NE)	46.8	15.4	115.4	37.6	1	4	1
Office 3.09	NE + Southeast (SE)	100.6	31.6	99.4	31.2	2	8	2
Office 3.10	SE + Southwest (SW)	131.7	46.2	90.9	31.9	2	12	5
Office 3.11	Southwest	53.3	20.5	76.6	29.5	1	4	1
Office 3.12	Southwest	107.0	41.1	76.4	29.3	3	8	3
Office 3.13	Southwest	49.4	19.0	82.6	31.8	1	4	1
Office 3.14	Southwest	56.3	18.5	84.9	27.9	1	4	1
Office 3.15	Southwest	44.8	15.5	101.3	35.0	1	4	1
Office 3.16	Southwest	57.0	18.8	83.5	27.5	1	4	1
Office 3.17	Southwest	107.1	41.1	76.4	29.3	3	8	3
Office 3.18	Southwest	71.6	27.5	76.4	29.3	1	8	1
Office 3.19	Southwest	51.0	19.6	80.1	30.8	1	4	1
Office 3.20	SW + Northwest (NW)	118.3	39.4	79.7	26.5	3	12	3
Office 3.21*	Northwest	44.4	14.7	106.8	35.4	-	2	-
Office 3.26	Northwest	60.7	20.1	156.2	51.7	1	8	1
Office 3.27	NW + NE	131.4	41.4	101.4	32.0	4	12	4
Office 3.28	NE + SE	86.2	26.8	156.7	48.7	2	8	2
Office 3.29	Southeast	57.5	18.9	166.1	54.6	2	4	2
Office 3.30	Southeast	41.6	13.6	154.4	50.5	1	4	1
Office 3.31*	Southeast	59.7	19.6	160.2	52.6	-	4	-
Office 3.36	Northeast	83.4	27.2	115.4	37.6	1	8	1

** These offices are currently file rooms.*

As it can be observed in Table 3.2, the thermal power installed by office volume varies from a minimum of 26.5 W_{th}/m^3 to a maximum of 54.6 W_{th}/m^3, which implies that some offices achieve the temperature comfort more quickly than others.

3.2 - GSHP SYSTEM DESCRIPTION

The GSHP system is constituted by seven ground heat exchangers or Borehole Heat Exchangers (BHE), the heat pump, three circulation pumps, piping networks, 33 fancoils and a buffer tank. A simplified schematic of the hydraulic system is shown in Figure 3.6. The external loop is a closed loop between the seven BHE and the heat pump and includes an external circulation pump (CPext). The internal loop is composed by two hydraulic circuits. The first circuit (internal loop 1) is the loop between the heat pump, the buffer tank and the decoupling tank and includes the internal circulation pump 1 (CPint 1). The second circuit (internal loop 2) is the loop between the decoupling tank and the 33 ceiling fancoil units installed in the 2nd floor and includes the internal circulation pump 2 (CPint 2). The decoupling tank serves to stabilize the pressure between the two internal loops and to separate demand and supply. The smallest internal circulation pump (CPint 1) serves to ensure that there is no lack of flow rate to the HP, regardless of the number of fancoils connected. A buffer tank with a volume of 1000 liters is installed on the supply line to the fancoils to smooth the HP start-up.

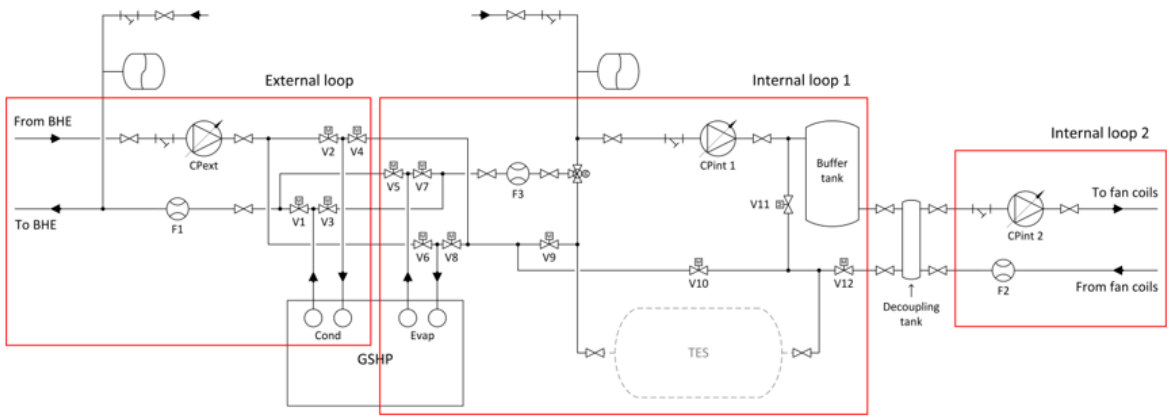


Figure 3.6: Schematic of the hydraulic system.

Although the Thermal Energy Storage (TES) tank was not installed, the hydraulic circuit was designed with all the necessary devices to allow its implementation and operation.

The GSHP system is a reversible system, which means that the same system can be used for cooling or heating the building. This reversibility was initially implemented by four 3 way motorized valves, which were replaced by eight 2 way motorized valves last summer, as represented in Figure 3.7. It was detected that the initial valves had small leaks with some influence of the cold water circuit on the temperature of the hot water circuit, leading to an efficiency loss.

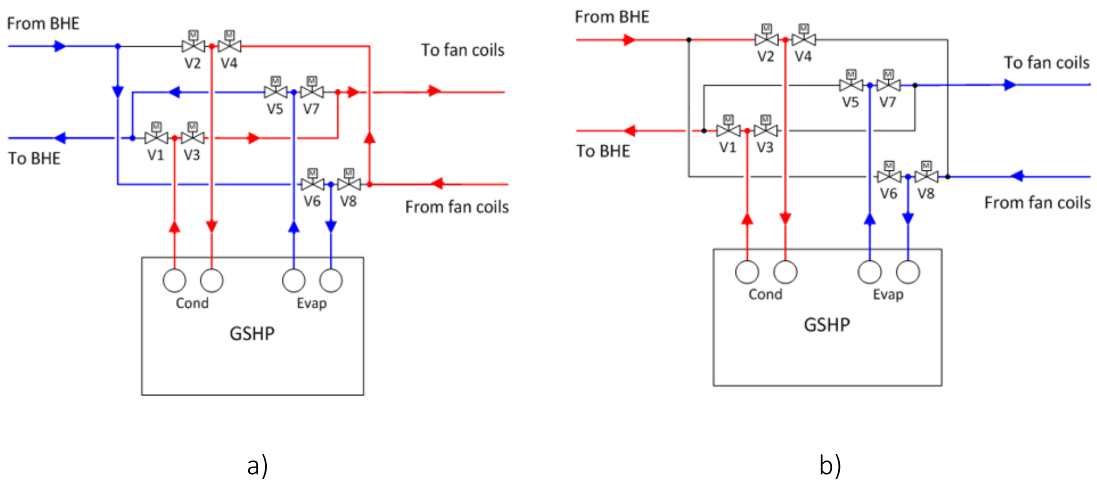


Figure 3.7: GSHP in: a) Heating mode; b) Cooling mode.

In the heating mode (Figure 3.7a), the heat source of the GSHP is the ground water loop (the evaporator is linked to the BHE) and the heat sink is the building (the condenser is linked to the inside water loop). The heat is extracted from the ground and released inside the building. The external circulation pump (CPext) is at the evaporator side and the internal circulation pump 1 (CPint 1) is at the condenser side. In the cooling mode (Figure 3.7b), the opposite happens: the external circulation pump is at the condenser side and the internal circulation pump 1 is at the evaporator side. In this case, the heat is extracted from the building (the evaporator is linked to the fancoils) and released into the ground (the condenser is linked to the BHE). Figure 3.8 shows pictures of the machinery room with the HP, the reversibility devices inserted in the hydraulic circuit above the HP, the decoupling tank and the buffer tank.

Figure 3.8: Pictures of the HP and associated equipment machinery room.

3.2.1 - GSHP CHARACTERISTICS

The GSHP advanced prototype installed was supplied by CIAT - Compagnie Industrielle d'Applications Thermiques (Ground-Med project partner) and has the following characteristics:

- The HP refrigerant used is the R410A.
- The two compressors are on/off tandem compressors to achieve a superior efficiency, particularly in partial load situations, and robustness.
- The HP has an heating capacity of 63.5 kW and a cooling capacity of 70.4 kW.
- The heat exchangers have been especially selected (brazed plates) to achieve high performance.
- Designed heating temperature of 40 °C in water supply and 35 °C in water return.
- Designed cooling temperature of 10 °C in water supply and 15 °C in water return.
- Reversibility is done on the water side by motorized valves, in order to be always in counter current on the heat exchangers.
- The HP has its own controller. Based on the temperature difference between HP set point and water return temperature, the controller activates or deactivates one or both compressors.

The performance of the heat pump prototype was tested at the CIAT Culoz research center, according to the Eurovent conditions, as presented in Table 3.3 and Table 3.4. The results supplied by the manufacturer are shown in Table 3.3 for the cooling mode and in Table 3.4 for the heating mode.

Table 3.3: Heat pump full load performance test results for cooling mode obtained by CIAT.

Conditions	Cooling mode						
	Evaporator		Condenser		Cooling capacity [kW]	EER	Eurovent Class A
	Inlet [°C]	Outlet [°C]	Inlet [°C]	Outlet [°C]			
Eurovent (CIAT)	12	7	30	35	56	5.05	>= 5.05
Coimbra demo site	15	10	25	30	63.5	6.19	–

Table 3.4: Heat pump full load performance test results for heating mode obtained by CIAT.

Conditions	Heating mode						
	Evaporator		Condenser		Heating capacity [kW]	COP	Eurovent Class A
	Inlet [°C]	Outlet [°C]	Inlet [°C]	Outlet [°C]			
Eurovent (CIAT)	10	–	40	45	61	4.58	>= 4.45
Coimbra demo site	15	–	35	40	70.4	5.65	–.

3.2.1.1 - GSHP OWN CONTROL

The heat pump has its own controller developed by CIAT with restricted parameterizations that only CIAT can access. The control of the GSHP is based on the water return temperature. The control temperature sensor is located on the condenser inlet for the heating mode and on the evaporator inlet for the cooling mode. The internal electronic module controls the operation of the compressors. Thus, depending on the temperature difference (deltaT) between the HP set point temperature and the return water temperature (cold or hot water), the electronic module will activate or deactivate the compressors in series.

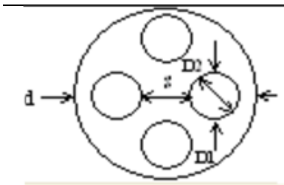
By default, one compressor is activated when the deltaT is 2 °C and after 60 seconds, if the deltaT is higher than 3.5 °C, the second compressor is activated. Each stage is turned ON at 60 seconds intervals and turned OFF at 1 second intervals (CIAT, 2010a). The second compressor is disabled when the deltaT is decreased to 1.5 °C and the first one is disabled when water return temperature achieves the HP set point temperature (deltaT = 0). For example, in the heating mode for an HP set point of 40 °C, the first compressor will be activated for a water return temperature of 38 °C, and after 60 seconds if the water return temperature is less or equal to 36.5 °C the second compressor will be activated. The second compressor will be turned off when the water return temperature achieves 38.5 °C and the first compressor will be turned off when the HP set point is achieved. The heat pump only starts 90 seconds after the operation conditions are achieved and the external circulation pump has received the control signal to be activated.

3.2.2 - BOREHOLE HEAT EXCHANGERS

The ground loop is composed by seven double U vertical borehole heat exchangers, each one with 125 meters of depth, which means that in each borehole there are two inlet pipes and two outlet pipes. The first borehole was a test borehole, which was used to determine the soil characteristics. The soil characteristics founded in the test borehole were alluvium granite (from 2 meters to 7 meters of depth), shale sediment with 20% quartz (from 7 meters to 55 meters of depth) and marl saturated with water (from 55 meters to 125 meters of depth). The groundwater level is 5 meters from surface and it was found along the entire drilling (Geominho, 2010). The Earth Sciences Department of the University of Coimbra estimated a conductivity of 2 W/(m.K) and a volumetric heat capacity of 2484 kJ/(K.m^3).

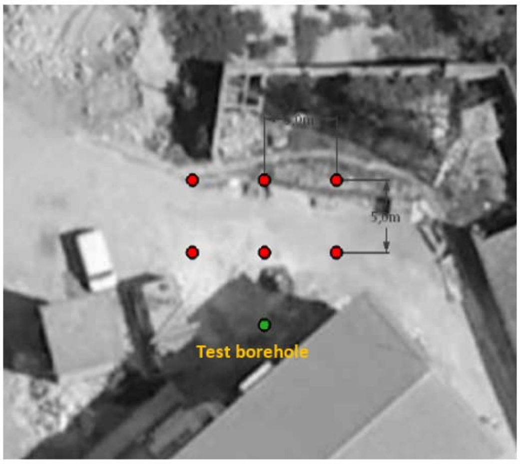
The diameter of the boreholes and of the pipes and the distance between the pipes within the borehole are presented in Table 3.5.

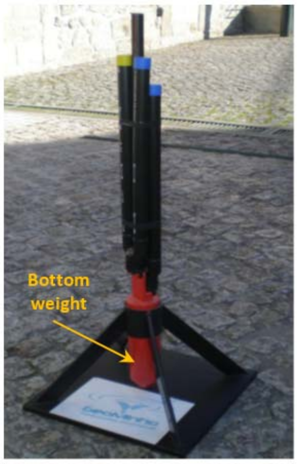
Table 3.5: Borehole Dimensions.



Double U Tube	Type of dimension	Dimension [mm]
	Borehole diameter (d)	140
	Pipe internal diameter (D2)	32
	Pipe external diameter (D1)	35
	Shank spacing (s)	56

The other six boreholes are arranged in a 5x5 meters rectangular grid as represented in Figure 3.9a), which corresponds to a minimum 5 meters of distance between each borehole. In each group of double U tubes, a weight of 12.5 kg was placed at the bottom, to guarantee its upright, and separators were mounted every 3 meters of depth, in order to ensure that the way tubes do not touch each other (Figure 3.9b).





a) b)

Figure 3.9: Boreholes heat exchangers: a) Boreholes position; b) Double U tubes with weight at the bottom.

The material used in the pipes is high density polyethylene (HDPE). Each borehole was filled with graded sand with a thermal conductivity of 2.4 W/(m.K). The depth between the earth surface and the top of each borehole is about 0.8 meters. The horizontal pipes that connect the seven BHE to the collectors and to the building are buried between 0.5 and 0.8 meters of depth from the surface and are insulated.

Figure 3.10: The seven pair of tubes connected to the collectors.

The boreholes are connected in a balanced parallel configuration and equalizing valves are used to guarantee a uniform flow rate distribution to the BHE (Figure 3.10). The working fluid used is plain water, since negative temperatures are not achieved. Some views of the drilling and installation of the borehole heat exchangers are presented in Figure 3.11.

Figure 3.11: Views of the drilling site.

3.2.3 - CIRCULATION PUMPS

There are three circulating pumps installed. Each loop, as described in the beginning of this section, has a circulation pump. High efficiency permanent magnet motor pumps with variable speed control from Grundfos were chosen (Figure 3.12).

Figure 3.12: Images of the circulation pumps

The circulation pumps set point (pressure head in meters of water) can be adjusted manually in a range of 1 to 10, in the control panel. The circulation pumps can be started or stopped by an external signal of 5 V with a connection to the digital start/stop input of the circulation pump.

In order to be possible to change the circulation pumps set point by an external signal, an expansion module (GENI module) was installed in each circulation pump, during summer 2014. These modules have an input for an external 0-10 V analog signal, which allows controlling the set point or speed between the minimum curve set point and the set point selected in the control panel. For an input voltage lower than 0.5 V, the pump will operate according to the minimum curve, as represented in Figure 3.13.

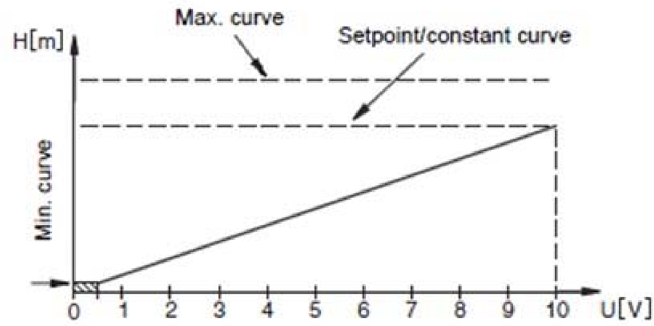


Figure 3.13: Control of pump set point or speed via an external 0-10 V signal (Grundfos, 2009).

The previous circulation pumps (Magna model) were substituted in October 2014 by new ones supplied by Grundfos, which are an updated model with more control capabilities and that can be directly commanded by external signals without additional hardware.

3.2.4 - FANCOIL UNITS

The heating/cooling distribution system in the building consists of 33 ceiling “Coadis 2” fancoil units (23 units of model 235/22 and 10 units of model 235/33), also manufactured by CIAT, installed in 22 offices. These fancoil units (Figure 3.14) were selected due to their high heat-exchanger efficiency, high performance of the fan and high efficiency of the brushless permanent magnet motor with variable speed control.

Figure 3.14: Fancoil units.

The fancoils can be regulated by means of a wall mounted thermostat, allowing users to select the desired comfort temperature. Besides the adjustment of the desired temperature, these thermostats only allow to adjust the neutral zone (allowed temperature fluctuation before fancoils start running again). The fan speed is controlled automatically by means of a 0-10 V control signal, depending on the differential between the room temperature and the set point. Voltage limiters were installed in the fancoil units to prevent excessive noise from the operation of the fans when the control voltage (0-10 V) was above 6 V, since at the maximum velocity (10 V) users feel some discomfort.

According to the fan maximum and minimum velocities, the useful heating capacity of the fancoils model 235/22 can vary from 1220 to 1840 W and the useful total cooling output can vary from 1270 to 1780 W. In the fancoils model 235/33, the useful heating capacity can vary from 1480 to 2620 W and the useful total cooling output can vary from 1660 to 2560 W (CIAT, 2010b).

3.3 - MONITORING SYSTEM

A solution for a Data Acquisition (DAQ) and monitoring system was developed, under the Ground-Med Project, by the Institute of Systems and Robotics - University of Coimbra (ISR-UC), to be implemented in the eight demonstration sites. All the information is stored in a database and can be accessed easily through a web interface. Under the Ground-Med project, mandatory and optional variables measurements were defined.

The electric power consumption of the three circulating pumps, the fancoils and the heat pump, is measured by electronic power meters from Carlo Gavazzi. The thermal power, the flow rate, the input temperature and the output temperature of each water loop are measured by thermal energy meters from Brunata, composed by a magnetic flowmeter and two temperature probes. In this way, it is possible to know how much thermal energy comes from boreholes and how much thermal energy is delivered to the building. These measurements are essential to determine the Seasonal Performance Factor (SPF) that should be calculated within the project, in order to assess the system performance. Outdoor and indoor temperatures are measured by PT100 sensors and the solar radiation by a pyranometer. The Coimbra pilot data acquisition system and the sensors used are represented in Figure 3.15.

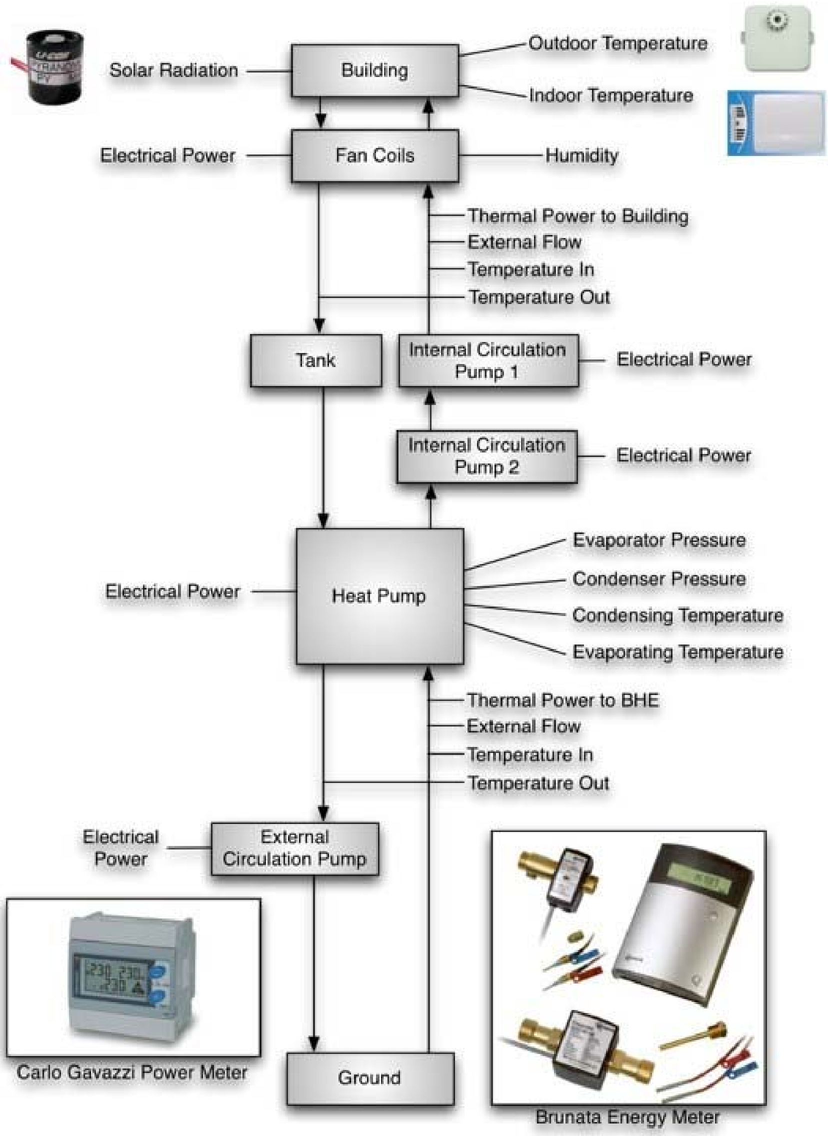


Figure 3.15: Coimbra demo-site monitoring system and sensors.

The DAQ system architecture with processing capabilities and a wide range of input/outputs connections for data collection was designed based on National Instruments “CompactRIO (cRIO)” hardware, under the Ground-Med project. The “CompactRIO” is an integrated real-time controller, which allows the integration of several input/output modules, as represented in Figure 3.16. The National Instruments “CompactRIO” is a high reliability programmable automation controller with a modular design and was chosen since it could meet all the functional requirements defined under the Ground-Med project.

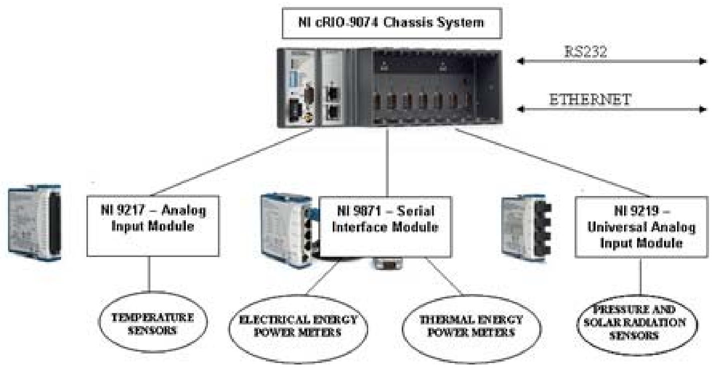


Figure 3.16: DAQ system architecture.

The cRIO-9074 provides ethernet connections suitable for Internet connections, and also allows programming with LabView. One of the cRIO-9074 ethernet ports is used to send periodically data through a FTP (File Transfer Protocol) connection to the web server. The RS232 connection port is used for communication tasks. The cRIO-9074 software runs within the board, independently of any external computer, and communicates with all the different energy meters and sensors to make all the required measurements, deploying all data into a web server. In order to see the measurements being done by the cRIO-9074, a measurement utility was developed just to read the variables and no configuration is allowed (Figure 3.17).

In the Coimbra demo-site, an additional thermal energy meter (Brunata 3) was installed in March 2014, in order to provide more conclusive results about the experimental tests, for adjusting some key parameters to improve the GSHP system performance. The thermal meter (Brunata 2) measures the thermal energy delivered to the building and is located after the buffer tank, which means that quite often the thermal energy produced by the heat pump, at the end of the operation period, is not delivered to the building being stored in the buffer tank. The sensor localization, namely the temperature probes, was carefully analyzed to guarantee that this energy meter could also measure the thermal energy associated to the TES tank charging and discharging, if this tank will be installed in the future.

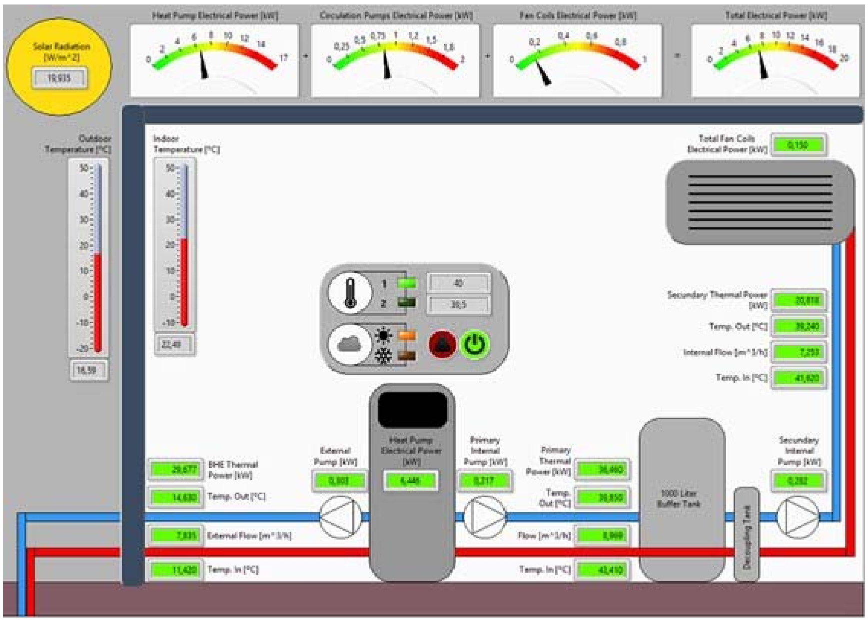


Figure 3.17: Coimbra demo-site measuring utility.

3.4 - CONTROL SYSTEM

Initially, under the project, the GSHP system, namely the heat pump and the circulation pumps, was planned to work continuously and the fancoils were switched manually by end users when they arrived to their offices. However, users complained that the GSHP system took longer time to achieve temperature comfort than the existent HVAC system. This situation was verified by the analyses of the offices temperature. Taking into account the users thermal comfort and the need of avoiding the night consumption of the GSHP system, a control system was developed in order to program the daily operation period.

Since the measuring system required for the Ground-Med project is using a powerful equipment like the National Instruments cRIO-9074, this technology was used to perform the overall system control with a low investment in extra hardware. The cRIO-9074 mainframe allows the installation of additional I/O modules, which are used to control external relays and therefore the start/stop operation of equipment such as the heat pump, the circulation pumps and the fancoil units.

This control schedule was implemented in late December 2013 allowing the pre-heating of the building in the morning, the automatic shutdown at the end of the day and the total system shutdown during weekends automatically, resulting in almost zero consumption (only the standby consumption) during nights and weekends. This is a very important feature to allow the implementation of load shifting strategies in order to minimize the operation costs and to guarantee temperature comfort when users arrive. The schedule control developed in LabView is represented in Figure 3.18.

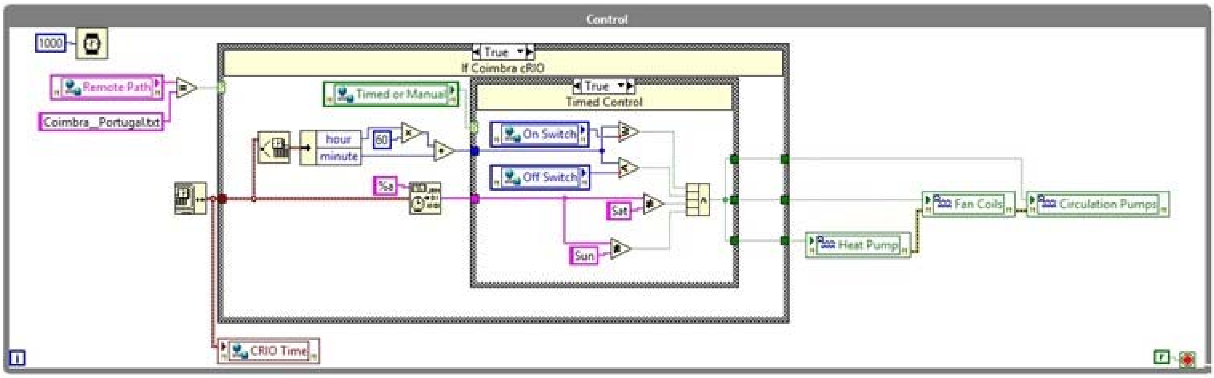


Figure 3.18: Schedule control developed in LabView.

Further capabilities were developed in the control system, besides the schedule programming, such as:

- Advanced start and delayed stop for circulation pumps. Circulation pumps are programmed to start 5 minutes before the HP operation starts and to stop 5 minutes after the HP operation stops. In the beginning, water starts circulating to give the real temperature of the water within the pipes. In the end, after the HP shutdown, by maintaining water in circulation it is possible to store this thermal energy in the buffer tank.
- Adjustment of the circulation pumps head in a range of 0-100% of the maximum pump head, by means of a 0-10 V control signal. This new control capability was essential to perform the pump velocity tests variation, in order to improve the GSHP system performance.
- Adjustment and selection of the GSHP set point for cooling and heating. All external control of the HP is done by the RS485 port communication.
- Selection of the GSHP operation mode: heating or cooling. The motorized valves, responsible by the reversibility of the system, are automatically switched to the corresponding position, according to the selected operation mode. There is a relay inside the HP that, according the selected operation mode, transmits a signal for an external relay that controls the switch of the valves position. Based on the same HP signal, the external changeover relay switches fancoils to operate in cooling or heating mode.

The parameters previously referred can be adjusted in a centralized and remotely way by the control display presented in Figure 3.19. This feature helped to overcome the restricted access to the building and to avoid the need to go to the building every day to adjust parameters.

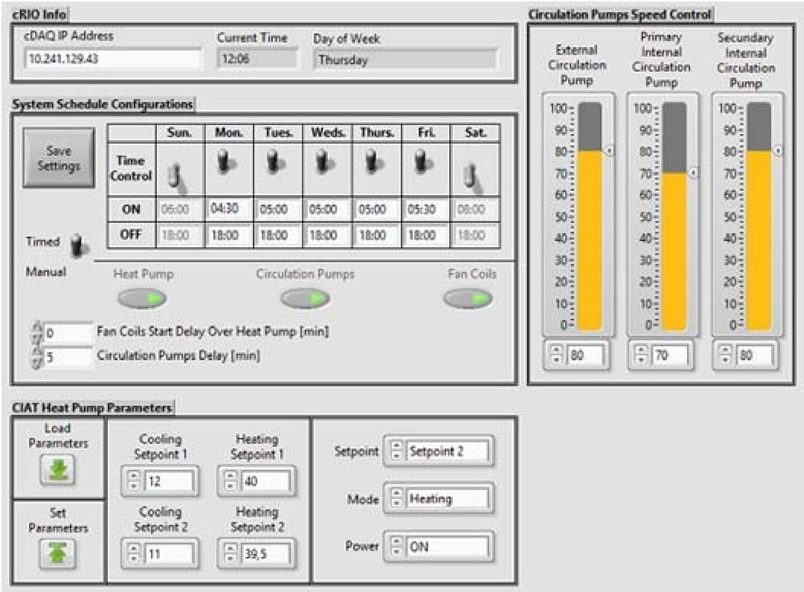


Figure 3.19: GSHP system control display.

The external circulation pump operation is linked to the heat pump compressor(s) in order to minimize the electricity consumption. When the conditions to turn ON the compressors are achieved, an electric signal is given from the HP to the circulation pump to start running and only after past 90 seconds, the compressors start working. The basic control of the system operation is to guarantee that the external circulation pump will always operate together with the heat pump compressor(s) and that the internal circulating pumps are always working, during the defined working period of the heat pump, in order to allow that heat pump always receives the information about the required return water temperature to meet the thermal needs.

The offices set points are only controlled by the end users, which results in a certain limitation to implement load shifting strategies, like to adjust the room temperature set point according to the electricity prices. In this specific installation, a centralized control of the rooms set points is very difficult to implement, since no communication connections were provided between the 22 wall controllers (located in the 2nd floor) and the control system (sited in the basement). The fancoils operation period has been controlled by the power cut off and not by control signals.

3.5 - STORAGE SYSTEM

As already referred, the peak load of the building in Winter occurs in the morning and is coincident with the national morning peak load demand period, when the electricity is about three times more expensive than in off peak periods. Preheating strategies associated with thermal energy storage are a good solution to minimize the costs operations with space heating, since the heating system can be use during cheaper electricity periods to charge thermal storage systems and to preheat the building.

Initially, it was planned to couple a thermal storage tank based on Phase Change Materials (PCMs) to the GSHP system, but due to external reasons it was not possible to install it. This process decision took about three years to be solved and this winter the lack of space was indicated as the main reason to not allow the installation of the tank. Nevertheless, and since some relevant research was already done, it was considered important to present here the work already developed, which can be used on a future occasion.

3.5.1 - THERMAL STORAGE CAPACITY

Since in Portugal the heating season is not very long, varying typically from 16 to 22 weeks, a full storage strategy was not considered, because it requires higher investment and higher space for the installation, and in this particular case, which is an office building, the heating system works, at maximum, during 2.5 peak hours per day. A partial storage strategy to meet at least the thermal needs during peak periods of electricity seems a more appropriate option.

The partial thermal storage capacity was determined based on the daily average thermal energy supplied to the 2nd floor, during the morning peak period (9 a.m. to 10:30 a.m.), without considering preheating time. After some simulations, using the thermal load profile of the 2nd floor in typical winter weeks, it was possible to determine that the necessary thermal capacity to avoid that GSHP operates during the peak period is about 70-90 kWh (Figure 3.20). Using preheating strategies, this capacity can be reduce to 40-60 kWh, depending on the preheating period.

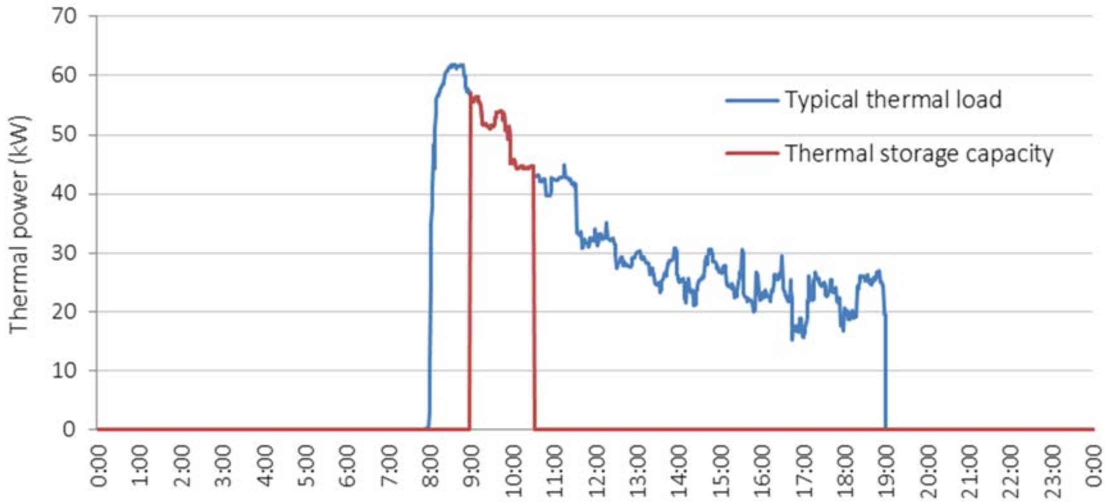


Figure 3.20: Average thermal load profile of the 2nd floor during a typical winter week.

3.5.2 - SELECTION OF THE PCM TECHNOLOGY

The desired operating temperature range is one of the most important criteria for the selection of phase change heat storage materials, since the heat transfer rate in the thermal storage unit and thus the performance of the system mainly depends on the difference between the heat transfer fluid temperature and the melting point of PCM (Agyenim et al., 2010). Other two main desirable characteristics for the PCM selection are the high latent heat of fusion per unit of mass (storage capacity per unit of mass) to minimize the tank dimensions and to ensure high thermal conductivity to guarantee good heat transfer rates.

Taking into account that the GSHP and the fancoils were designed to operate in the space heating loop with a water supply temperature of 40 °C and a water return temperature of 35 °C, and that it is needed at least a temperature difference between water and PCM material of 5 °C to charge and discharge the PCM, the ideal melting point should be within the range of 44-48 °C, ideally 45 °C.

Since the thermal storage system was to be installed in a real building, the use of PCM commercial solutions already encapsulated and used in others buildings in Europe was considered the best choice, in order to have some guarantee that the system can work properly without any risk to the space conditioning system. Table 3.6 presents a comparative analysis of the commercial PCMs within the desired range temperatures.

Table 3.6: PCMs available in the market within the desired melting point range.

PCM name	Type of Product	Melting Point [°C]	Heat of Fusion [KJ/kg]	Specific Heat [kJ/(kg.K)]	Thermal Cond. [W/(m.K)]	Encapsulation	Supplier
RT42	Paraffin	41	174	2.1	0.2	CSM Panel (Rectangular)	Rubitherm (Germany)
A42	Paraffin	42	230	2.20 – 2.22	0.21 – 0.2	Ball-Ice (Spherical)	PCM products Ltd (UK)
S44	Salt hydrate	44	105	1.61	0.43	Tube-Ice (Cylindrical)	PCM products Ltd (UK)
S46	Salt hydrate	46	190	2.41	0.45	Tube-Ice (Cylindrical)	PCM products Ltd (UK)
Climsel C48	Sodium Sulphate, Water and Additives	48	226.8	3.6	0.5-0.7	-	Climator (Sweden)
Latest™48S	Salt hydrate	48	>230	2	0.6	Panels / balls	PCM Energy P. Ltd (India)

Looking to the solutions which present melting points temperatures closer to 45 °C, the S46 salt hydrate from PCM products Ltd seems to be the most suitable candidate for the required temperatures operation. Paraffins have a very low thermal conductivity, about 2 or 3 times lower than salt hydrates. The S44 salt hydrate has a lower heat of fusion (105 kJ/kg) than the S46 salt hydrate, which requires higher storage volume. Although the Climsel C48 solution from Climator presents better characteristics, it requires a higher water supply temperature (around 53 °C).

Since it was possible to have a sample container filled with the S46 material, it was asked to the Chemical Process Engineering and Forest Products Research Centre of Coimbra University (CIEPQPF) to do some tests. They concluded that the S46 material has a composition very similar of the sodium thiosulfate pentahydrate ($Na_2S_2O_3.5H_2O$) and that the melting point was 46.9 °C, which is not very different from the manufacturer specifications (CIEPQPF, 2010).

They also tried to test the material stability with Differential Scanning Calorimeter (DSC), but the sample used only allowed 30 cycles (melting and solidification) as it can be observed in Figure 3.21. It should be noted that the sample had only 5 mg and was removed from a sealed container of 6 kg, not being possible to assure that it is a representative sample or if the material characteristics didn't change with the opening of the container.

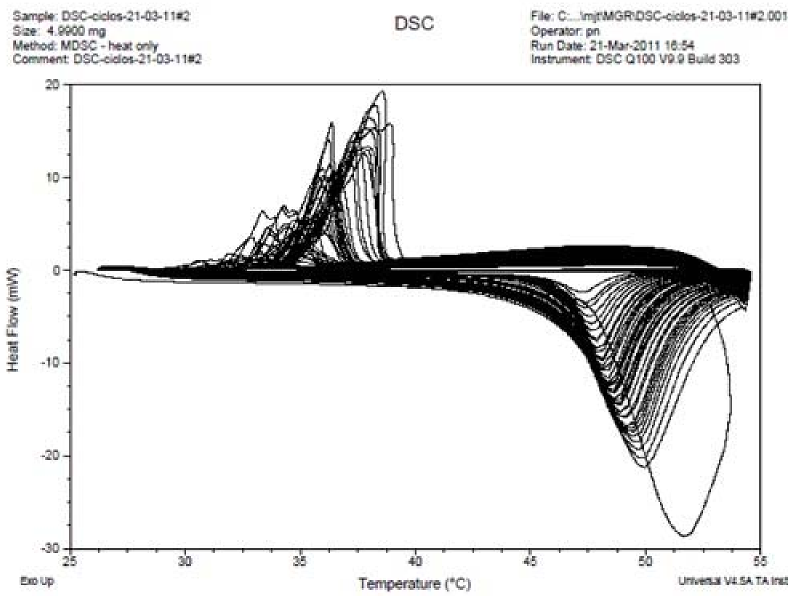


Figure 3.21: Differential Scanning Calorimeter (DSC) analyses performed by CIEPQPF.

These commercial PCMs are already available in plastic containers that can be stacked on top of each other forming a self-assembling large heat exchanger within the tank. Salt hydrates are available in rectangular or cylindrical containers, as presented in Figure 3.22.

Figure 3.22: Plastic containers filled with PCM (PCM Products, 2011).

3.5.3 - TES TANK DESIGN

The initial idea was to install a useful thermal storage capacity of 100 kWh in order to have higher flexibility to perform experimental tests. Since the thermal storage tank was planned to be installed in a room adjacent to the HP machinery room, the restrictions associated to this space were also taken into account for the final dimensions of the tank. A thermal capacity of

100 kWh, taking into account those space restrictions, would only be possible to install with a rectangular tank geometry. Considering the rectangular PCMs container dimensions and since 0.5 meters for each tank header is necessary to provide uniform flow across the section tank, the tank should have a length of 3 meters, a width of 0.8 meters and a height of 1.5 meters as internal dimensions. To ensure a thermal storage capacity of 100 kWh it would be necessary about 400 rectangular PCM containers. According to the manufacturers, 40% of the tank volume must be occupied by the heat transfer fluid (water) to guarantee a good transfer heat rate (PCM Products, 2011). The resulting tank would have a total volume of 3.5 m^3.

However, after consulting experts and manufacturers and since the tank is a pressurized one, they indicated that to guarantee enough strength, the tank would need to be extremely reinforced. Strengthening the tank would make it very heavy and difficult to install inside the building (it should pass in corridors and through the room door), as well as much more expensive. In this way, it was decided to install a cylindrical tank. The same dimensions were considered, with 3 meters of length and a diameter of 0.8 meters. This cylindrical configuration reduced the possible thermal capacity to install in half. The new tank would have a volume of 1.5 m^3, being possible to install inside it 352 cylindrical containers filled with PCMs, corresponding to a thermal storage capacity of around 52 kWh. Still, experimental tests could be performed and GSHP operation during peak hours could be avoided if building preheating strategies were used. Figure 3.23 shows the space required for the installation of the cylindrical tank.

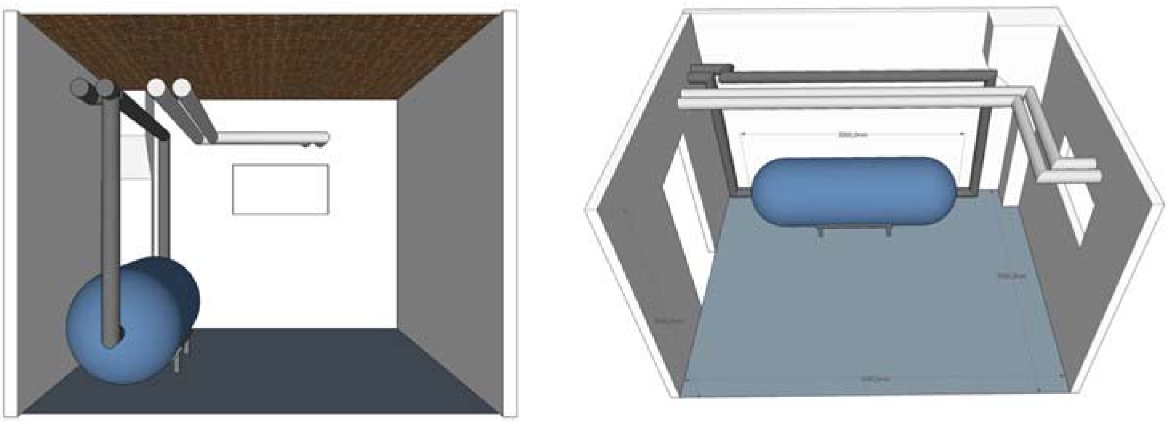


Figure 3.23: Space required to install the cylindrical TES tank within the adjacent room to the machinery room.

3.5.4 - GSHP COMBINED WITH TES OPERATION MODES

As already explained, the storage system intends to be used as a partial storage strategy to meet thermal needs during the day with priority during peak demand periods. Depending on the thermal needs of the day, the tank thermal storage capacity may be totally released during the 1.5 hours of the peak period or it may still be used in the next period to fully discharge the tank. In this way, the system may operate in the following different modes:

1) Charging of the thermal storage tank by the GSHP during night, in the special off peak period (between 2 a.m. and 6 a.m.);
2) Building preheating directly by GSHP during special off peak period and off peak period (6 a.m. to 8 a.m.), depending on the climate conditions;
3) Space heating directly by GSHP during the 1st hour of the middle peak period (8 a.m. to 9 a.m.);
4) Space heating directly by the storage tank during peak period and subsequent use of the thermal energy still available within the tank (9 a.m. to 10:30 a.m.);
5) Space heating directly by GSHP until the second peak period of the day (10:30 a.m. to 6 p.m.);
6) Shutting down the GSHP system at 6 p.m., one hour earlier than the end of the working period, to profit from the building thermal mass.

The operation modes corresponding to the charging and discharging of the TES tank are represented in Figure 3.24 and Figure 3.25, respectively.

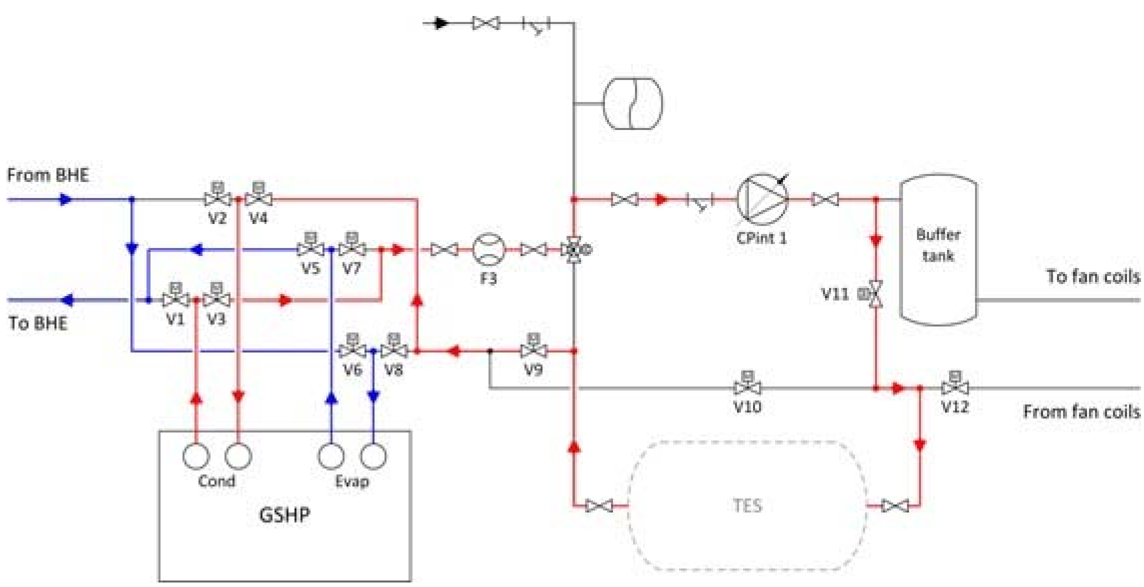


Figure 3.24: Charging of the thermal storage tank by the GSHP.

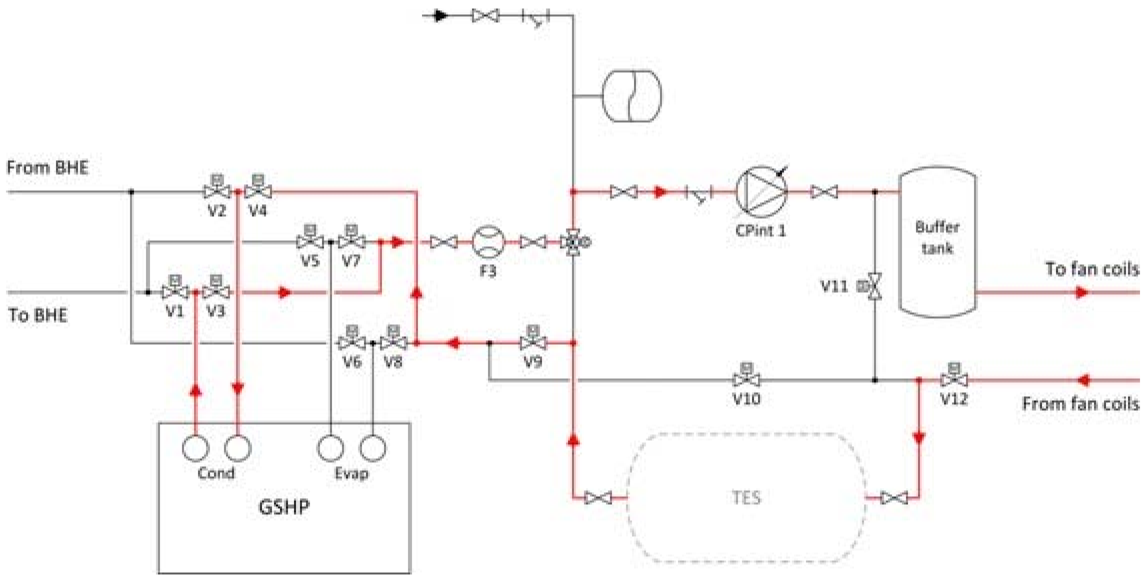


Figure 3.25: Space heating directly by the thermal storage tank (discharging of the TES tank).

Although the hydraulic circuit allows performing a bypass to the HP, it is thought that the passage by the HP could bring benefits. With this configuration, HP may automatically be turned ON depending on the water temperature return, guaranteeing that there are no heating supply interruptions to the building.

As already referred, this section 3.5 aims to be a starting point for future research, since it was not possible to install the proposed storage tank within the timeline of this thesis work.

CHAPTER 4

EXPERIMENTAL RESULTS AND PERFORMANCE ANALYSIS OF THE GSHP SYSTEM

This chapter starts by presenting relevant considerations about the GSHP system and the procedures used in the analysis of the results. The experimental tests and analyses carried out during three different seasons, in order to optimize the Ground Source Heat Pump (GSHP) system operation and performance, are also presented in this chapter. It was decided to present data and results by season, since the GSHP system had hydraulic and control changes over time, including correction of installation errors, replacement of the valves responsible by the heat pump reversibility, adjustment of the hydraulic circuit to test the geo-cooling, replacement of the circulation pumps, installation of additional monitoring and control devices, as well as evolution of the control system capabilities. At the end of this chapter a global analysis of the system is presented, including the ground temperature evolution over these periods and some concluding remarks.

4.1 - ANALYSIS CONSIDERATIONS AND PROCEDURES

4.1.1 - SEASONAL PERFORMANCE FACTOR CALCULATION

The efficiency of the GSHP system is assessed considering the Seasonal Performance Factors (SPFs) for the heating season and the cooling season, which corresponds to the average system performance for the entire season. Four different SPFs are calculated for each season, based on the monitored data of the thermal energy supplied to the building (thermal energy meter 2 – Brunata 2) and the electric energy consumptions of the Heat Pump (HP), External Circulation Pump (CPext), Internal Circulation Pumps (CPint1 and CPint2) and Fancoils (FCs) of the entire season. The SPFs were calculated using Equation (4.1), as defined in the Ground-Med Project (Ground-Med, 2014).

$$SPF_i = \int_0^t Q.dt \Big/ \int_0^t P_i.dt \tag{4.1}$$

being Q the thermal power supplied to the building, t the time interval and i the variable, between 1 and 4, that selects each SPF, as follow:

- SPF1: P1 = HP compressor power
- SPF2: P2 = HP compressor power + CPext power
- SPF3: P3 = HP compressor power + CPext power + (CPint1 + CPint2) power
- SPF4: P4 = HP compressor power + CPext power + (CPint1 + CPint2) power + FC power

The SPF1 measures the performance of the HP technology. The SPF2 gives the performance of the GSHP technology since it includes the circulation pump of the ground loop and can be used to compare the GSHP with other space heating technologies. The SPF3 includes the other auxiliary equipment used to distribute thermal energy to the building (circulation pumps) and the SPF4 gives the performance of the overall GSHP system, since it also includes the terminal units consumption.

Although SPF is related with seasonal performance, the analysis of some results is presented in daily or weekly time bases. The performance results can also be presented as Daily Performance Factors (DPFs) or as weekly SPFs.

4.1.2 - CIRCULATION PUMPS OPERATION IN HEATING AND COOLING MODE

As already referred in Section 3.2, the GSHP installed in the pilot building is reversible and the reversibility is done at the water side. In the heating mode, the external circulation pump (CPext) is at the evaporator side and the internal circulation pump 1 (CPint1) is at the condenser side. In the cooling mode the opposite happens, i.e., the external circulation pump is at the condenser side and the internal circulation pump 1 is at the evaporator side.

The HP, among others, has two important protections: one against the flow rate fault (evaporator side) and another one against overheat (condenser side). During the continuous system operation below a certain flow rate level, the HP gives “alarm” and after three faults the HP is automatically stopped for protection and requires local intervention. The GSHP has a minimum flow rate at the evaporator side (6.2 m^3/h) and a minimum flow rate at the condenser side (5.4 m^3/h) defined by factory setting.

This means that each circulation pump (CPext and CPint1) must be programed differently in winter and in summer (with different minimum operation points), depending if it is linked to the evaporator or to the condenser. In order to optimize the GSHP system performance, the circulation pumps set point variation will be limited by these minimums flow rates required, leading to different ranges of set point variations in cooling and in heating mode, also because the pressure drop of the ground loop is higher than the pressure drop of the internal loop 1. This issue will be further developed in sections related to the optimization of the circulation pumps operation in winter and summer (Sections 4.3.2 and 4.4.1).

4.2 - HEATING SEASON 2013/2014

The GSHP system worked in heating mode from 11 November 2013 to 11 April 2014, but the data is only presented after 25 November since some monitoring problems occurred. The experimental tests and analyses carried out in this winter season were mostly intended to understand the GSHP system operation, to verify the users receptivity and satisfaction related to this new space conditioning system and if the GSHP system was working properly, as explained along this section. Some results about system operation, energy savings and system performance improvement were also possible to achieve.

The main situations and developments occurred during this heating season, in a summarized way, were:

- The external circulation pump operation was linked to the HP compressors operation, in order to reduce energy consumption of auxiliary equipments.
- Monitoring of temperature in different offices was carried out, in order to assess the indoor temperature evolution after turning ON the fancoils. Initially, the GSHP worked 24 hours per day and fancoils were only turned ON when users arrived. This situation was not satisfactory to the users, since the GSHP system operates with lower temperatures than conventional heating systems and takes longer time to reach temperature comfort. Users chose to use the existing system rather than the GSHP system. To overcome this situation it was decided to let fancoils switched 24 hours per day until the daily control operation system was developed.
- The daily operation period control of the overall system presented in Section 3.5, was implemented and tested. This was the most important feature to implement preheating strategies to guarantee thermal comfort when users arrive, and to avoid the use of the existing HVAC system, in order to obtain reliable results without a 24 hours system operation. At the same time it allows that the higher electrical power demand happens during lower electricity prices, contributing to save costs operations (the assessment of costs savings by the application of load shifting strategies is carried out in Chapter 5).
- At this stage, with the daily operation control, reliable results were expected to be achieved. After analyzing the offices temperature and after carrying out further measurements with a thermal imaging camera, it was detected that some fancoils were not working properly. This situation led to the need of requesting and monitoring various technical interventions at the fancoils level. It was found that, in several fancoils, the water flow rate was limited due to existing bottlenecks in the pipes created by installation errors, namely incorrect thermal welding of pipes. After these interventions, all 22 offices became heated only by the GSHP system.
- Some experimental tests were implemented to assess the influence of some parameters in the GSHP system performance, such as the circulation pumps operation point (head position and corresponding flow rate) and the HP temperature operation (HP set point). For this purpose an additional thermal energy meter (Brunata 3) was installed to measure the thermal energy at the HP outlet. This issue is further explained below.
- The seasonal and weekly performance factors were calculated and analyzed and the results achieved are explained according to the parametrizations variation.

4.2.1 - ANALYSES OF THE GSHP SYSTEM AND MONITORED DATA

4.2.1.1 - ANALYSES OF FANCOILS OUTLET AIR TEMPERATURE

During this winter season it was detected a problem in the plumbing of some of the fancoils due to an incorrect installation. Some of the welds on the plastic tubes from the main plumbing of the internal circuit to the fancoils were, in some cases, total or partially clogged. A thermographic camera was used to identify all the clogged fancoil units to be repaired. By using this method, 12 of the 33 fancoils installed, were detected as total or partially clogged, requiring immediate intervention.

One of the examples is shown in Figure 4.1, corresponding to two different fancoil units installed in the same room: on the left, a working fancoil that did not deliver heat to the room because of a clogged pipe; on the right, a fancoil unit working in normal conditions. It should be noted that the thermal scale is the same, between 20 °C and 37 °C.

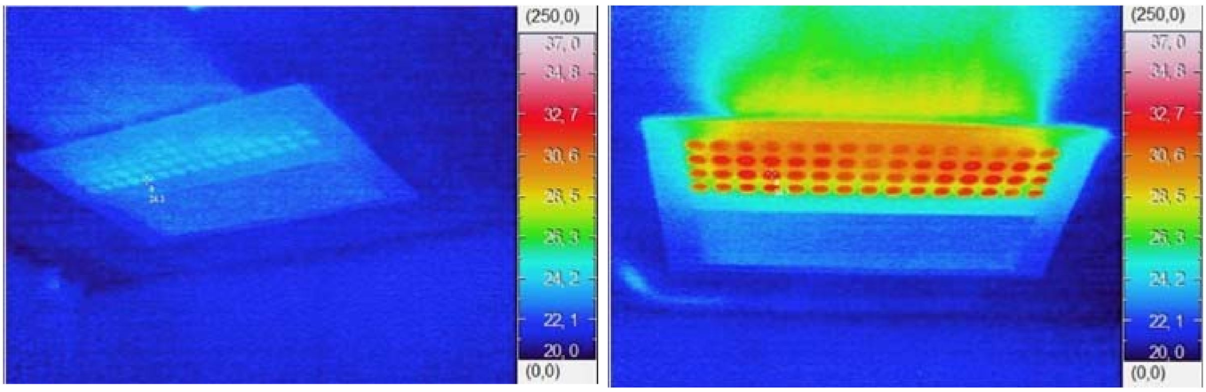


Figure 4.1: Infrared fancoil images.

These interventions occurred in middle December. After, to ensure that fancoils stay working properly, the thermographic camera was used again and it was detected that problems in some fancoils were not totally solved, leading to a new intervention in middle January. The biggest problem associated to the technical intervention at the water plumbing level is that air bubbles enter into the piping, requiring that all fancoils had to be purged one by one, in spite of the existence of air traps throughout the hydraulic circuit. After these two interventions, all fancoils started to work properly.

4.2.1.2 - CIRCULATION PUMP OPERATION LINKED TO HP COMPRESSOR

In order to reduce energy consumption of auxiliary equipments, the external circulation pump operation was linked to the HP compressors operation, as presented in Figure 4.2. This strategy could be applied in the internal loops, since water must be always in circulation to feed fancoil units and a minimum flow rate must be guaranteed so that HP works without damage risks, being more critical at the evaporator side.

The energy savings of the circulation pump operation linked to HP compressor depends on the operation period, on the load factor and on the circulation pump set point. In this case, savings of 11% were achieved in the external circulation pump consumption. The DPF2, DPF3 and the DPF4 increased 1.2%. This improvement can be more significant in days with lower thermal needs.

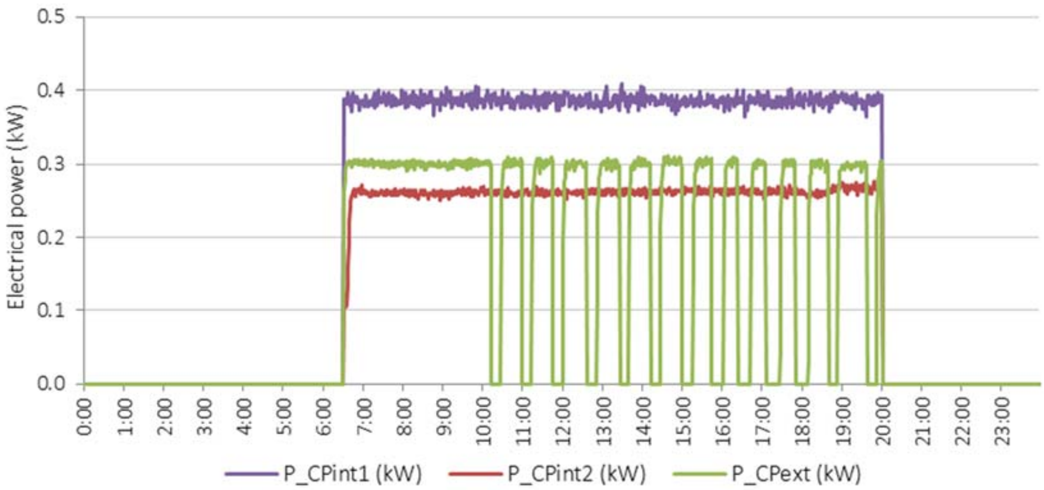


Figure 4.2: Circulation pumps operation modes: CPext (cycling) and CPints (continuous).

4.2.1.3 - GSHP PERIOD OPERATION AND ENERGY CONSUMPTION

Two different days, with different operation periods and with similar outdoor temperatures, were used to compare the electricity consumption. The main purpose was to determinate how much energy could be saved by setting an intermittent operation period (Figure 4.3) for the GSHP system, instead of the 24 hours operation period (Figure 4.4). With a 24 hours operation period the GSHP system has consumed around 125 kWh, while with an operation period of

14.5 hours (between 6:30 a.m. and 8 p.m.) the electric consumption was around 80 kWh, leading to energy savings of 35%.

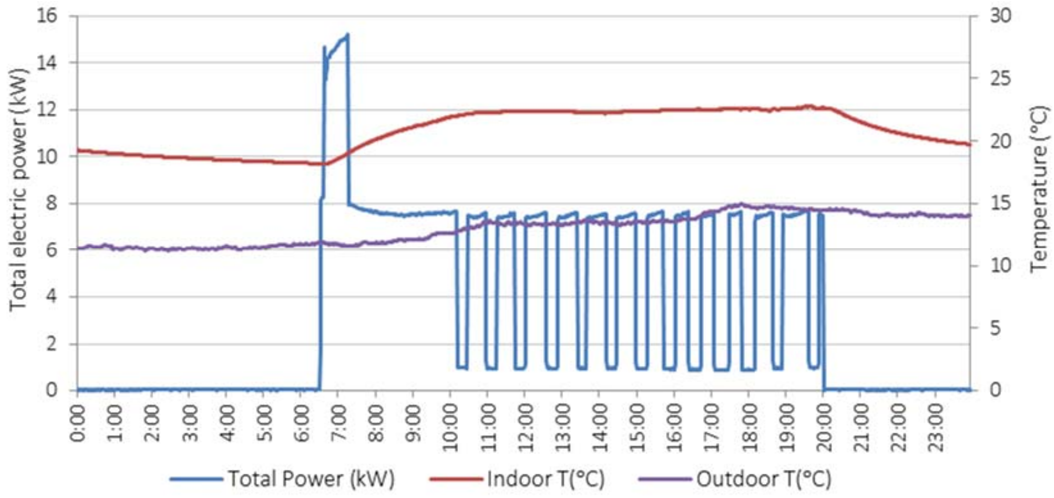


Figure 4.3: GSHP system electric load profile with an intermittent operation period.

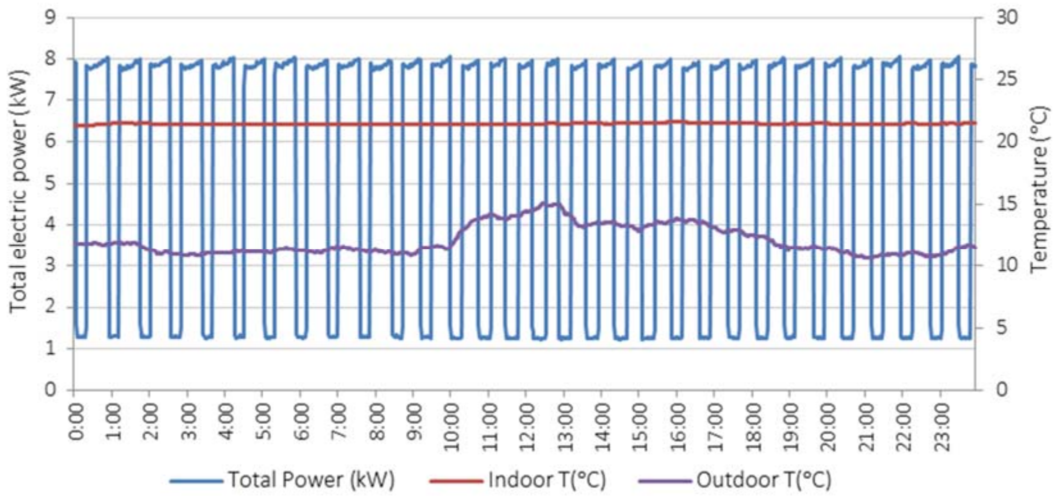


Figure 4.4: GSHP system electric load profile with 24 hours operation period.

4.2.1.4 - OFFICES TEMPERATURES

The indoor temperature represented in Figure 4.5 is the only office temperature that is monitored by the Data Acquisition (DAQ) system. Other temperature sensors were used to monitor other offices. Two different winter weeks are presented. The temperature evolution is very different from office to office due to many reasons such as the office orientation, the thermal power per square meter installed by office area, the temperature set point choosen by users, if users let door office open or closed, among others.

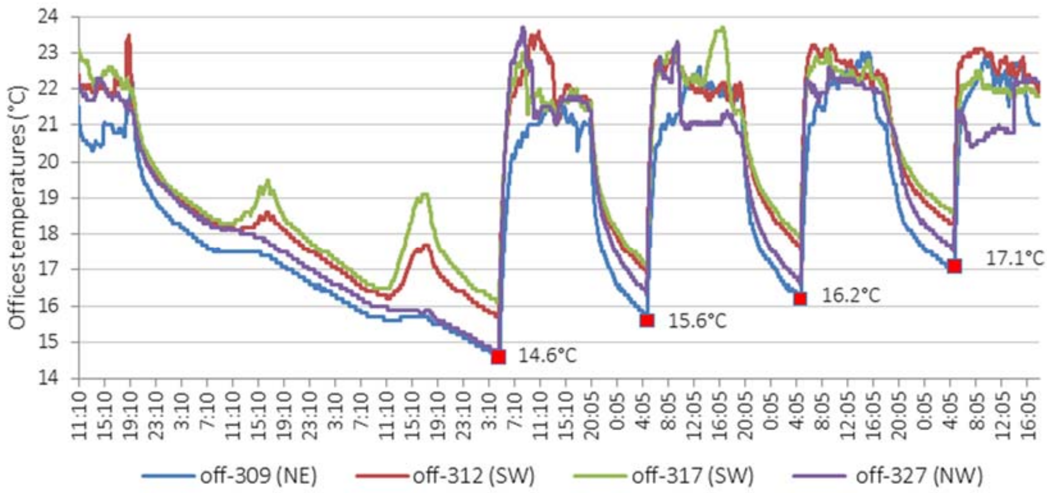


Figure 4.5: Offices temperatures in a winter week (14-21 February).

After the daily operation control implementation, different operation periods can be programmed. The offices temperature decreases differently from day to day, depending on the external weather conditions and on the amount of the stored energy in the building thermal mass during the day before. On Mondays morning, the offices temperatures are much lower than in other week days, since the building cooled down all the weekend. This means that the GSHP system should be turned ON earlier in Mondays than in other week days.

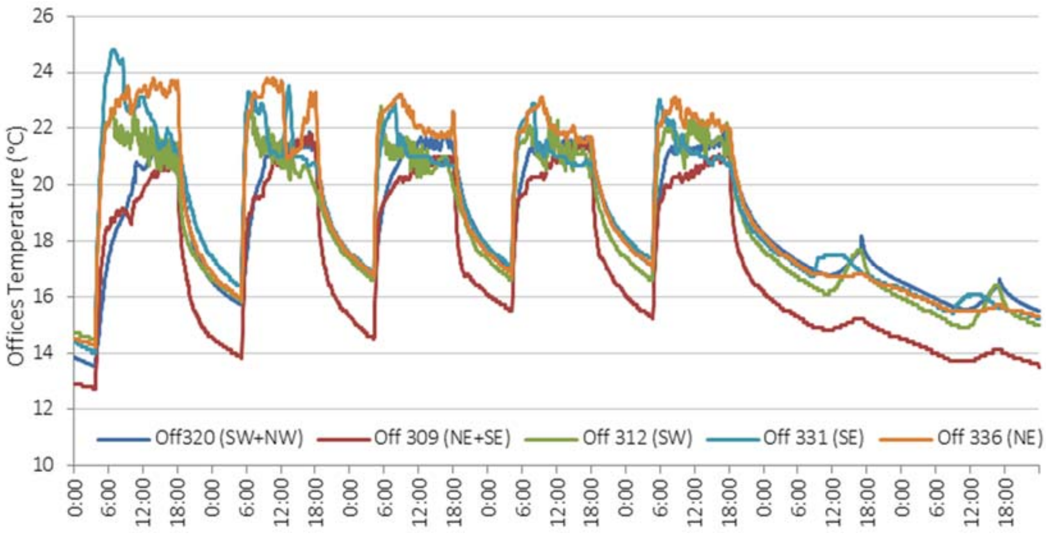


Figure 4.6: Offices temperatures in a winter week (6-12 January).

4.2.1.5 - HP PERFORMANCE RESULTS AND THERMAL ENERGY MEASUREMENTS

The first thermal energy meter (Brunata 2 - Br2) is installed after the buffer tank and the decoupling tank. In this way, when the system is turned OFF, some of the thermal energy delivered by the HP, instants before to the stop hour, is not supplied to the building, but stored in the buffer tank and, for this reason, is not measured by Brunata 2 . In order to be possible to compare results about the daily HP performance (DPF1), when some key parameters are changed, it is necessary to measure the thermal energy delivered by the HP (HP outlet). For this reason it was decided to install an additional thermal energy meter (Brunata 3 - Br3), which was not required by the Ground-Med project. A technical intervention at the hydraulic circuit was carried out and the new thermal energy meter became operational only in March 2014. The difference between the two thermal energy points of measurement and its influence in the HP performance factors (DPF1) is presented in Figure 4.7.

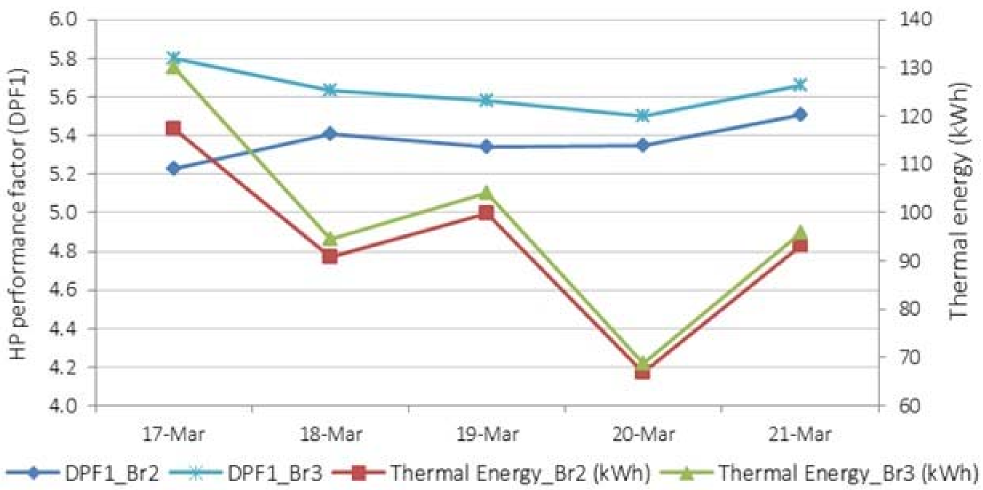


Figure 4.7: DPF1 variation with thermal energy measurements in two different points.

As it can be observed in Figure 4.7, at the end of the day after stopping the GSHP system, the thermal energy measured by Brunata 2 is always different from the energy measured by Brunata 3. This happens when HP compressor operation is stopped by the time schedule external signal or very nearly to the end of the operation time. A big difference between DPF1_Br2 and the DPF1_Br3 can be observed on 17 March. For this reason, whenever possible, the DPF results comparison will be presented calculating the performance factor with the thermal energy measured by Brunata 3.

4.2.1.6 - WATER TEMPERATURE SUPPLY AND HP SET POINT

The HP is controlled based on water return temperature, which means that the HP set point is the desired temperature that return water must reach. The water temperature supplied to the building is analyzed in terms of the set point established for the water return temperature to obtain the relation between both values, as presented in Figure 4.8.

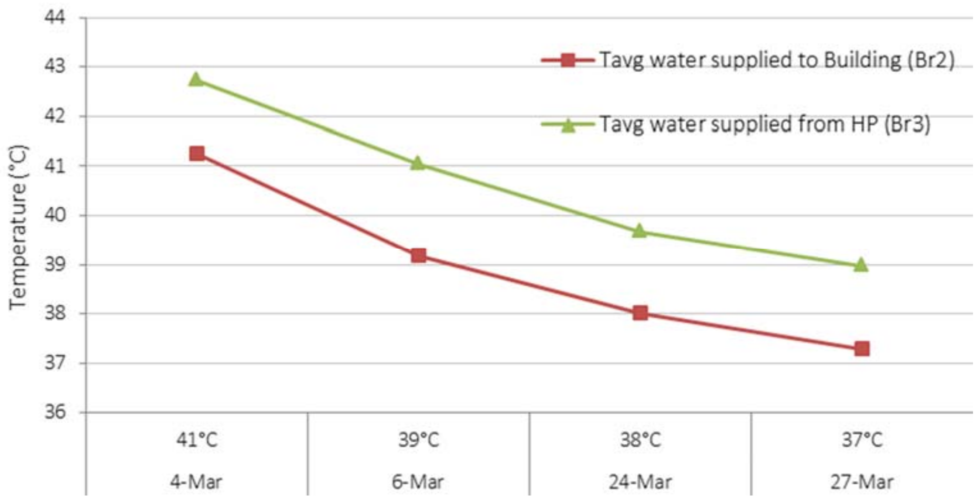


Figure 4.8: Water supply temperature with HP set point variation.

The temperature difference between the HP set point and the average temperature of the water supply from the HP only when compressors are working is around 1.7 to 2 °C. The water temperature supplied to the building is the average temperature (Tavg) of the water supplied during all the operating period of the system. By the results analysis it can be concluded that the water supply temperature to the building is very close to the set point value, with a maximum variation of 0.7 °C. In this way, based on the set point value, the water average temperature supplied to the building is always known.

4.2.1.7 - DPFS RESULTS AND WATER TEMPERATURE SUPPLIED FROM BHE

Figure 4.9 presents the HP performance results for 3 weeks of high heating period. These results were obtained with no variation of the HP water return set point (40 °C) and no variation of the three circulation pumps flow rates. The DPF1 results are presented based on the thermal energy measured by Brunata 2, since at this time the Brunata 3 was not installed.

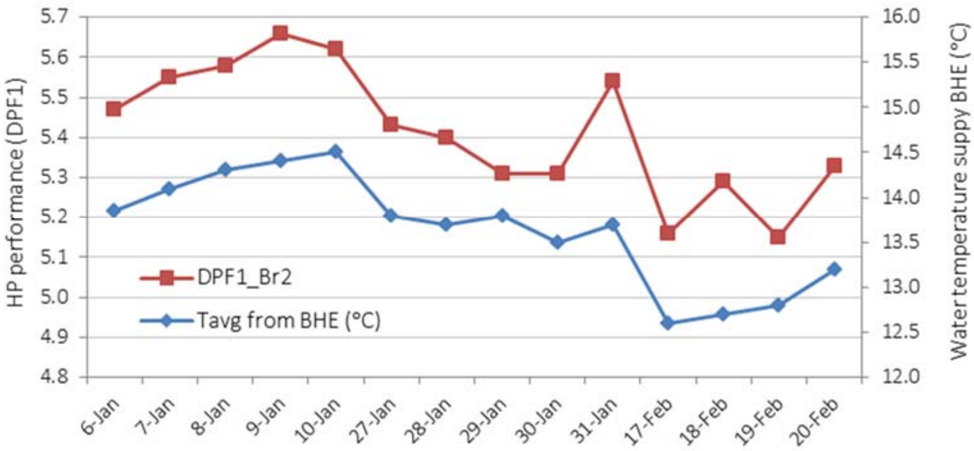


Figure 4.9: DPF1 variation with water supply from BHE.

Analyzing Figure 4.9, it is possible to conclude that DPFs results are largely influenced by the water supply temperature from the Borehole Heat Exchanger (BHE). Considering the same HP set point (temperature of the hot heat source), a higher water temperature from BHE corresponds to a lower temperature difference between the hot and cold heat sources, and consequently to higher DPFs results, as expected for ideal Carnot heat pumps, Equation (2.5). There are few days that this dependency is not clear, but it can be explained by the thermal energy that was not supplied to the building as referred before.

4.2.2 - INFLUENCE OF THE HP SET POINT ON THE DPFS

After the installation of the Brunata 3, it was possible to analyze the influence of HP set point variation on the thermal power supplied from HP and the HP performance factor (DPF1).

Figure 4.10 shows the HP thermal power variation with the HP set point variation. The thermal power increases with the set point decrease, since the temperature difference between the water supply from the BHE and the water supply from HP decreases. The average temperature of the water supply from BHE for each day test was different, varying from 13.1 °C to 14.6 °C, which also influence the results. The circulation pumps set points were maintained the same to perform these tests with the parameterization indicated in Table 4.1.

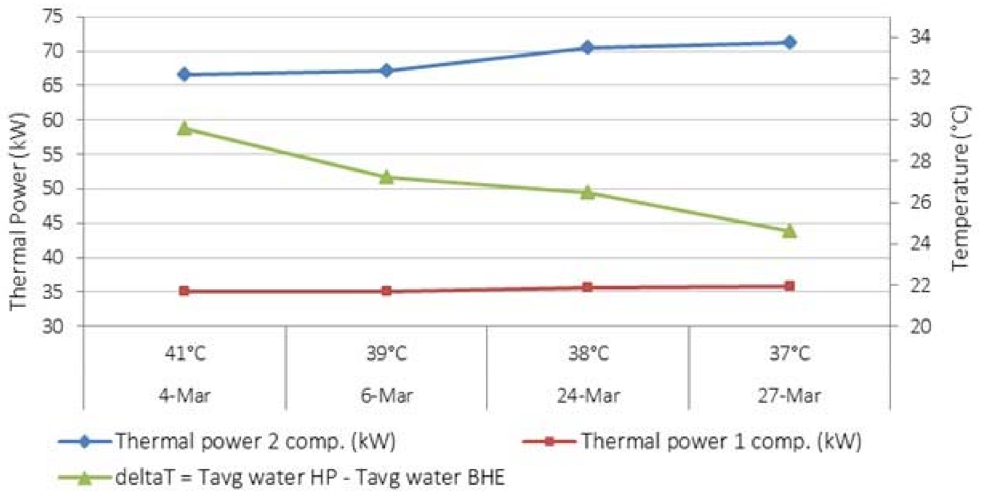


Figure 4.10: HP thermal power with HP set point variation.

Table 4.1: Circulation pumps parametrization.

Circulation Pumps	CP set point H [m]	Flow rate [m³/h]	Electric Power [W]	deltaT [°C]
External loop (CPext) - HP evaporator	10	8.1	285	3.4
Internal loop 1 (CPint1)- HP condenser	7	8.8	245	3.8-3.9
Internal loop 2 (CPint2)- Building	10	7.7	280	-

As expected, the HP performance factor DPF1 increases as the HP set point temperature decrease (Figure 4.11) and consequently the other daily performance factors also increases, since, for the same electric energy consumption, more thermal energy is produced.

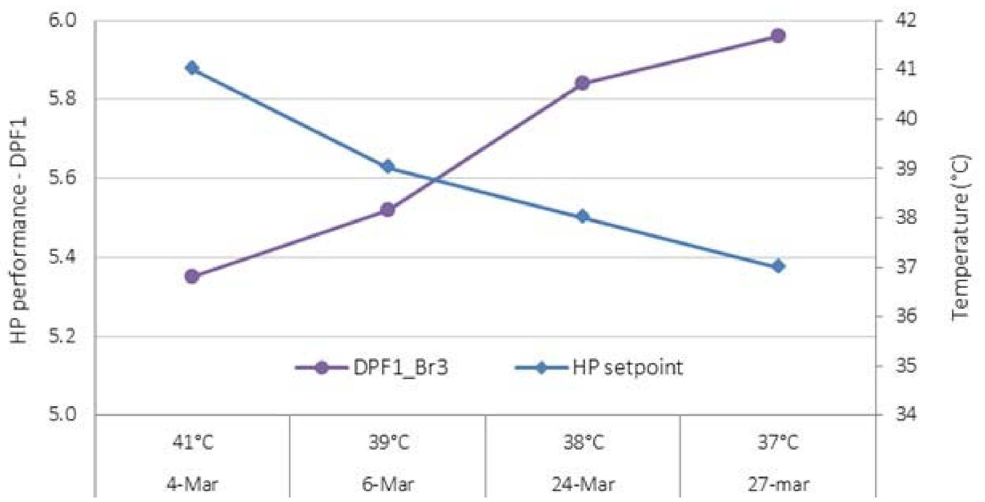


Figure 4.11: HP set point Influence in HP performance.

The DPF1 has decreased 10% with the increase of 4 °C in the HP set point (37 °C – 41 °C), which suggest that for an increase of 1 °C in the set point temperature the DPF1 decreases around 2.5%. The heat pump performance also depends on the water supply temperature from the BHE. The water temperature from the soil had a variation of 1.5 °C during this period. It can be observed that the DPF1 had a higher increase when the HP set point was reduced from 39 °C to 38 °C than in the other situations, due to the water supply temperature from the BHE, which has increased from 13.8 °C to 14.5 °C (the deltaT has decreased as a result of the decrease of the hot heat source temperature and increase of the cold heat source temperature). In the other two situations the DPF1 did not increase much since the water temperature from boreholes had decreased.

4.2.3 - INFLUENCE OF THE CIRCULATION PUMPS SET POINT ON DPFS

The heat pump manufacturer recommends a temperature difference between the water supply and the water return of 3 °C in the evaporator side ($deltaT_{in}$), and a temperature difference of 5 °C in the condenser side ($deltaT_{out}$) and that the CPint1 flow rate must be higher than the CPint2 flow rate, in order to guarantee a higher efficiency.

The flow rates of the three circulation pumps were measured and tested to determine the minimum flow rates supported by the heat pump. In order to vary the flow rate the circulation pumps head were varied manually. The heat pump gives a flow rate error in the evaporator side when the external circulation pump is in the head position 9 after some time, which corresponds to a flow rate of 7.6 m^3/h, in a steady state. The external circulation pump worked properly in the head position 10 (maximum adjustment), with a flow rate of 8.1 m^3/h, in steady state. With this flow rate the $deltaT_{in}$ had a variation between 3 and 3.3 °C at the evaporator side during all winter (2014), which is approximately the recommended value.

The influence of the circulation pumps variation in the HP performance was only carried out at the condenser side (CPint1), since no variations were possible to perform at the evaporator side. In the condenser side the heat pump gave an error of overheat when the flow rate of the primary loop was set to 6.7 m^3/h. In this way, only 3 head positions variation were tested, always keeping the flow rate of the CPint1 slightly higher than the flow rate of the CPint2, as presented in Table 4.2. These tests were performed in days with similar conditions: a similar average outdoor temperature (13.6 – 15.3 °C), a similar load factor (40 - 46%), a similar water temperature supply from the BHE (14 - 14.1 °C) and the same HP set point temperature (37 °C).

Table 4.2: Internal circulation pumps flow rates combinations tested.

Internal circulation pump 1				Internal Circulation pump 2			DPF1	DPF3
H [m]	Q [m³/h]	P [W]	deltaT$_{int1}$	H [m]	Q [m³/h]	P[W]		
7	8.8	245	3.8	10	7.7	263	5.96	4.97
6	7.7	163	4.3	9	6.8	203	5.90	5.02
5	7.2	120	4.7	9	6.8	203	5.82	5.11

At the condenser side, for the minimum flow rate affordable (7.2 m^3/h) by the HP, a deltaT$_{out}$ of 4.7 °C was obtained. Increasing the flow rate, the HP performance (DPF1) also increases. This effect can be justified because as the flow rate increases, deltaT$_{out}$ decreases, facilitating the heat transfer process between the water and the refrigerant fluid. However, increasing the circulation pumps flow rate also increases their energy consumption, which penalizes the overall performance system. The DPF3, which includes the three circulation pumps electric consumption, is improved when the flow rate was decreased. It was expected a fancoils consumption increasing with the flow rate reduction, but this effect was not noticed in the results, since the DPF4 (global DPF) also increased (Figure 4.12).

It can be concluded that to obtain a higher DPF1 is better to have a higher flow rate. Still, it can be noticed that the DPF3 increases with the decreasing of the flow rate, which is consistent, since for a lower flow rate the circulation pump consumes less energy.

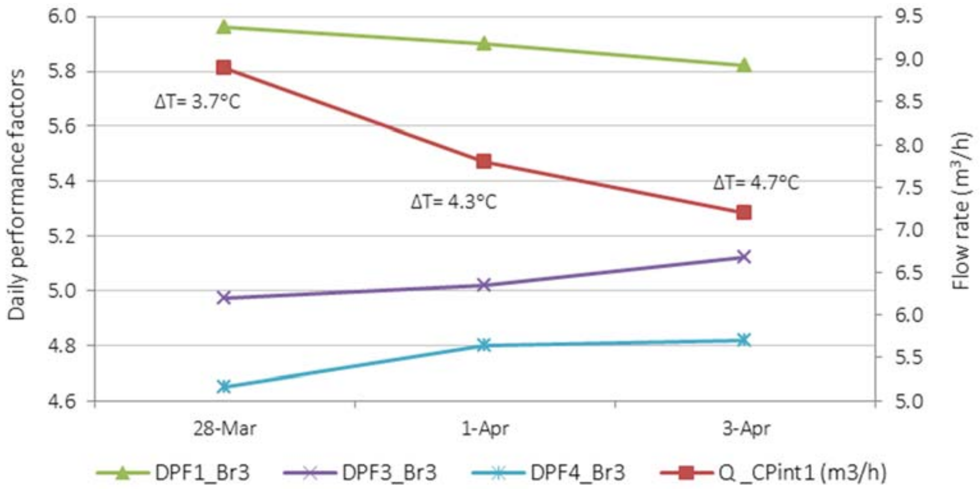


Figure 4.12: DPFs variation with flow rate variation in the internal circuit.

From the results achieved, it was concluded that to improve the overall performance of the GSHP system, the circulation pumps should be set to the minimum set point affordable by the

HP. In the case of the internal circulation pump 2, the minimum set point should guarantee the necessary flow rate in order that the fancoils heating performance is not compromised. It was verified that the CPint2 set point can vary between 9 to 10 m depending on being a less cold day or a colder day, without interference in the users thermal comfort.

4.2.4 - SPFS RESULTS OF THE HEATING SEASON 2013/2014

The heating season considered is from 25 November 2013 to 11 April 2014. In Figure 4.13, the daily average thermal and electric energy consumption are represented for each winter week. In average, per day, during this season, around 230 kWh of thermal energy were extracted from the ground and 272 KWh were delivered to the building, corresponding to a daily electric consumption of 63 kWh (including circulation pumps and fancoils consumption). The daily average outdoor temperature per week was within the range of 11.5 °C to 14 °C until the end of February, and increased in March, as it can be observed in Figure 4.13. This winter season was very rainy. The thermal energy variation along the weeks was not only due to climate conditions variations, but also due to the variation of the operation period, which varied between 11 hours, in the less cold days, and 14 hours, in the coldest days.

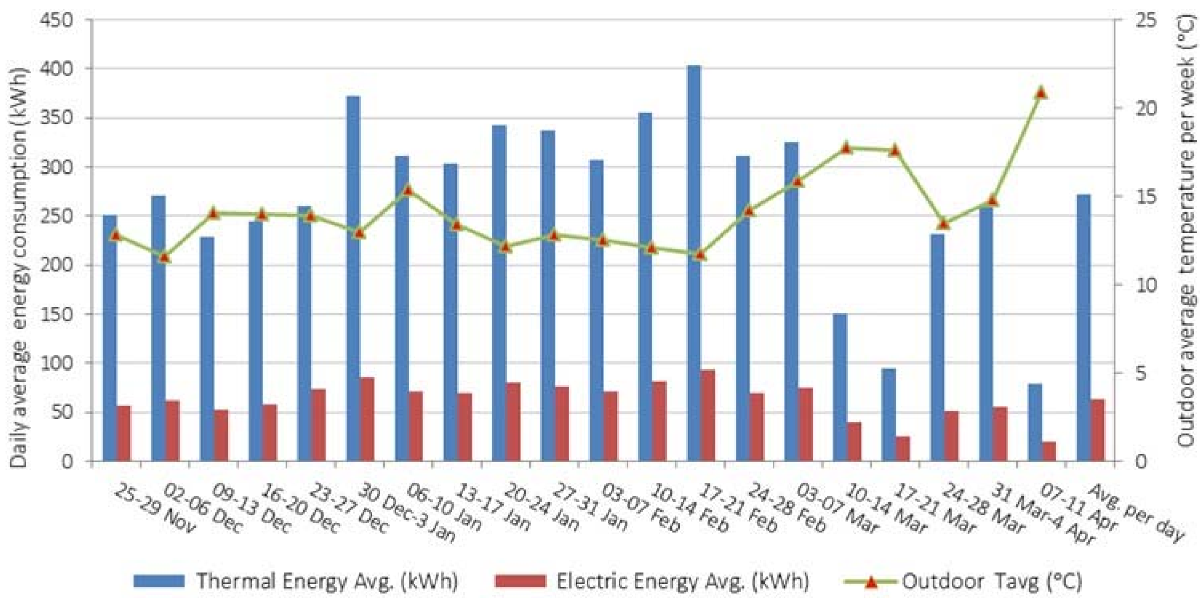


Figure 4.13: Daily average energy consumption and outdoor temperature per week in the heating season 2013/2014.

In this winter season the following seasonal performance factors were obtained:

- A SPF1 of 5.4 for the HP technology;
- A SPF2 of 5.1 for GSHP technology (including the ground loop circulation pump);
- A SPF3 of 4.5 for the GSHP system excluding fancoils consumption;
- A global SPF4 of 4.3 for the overall GSHP system.

Figure 4.14 presents the variation of the weekly average performance factors (weekly SPFs) along the 20 weeks that the GSHP system operates in the heating mode.

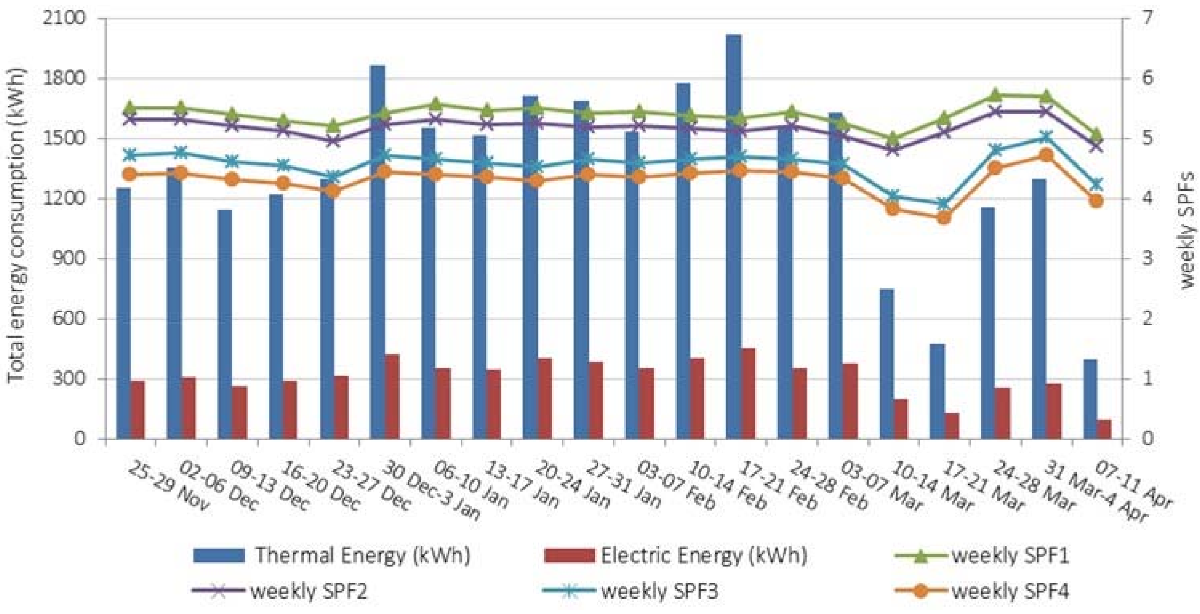


Figure 4.14: Total energy consumption per week and the corresponding weekly SPFs in the heating season 2013/2014.

As it can be observed in Figure 4.14, there is a significant difference between the SPF2 and the SPF3. For SPF3 the consumption of the two internal circulation pumps is also considered and they are always working during the operation period of the system, while external circulation pumps follows the HP compressors cycling operation. The parameterization of this winter season was done, during most of the time, with a HP set point of 40 °C for water return temperature and CPs head positions of 10 m for the CPext (P = 285 W; Q = 8 m^3/h) and CPint2 (P = 280 W; Q = 7.7 m^3/h) and of 7 m for the CPint1 (P = 244 W; Q = 8.8 m^3/h).

Some tests were carried out during March and April to improve the SPFs by reducing the water supply temperature (i.e., reducing the HP set point) and by decreasing the circulation pumps velocity, as already explained previously. The effect of the parameters adjustment can be observed in Figure 4.14 for the last week of March (24 - 28) and first week of April (31 March –

4 April), where the SPFs presented the best values of the season (SPF1 = 5.7; SPF2 = 5.5; SPF3 = 5; SPF4 = 4.7). The HP set point was decreased in March to 38 ˚C and to 37 ˚C during the last week of March. The CPs set points were also decreased to 9 m for the CPint2 (P = 219 W; Q = 7.3 m^3/h) and to 6 m for the CPint1 (P = 163 W; Q = 7.7 m^3/h).

By the tests performed, it was concluded that the HP set point can vary from 40 ˚C in cold days to 37 ˚C in the less cold days and the circulation pumps set points can also be decreased, during mild weeks, without decrease of users thermal comfort.

4.2.5 - REPRESENTATIVE DAY OF THE HEATING SEASON 2013/2014

A day of the winter season, that had the four daily performance factors very close to the SPFs values obtained, was chosen as a representative day (26 February) to present the thermal load and electric load profiles as well as the water temperature in the hydraulic circuits of the GSHP system (SPF1 = 5.34; SPF2 = 5.10; SPF3 = 4.55; SPF4 = 4.33).

The total electric power profile of the GSHP system is represented in Figure 4.15. The main electric consumption is from the heat pump as shown in Figure 4.16. The indoor temperature at 4:30 a.m. was 18.8 ˚C and two hours later reached 21 ˚C, but took three hours more to reach the office temperature set point (22 ˚C). This monitored office is the office that has the highest volume and the lowest thermal power capacity installed per volume (26.5 W_{th}/m^3)

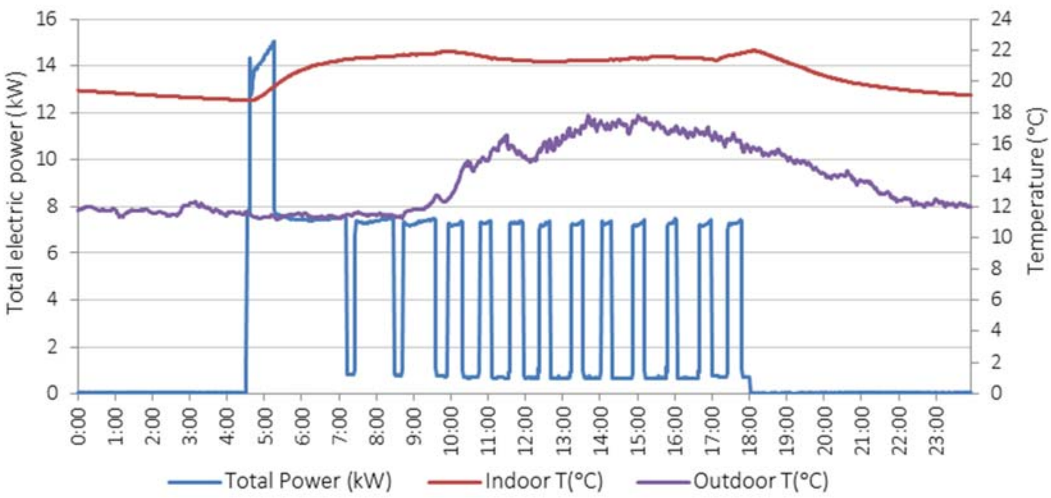


Figure 4.15: GSHP system electric load profile and indoor and outdoor temperatures in the heating mode.

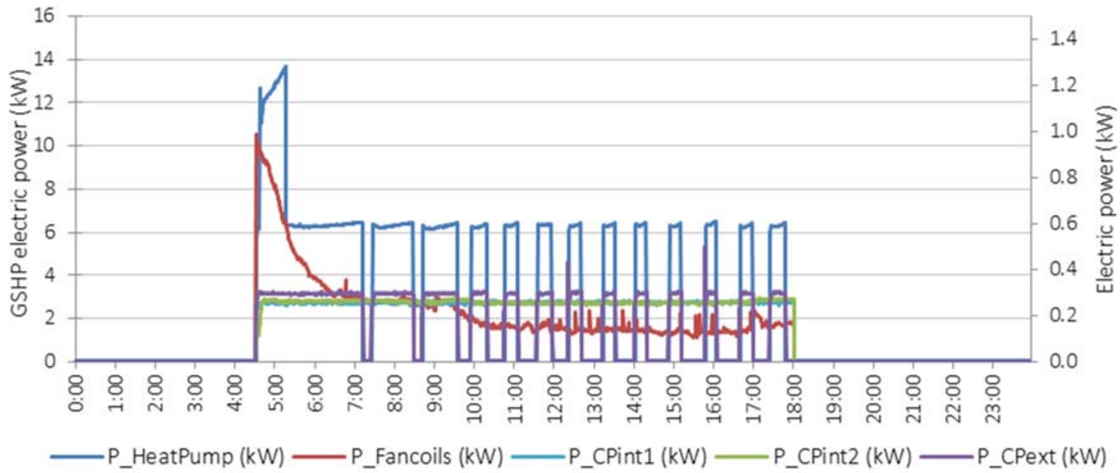


Figure 4.16: Electric load profile of each GSHP system component.

Analyzing Figure 4.15 and Figure 4.16, it can be observed that the HP and fancoils consumption stabilizes at a time very close to the reaching of the office temperature set point. In this way, this office temperature can be used as an indicative temperature of the 2nd floor temperature evolution. The indoor temperature decrease of around 0.6 ˚C, after 10 a.m., is due to users leaving the door open after arriving.

In the heating season, only in the morning period, both compressors work to compensate the high temperature difference between the water return and the HP set point (deltaT > 3.5 ˚C). When the temperature difference is above 1.5 ˚C the second compressor (second stage) is turned OFF, and is only turned ON again if deltaT becomes again higher than 3.5 ˚C. This last situation only happens in very cold days and until the building temperature stabilizes.

The water temperature supply and the water temperature return of the ground loop circuit and the water temperatures of the internal circuit loop which feeds the fancoils of the 2nd floor, are shown in Figure 4.17. In this particular day the water came from BHEs at 13.6 ˚C and returned at 10.5 ˚C, in average. The hot water was supplied to the fancoils at 40.3 ˚C and returned at 38 ˚C, in average. The HP was set to operate at 40 ˚C and the circulation pumps head positions were set as follows: CPext and CPint2 with 10 m and CPint1 with 7 m.

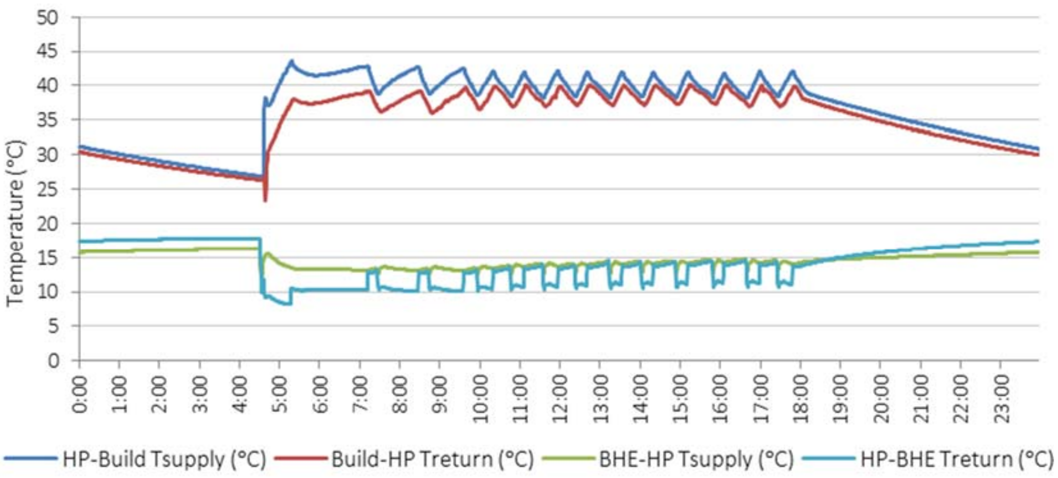


Figure 4.17: Water temperatures in the external and internal hydraulic loops.

During this day around 230 kWh of thermal energy was extracted from ground and around 290 kWh were delivered to the building, corresponding to a total electric consumption of 66 kWh, where HP represents around 80% of the total electric consumption.

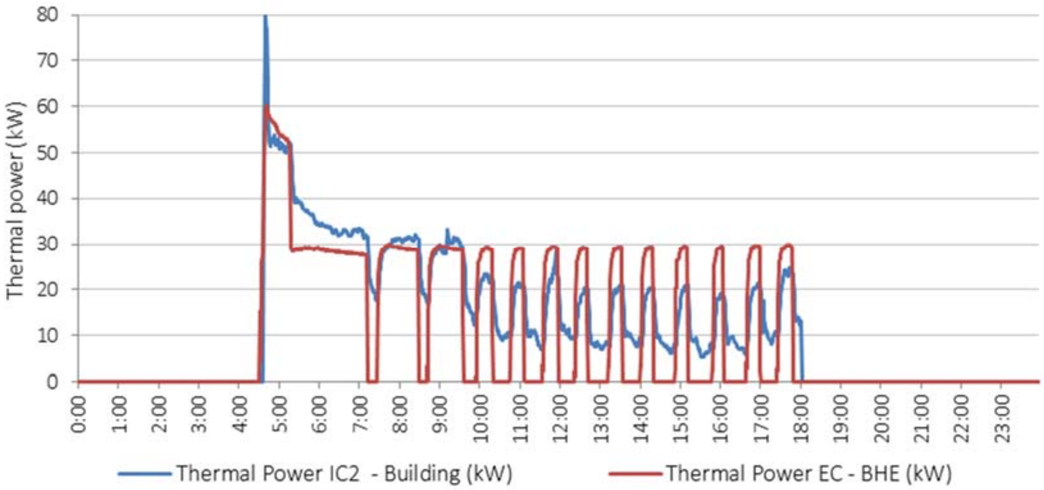


Figure 4.18: Thermal energy extracted from ground and delivered to the building.

4.3 - COOLING SEASON 2014

The cooling season considered is from 16 June to 24 October 2014 (19 weeks). During this cooling season the main developments were:

- Expansion modules (GENI modules) were installed in each circulation pump, in order to be possible to change the circulation pumps set points remotely by an external signal. This was very useful to perform the experimental tests to find the best circulation pumps set points in order to improve the global system performance.
- The initial four 3 way motorized valves, responsible for the HP reversibility, were replaced by eight 2 way motorized valves, before the starting of the cooling season.
- The seasonal and weekly performance factors were calculated and analyzed. The results achieved are explained according to the parametrizations variation and the load factor variation.

4.3.1 - CIRCULATION PUMPS AND GSHP SYSTEM OPTIMIZATION IN COOLING MODE

After installing the GENI modules, in August 2014, several tests were performed to find the combination of the circulation pumps speeds which optimize the GSHP system performance. As referred in Section 3.2.3, these modules allow changing the circulation pumps set points in a range of 0 - 100% of the maximum set point of 10 m, which is presented in graphs and tables as the circulation pumps velocities variation. These new control modules allow smaller CPs velocity variations than in the CPs control panels.

The set point of each circulation pump was varied in a range of 40% to 100%, in order to obtain, for each position, the corresponding flow rate and electric power consumption, as presented in Figure 4.19.

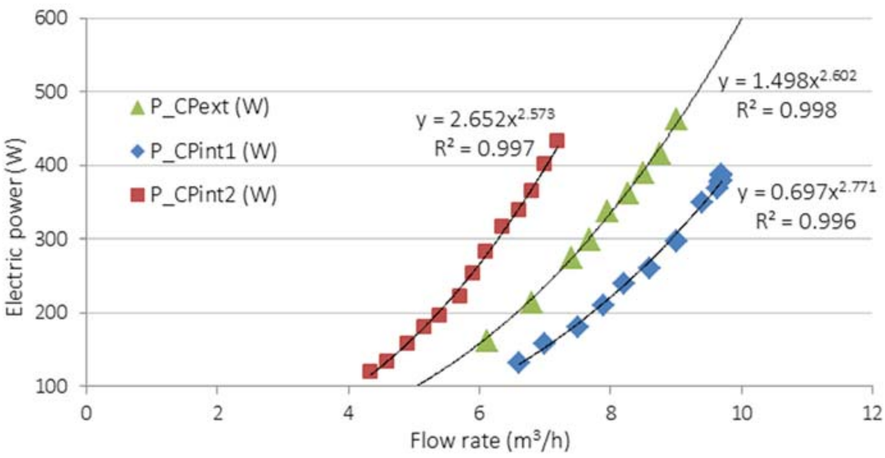


Figure 4.19: Electric power curves of the three circulation pumps.

In the cooling mode, the internal circulation pump 1 is at the evaporator side and the external circulation pump is at the condenser side. The minimum flow rates for each circulation pump are different from those obtained in the heating mode. Still, the hydraulic circuit was changed by the replacement of four valves by eight valves, leading to different pressure drop in the hydraulic circuits. At the evaporator (CPint1), the minimum flow rate affordable by the HP is 7 m^3/h and at the condenser side (CPext) is 6.7 m^3/h.

The velocity variation of the two internal circulation pumps was made in a way to ensure that the flow rate of the CPint1 is always slightly higher than the CPint2 flow rate. For each CPint1 velocity variation, a specific CPint2 velocity variation was defined, as represented in Table 4.3. The corresponding flow rates and electric power consumption are also presented in Table 4.3 and the same information is given in Table 4.4 for the external circulation pump.

Table 4.3: Flow rate and electric power of the internal CPs according to the velocities variation.

Internal circulation pump 1				Internal circulation pump 2		
V_CPint1 [%]	Q_CPint1 [m^3/h]	P_CPint1 [W]	DeltaT int1 [°C]	V_CPint2 [%]	Q_CPint2 [m^3/h]	P_CPint2 [W]
45	7.0	160	3.5	60	6.0	220
50	7.4	180	3.3	70	6.5	270
60	8.1	240	3.0	70	6.5	270
70	8.9	300	2.7	80	7.0	330

Table 4.4: Flow rate and electric power of the external CP.

External circulation pump			
V_CPext [%]	Q_CPext [m^3/h]	P_CPext [W]	DeltaT ext [°C]
60	6.7	214	4.2
70	7.3	274	3.8
80	7.9	340	3.6
90	8.4	390	3.3

The combination of the circulation pumps velocities is made between the external circulation pump and the internal circulation pump 1. For each circulation pump (external pump and internal pump 1) four velocities variation were tested, which resulted in sixteen combinations and one different combination of velocities was tested per day, as presented in Figure 4.20.

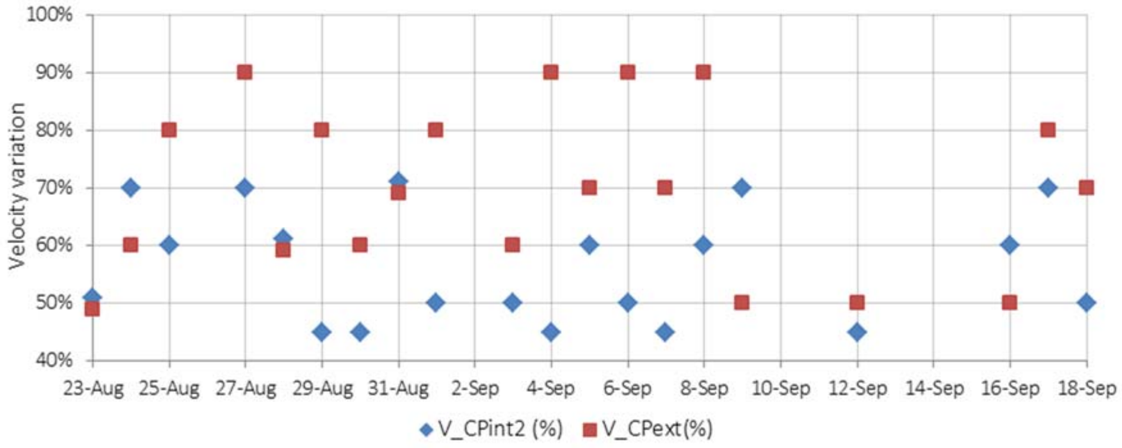


Figure 4.20: Circulation pumps velocities combinations tested in cooling mode.

The circulation pumps velocities were varied randomly from day to day to minimize the load variation influence in results. However, the weather of this cooling season was very variable, which resulted in very different thermal needs and in very different HP load factors during the period tests (from 25% to 55%), as represented in Figure 4.21. The HP load factor was calculated by the ratio of the average HP power consumption, during the operation period, by the maximum electric power. In the cooling season just one HP compressor is considered since the second compressor never worked during this season.

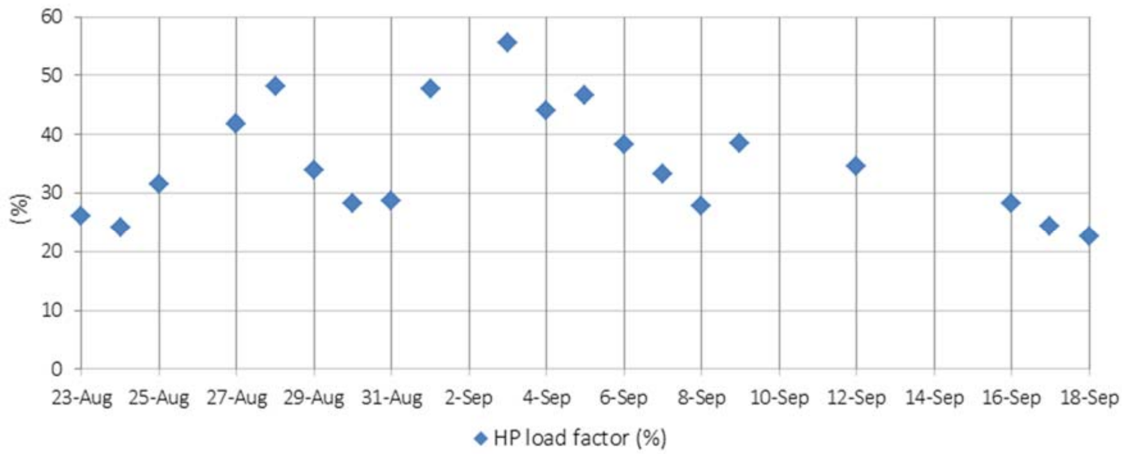


Figure 4.21: Heat pump load factor during tests periods in cooling mode.

In order to compare the results achieved, the internal circulation pumps energy consumption was only considered when the HP compressor was working, to reduce the thermal needs variation influence in the DPFs calculations. As already referred, these circulation pumps are always turned ON during the operation period of the system, which was always from 7 a.m. to 6:30 p.m. in this cooling season. For a day with lower thermal needs, the internal circulation pumps electric consumption will have a higher share in the total electric consumption than usual.

For the DPFs calculations, the thermal energy produced by the HP (measured by Brunata 3) was used and not the thermal energy delivered to the building (measured by Brunata 2), since the thermal energy that stays stored in the buffer tank of 1000 liters at the end of the day varies from day to day. The DPFs results obtained by varying the circulation pumps velocities are presented in Figure 4.22.

DPF1 improves with CPs velocities and flow rate increase, but this improvement does not compensate the increase of the electric energy consumption associated to higher velocities. Analyzing the DPF3 and DPF4 results it can be concluded that the combination of the circulation pumps velocities that optimizes the system performance corresponds to 60% for the external pump and to 50% for the internal pump 1, which corresponds to 70% for the internal pump 2, as defined in Table 4.3. It must be noticed that the DPFs obtained for this analysis are higher than the real DPFs due to the reasons presented before. Applying these setting velocities to the circulation pumps, the overall performance factor DPF4 improves around 8% when compared with the maximum velocity pump settings tested.

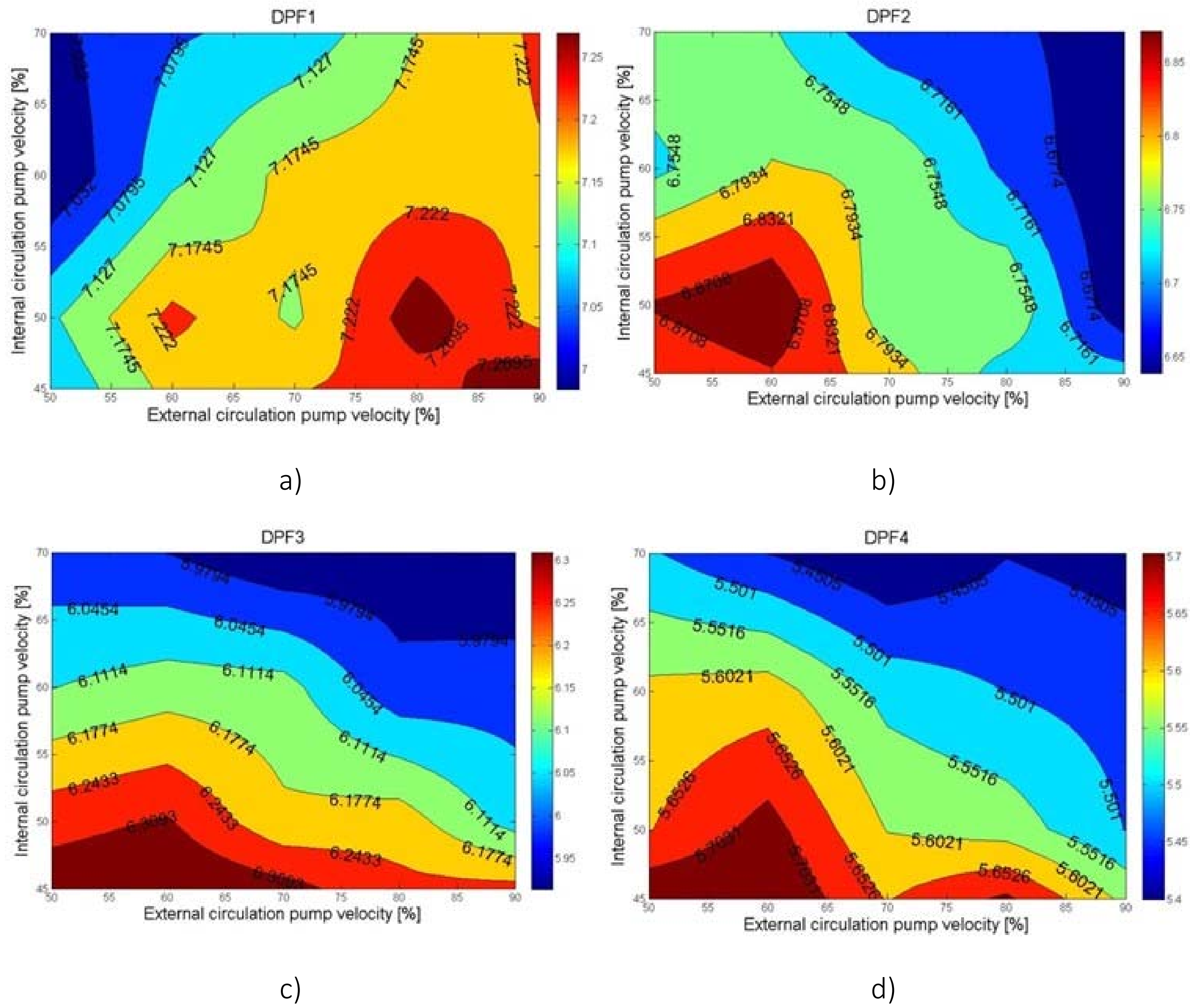


Figure 4.22: DPFs results with circulation pumps velocities variation.

4.3.2 - SPFS RESULTS OF THE COOLING SEASON 2014

The cooling season was considered to be between 16 June 2014 and 24 October 2014, as already mentioned. In Figure 4.23, the daily average thermal and electric energy consumption are represented for each summer week. In average, per day, during this season, around 120 kWh of thermal energy were extracted from the building and were delivered to the ground, corresponding to a daily electric consumption of 29 kWh (including circulation pumps and fancoils). The operation period was from 7 a.m. to 6:30 p.m. for the entire season.

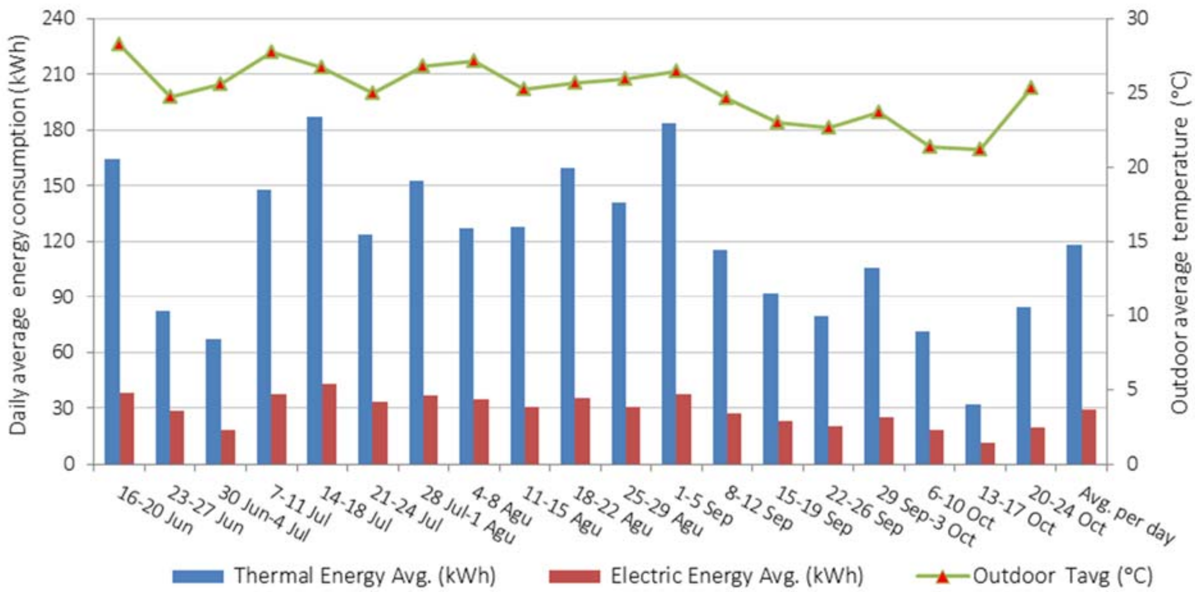


Figure 4.23: Daily average energy consumption and outdoor temperature per week in the cooling season 2014.

In this summer season the following seasonal performance factors were obtained:

- A SPF1 of 6.4 for the HP technology;
- A SPF2 of 5.8 for GSHP technology (including the ground loop circulation pump);
- A SPF3 of 4.2 for the GSHP system excluding fancoils consumption;
- A global SPF4 of 3.9 for the overall GSHP system.

These are very good results taking into account that the load factor had a variation between 22 and 50%, since the summer of 2014, in Portugal, was much cooler than normal. Figure 4.24 presents the variation of the weekly average performance factors (weekly SPFs) along the 19 weeks that the GSHP system operates in the cooling mode.

The average outdoor temperature was about 28 – 30 ˚C only in the following weeks: 3rd week of June, 3rd week of July, 3rd week of August and 1st week of September, which corresponds to the higher HP load factors (40 - 50%) and consequently to the best SPFs (6.8 - 6.9). The lowest SPFs observed in Figure 4.24 are those that correspond to days with HP load factors below 25%.

At the start of the cooling season the set point temperature was 10 ˚C in water return and the velocity was set to 100% for all CPs, and during 4 weeks the CPext was not turned OFF with the HP cycles operation, which justify the worst results. After that the HP set point was adjusted to 12 ˚C and to 13 ˚C in last 2 weeks. Since the last week of August the CPs velocities were reduced and adjusted to lower set points. This strategy contributed to a global performance above 4, even with low factors of 20 - 25%. The exception is for the last but one week, with a HP load factor of 10%.

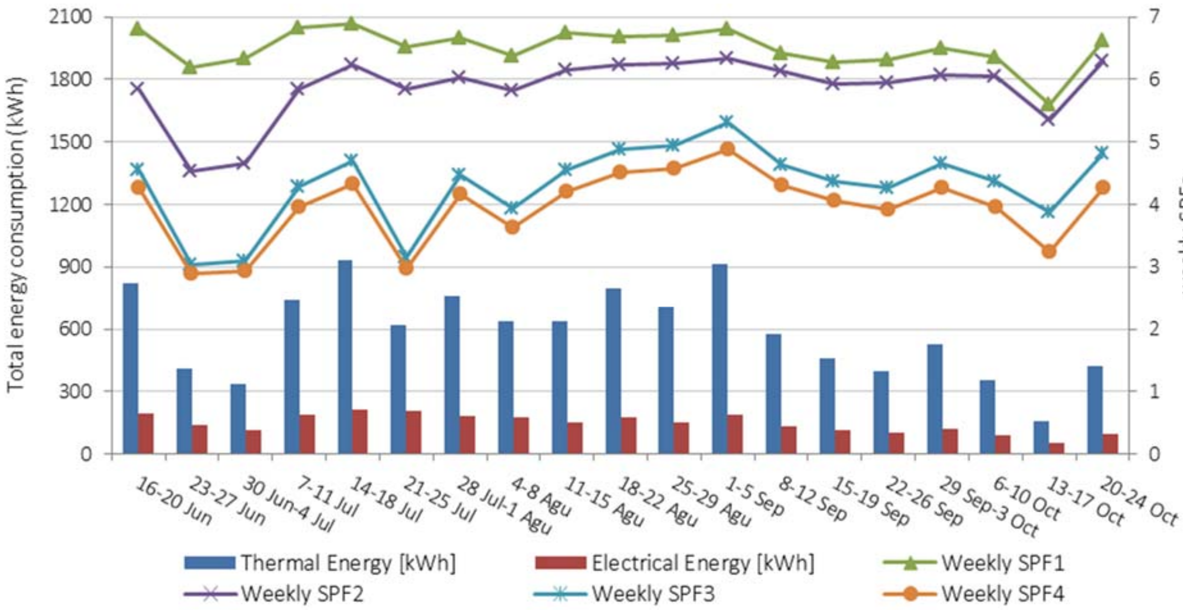


Figure 4.24: Total energy consumption in the cooling season per week and the corresponding weekly SPFs.

By the tests performed, it was concluded that the HP set point can vary from 10 ˚C in hot days to 13 ˚C in the less hot days and that the circulation pumps set points can also be decreased, during mild weeks, without decrease of users thermal comfort.

4.3.3 - REPRESENTATIVE DAY OF THE COOLING SEASON 2014

A day of the cooling season, that had the four daily performance factors very close to the SPFs values obtained, was chosen as a representative day (8 September) to present the thermal load and electric load profiles, as well as the water temperature in hydraulic circuits of the GSHP system (SPF1 = 6.37; SPF2 = 5.82; SPF3 = 4.30; SPF4 = 3.99).

The total electric power profile of the GSHP system is represented in Figure 4.25. The indoor temperature at 7 a.m. was 24.7 ˚C and the office temperature set point was 25 ˚C. During the afternoon the indoor temperature increased until 25.5 ˚C and increased more after turning OFF the GSHP system at 6:30 p.m., since this office has a SW and NW orientation. The outdoor average temperature was 25 ˚C, the minimum temperature was 20 ˚C and the maximum was 29 ˚C. In this particular office and also in other offices, users prefer open windows than having the air conditioned system turned ON, which may explain why office temperature increased above the office set point temperature. This office, in the cooling season, is not representative of the 2nd floor temperature, as it happened in the heating season.

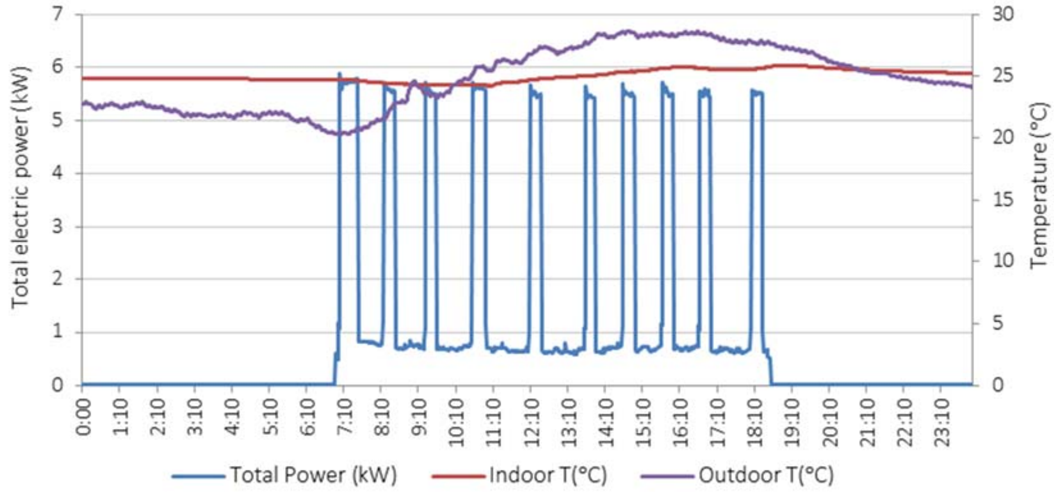


Figure 4.25: GSHP system electric load profile in cooling mode and indoor and outdoor temperatures.

In the cooling season, just one compressor works as it can be observed in Figure 4.25. The HP compressor starts working when the water return temperature of the fancoils is 2 ˚C higher than the HP set point. In Figure 4.26, it can be observed the higher contribution of the internal circulation pumps consumption for the total electric consumption in a day with lower thermal needs (HP load factor of 25%).

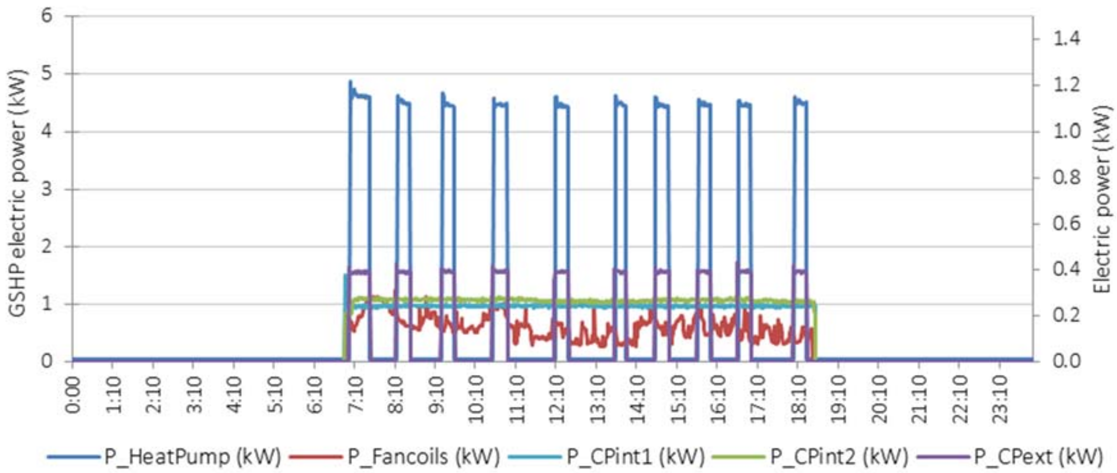


Figure 4.26: Electric load profile of each GSHP system component in cooling mode.

The water temperature supply and the water temperature return of the ground loop circuit and of the internal circuit loop which feed the fancoils of the 2nd floor are shown in Figure 4.27. In this particular day the water came from BHEs at 20.1 ˚C and returned at 23.8 ˚C, in average.

The cold water was supplied to the fancoils at 11.5 ˚C and returned at 13.5 ˚C, in average. The HP was set to operate at 12 ˚C and the circulation pumps were set as follows: V_CPext - 90% (Q_CPext = 8.5 m^3/h); V_CPint1 - 60% (Q_CPint1 = 8.1 m^3/h) and V_CPint2-70% (Q_CPint2 = 6.5 m^3/h).

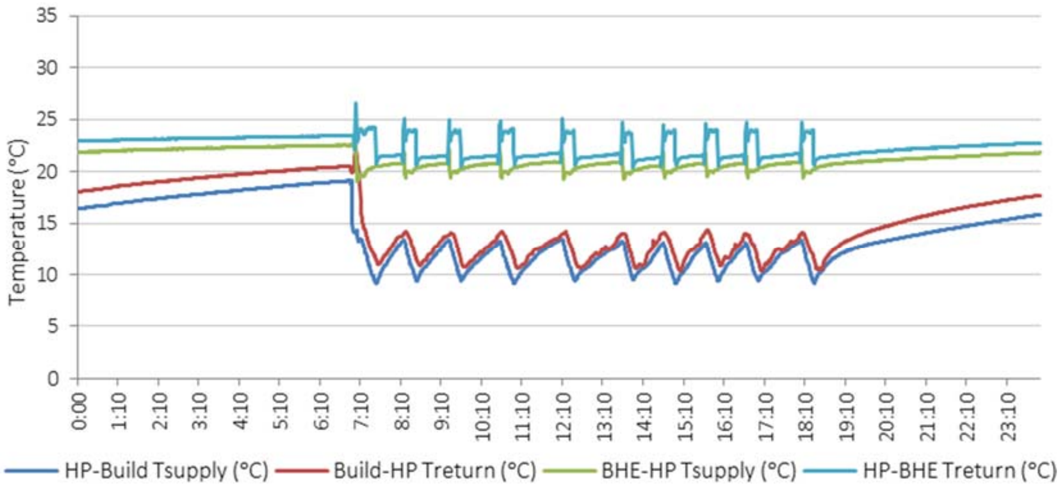


Figure 4.27: Water temperatures in the external and internal hydraulic loops in cooling mode.

During this day around 98 kWh of thermal energy was extracted from building and around 124 kWh were delivered to the building, corresponding to a total electric consumption of 25 kWh, where HP represented around 63% of the total electric consumption. The circulation pumps consumption had a higher contribution (30%) than in other days with higher HP load factors. For example, in 3 September, which was the day with the best SPFs of the cooling season (SPF1 = 6.86; SPF2 = 6.53; SPF3 = 5.56; SPF4 = 5.08) and with the higher HP load factor, the contribution of the circulation pumps was 17% for the total electric consumption.

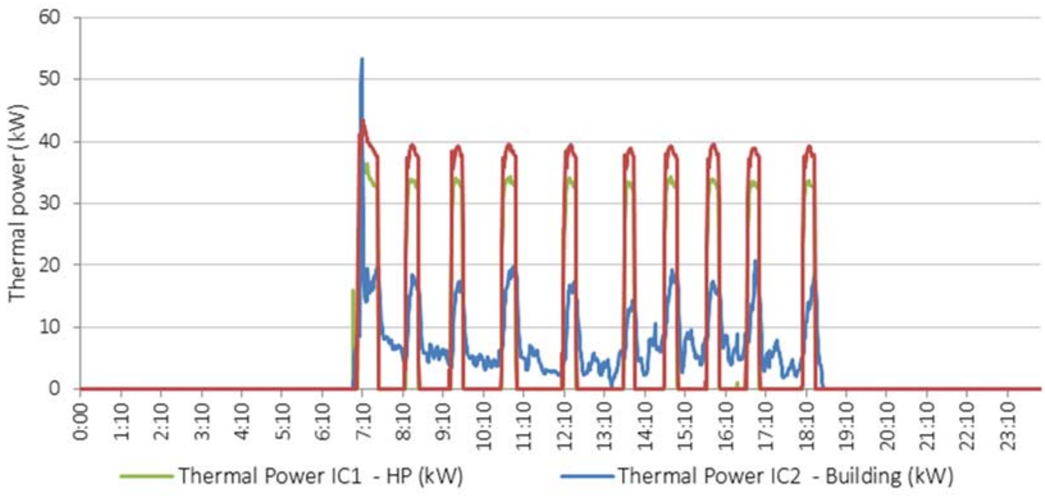


Figure 4.28: Thermal energy extracted from building and delivered to the ground in cooling mode.

4.4 - HEATING SEASON 2014/2015

The GSHP system worked in heating mode from 11 November 2014 to 11 April 2015, but the data is only presented after November 17 since some problems occurred. During this heating season the main developments were:

- The previous circulation pumps (Magna model) were substituted in October 2014 by new ones supplied by Grundfos, which are an updated model with more control capabilities. They can be directly commanded by external signals without additional hardware.
- Experimental tests to find the best circulation pumps set points in the heating mode, in order to improve the global system performance, were also carried out during this season, since in the previous winter season 2013/2014 the circulation pumps control was not installed.
- The seasonal and weekly performance factors were calculated and analyzed. The results achieved are explained according to the parametrizations variation and the load factor variation.

4.4.1 - CIRCULATION PUMPS AND GSHP SYSTEM OPTIMIZATION IN WINTER

During the heating season 2014/2015, some tests were performed to find the combination of the circulation pumps speeds, which optimize the GSHP system performance in the heating mode. The variation velocities range is lower than in the cooling season, due to the minimum flow rates supported by the HP.

In the heating season, the external pump is linked to the evaporator side of the HP and the internal pump 1 is linked to the condenser side. After performing some tests it was concluded that the minimum velocities that guarantee that system works continuously without errors/faults are 80% (Q_CPext = 8m^3/h) for the external pump and 65% (Q_CPint1 = 8.6m^3/h) for the internal pump 1. In the case of the internal circulation pump 1, the minimum flow rate affordable by the HP has increased when compared with the previous heating season 2013/2014. The hydraulic circuit had changes, the circulation pumps and the valves responsible by the reversibility of the GSHP system were replaced, but besides that no other explanation was found.

For the different velocity positions (from 50% to 100%), the corresponding flow rates and power consumption for each circulation pump were obtained, as presented in Figure 4.29.

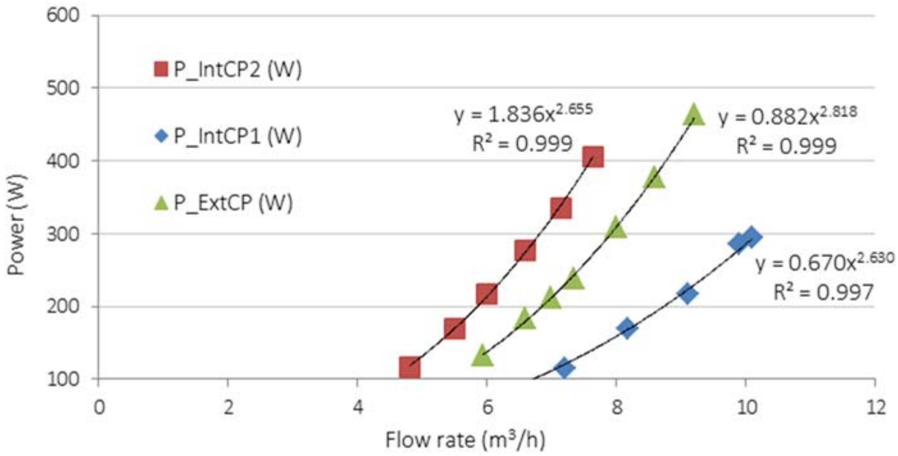


Figure 4.29: Electric power curves of the three circulation pumps in heating mode.

To obtain the best combination for the circulation pumps velocities that optimizes the overall GSHP system performance, a similar strategy used in the two previous seasons was used for this heating season. In this winter season, since it was colder than the previous one, the velocity of the internal pump 2 (which feeds the fancoils circuit) was always set to 100%, in order to guarantee that fancoils could provide the maximum heat.

Table 4.5 and Table 4.6 present the corresponding flow rates and power consumption to the velocities range tested for each circulation pump. The different flow rates for the same position velocity of the internal circulation pump 2 are due to the number of fancoils which are working.

Table 4.5: Flow rate and power demand of the internal CPs according to the velocities variation.

Internal circulation pump 1				Internal circulation pump 2		
V_CPint1 [%]	Q_CPint1 [m³/h]	P_CPint1 [W]	DeltaT int1 [°C]	V_CPint2 [%]	Q_CPint2 [m³/h]	P_CPint2 [W]
65	8.6	190	3.9	100	7.6 - 8.5	406 - 430
70	9.1	217	3.8	100	7.6 - 8.5	406 - 430
80	9.9	285	3.5	100	7.6 - 8.5	406 - 430

Table 4.6: Flow rate and electric power of the external CP.

External circulation pump			
V_CPext [%]	Q_CPext [m^3/h]	P_CPext [W]	DeltaT ext [°C]
80	8.0	309	3.5
90	8.6	378	3.3
100%	9.2	463	3.0

The combination of the circulation pumps velocities tested (nine combinations) are presented in Figure 4.30.

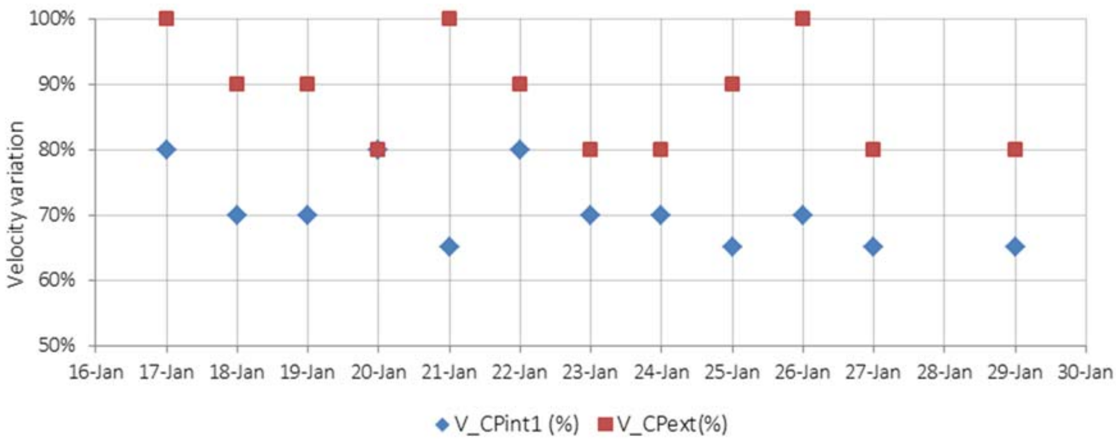


Figure 4.30: Circulation pumps velocities combinations tested in heating mode.

To calculate the DPFs in this heating season the methodology used in the cooling season was followed. Since in the heating mode the two compressors work at least at the beginning of the system operation, the HP load factors are calculated using the maximum power of the two compressors and are presented in Figure 4.31.

The DPFs results obtained are presented in Figure 4.32. Analyzing the results of the overall system performance (DPF3 and DPF4) it can be concluded that the combination of the circulation pumps velocities that optimizes the system performance corresponds to 80% for the external pump and 65% for the internal pump 1 (which corresponds to the minimum velocity positions supported by HP). The overall performance system (DPF4) has improved 2% when compared with the maximum velocity pump settings tested.

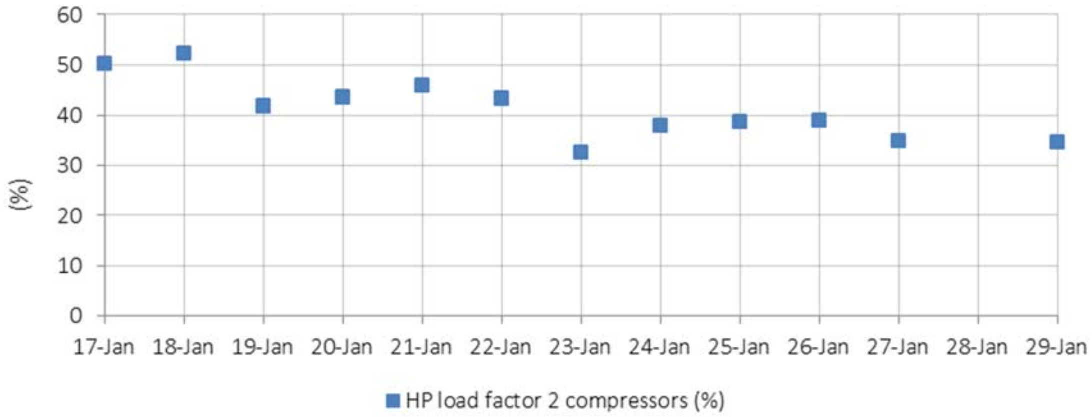
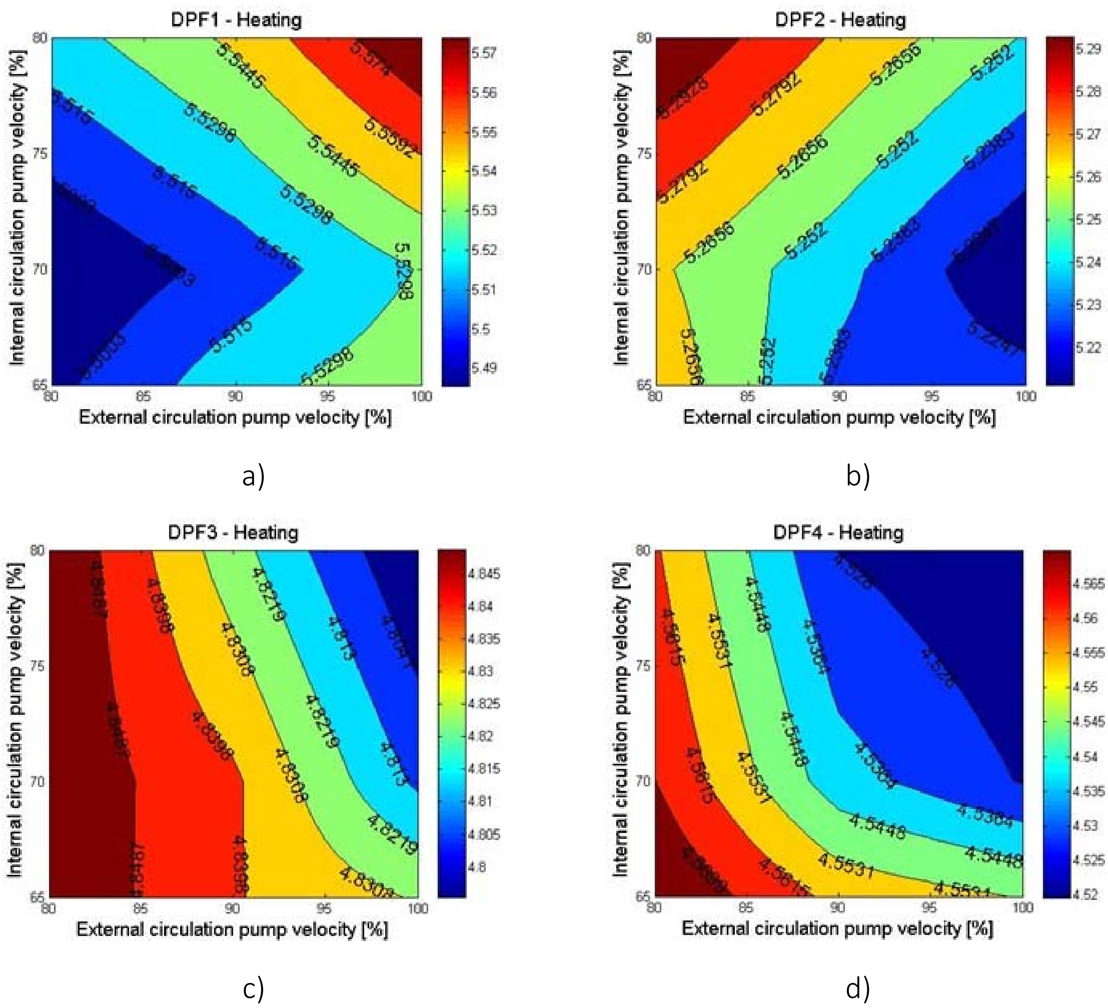


Figure 4.31: Heat pump load factors during the period tests in heating mode.



a)

b)

c)

d)

Figure 4.32: DPF results with circulation pumps velocities variation.

4.4.2 - SPFS RESULTS OF THE HEATING SEASON 2014/2015

The heating season considered is from 17 November 2014 to 3 April 2015. In average, per day, during this heating season around 290 kWh of thermal energy were delivered to the building corresponding to a daily electric consumption of 67 kWh (Figure 4.33).

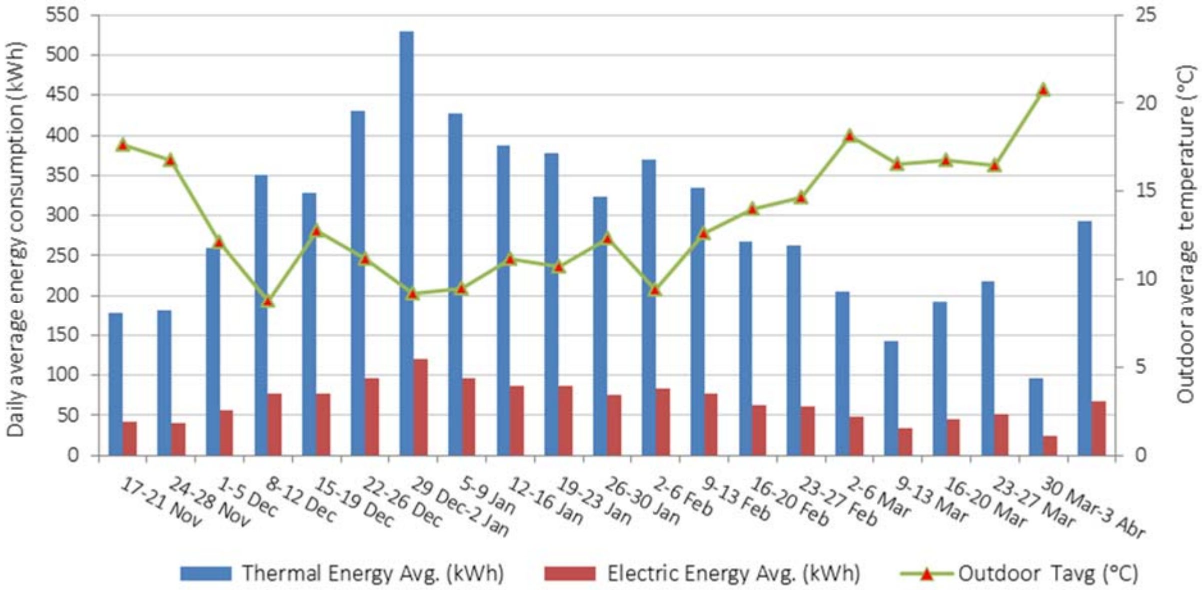


Figure 4.33: Daily average energy consumption and outdoor temperature per week in the heating season 2014/2015.

During the peak winter period, which was 1 December to 15 February, the average outdoor temperature was 10 ˚C (from 8.8 ˚C to 12.7 ˚C) and the minimum average temperature was 8 ˚C (from 5.9 ˚C to 10.2 ˚C). This winter was much colder than the winter 2013/2014, which justify the higher energy consumption. The operation period varied between 11 hours, in the less cold days, and 14 hours, in the coldest days. The HP set point was also 40 ˚C and was reduced to 39 ˚C and 38 ˚C in March.

Figure 4.34 presents the variation of the weekly average performance factors (weekly SPFs) along the 20 weeks that the GSHP system operated in the heating mode.

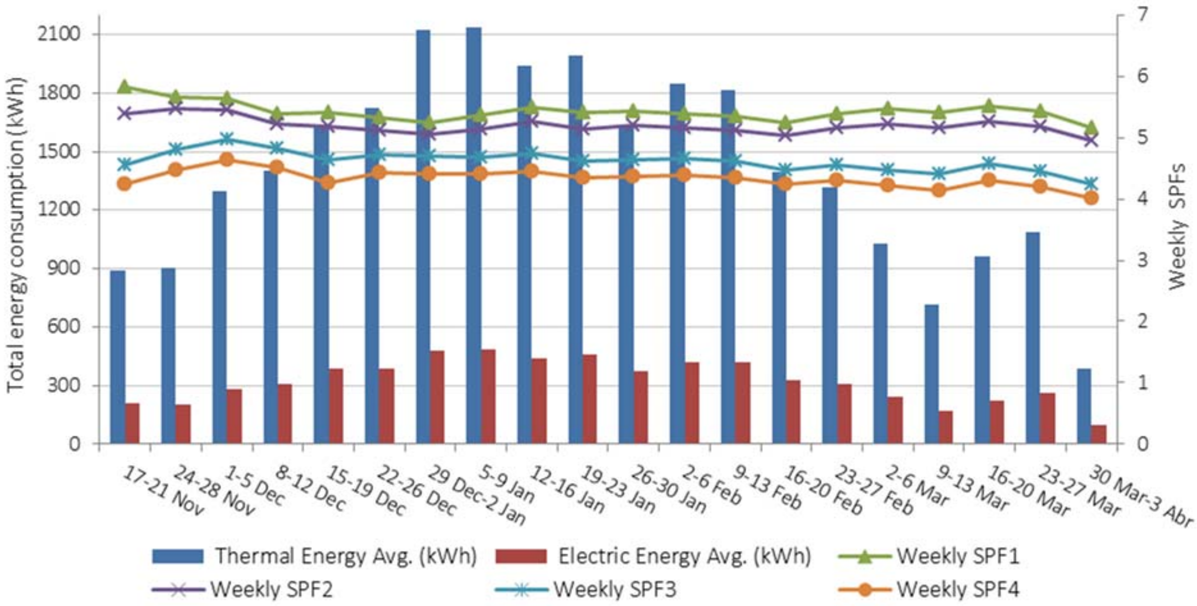


Figure 4.34: Total energy consumption per week and the corresponding weekly SPFs in the heating season 2014/2015.

In this winter season 2014/2015, the seasonal performance factors obtained are very similar to those obtained in the winter season 2013/2014. The SPF3 and the SPF4 have improved, slightly increasing from 4.5 to 4.6 and from 4.25 to 4.33, respectively.

4.5 - GLOBAL ANALYSES OF THE GSHP SYSTEM

The thermal energy delivered to the building during the two heating seasons and the thermal energy extracted from the building during the cooling season (thermal needs) and the corresponding electric energy consumption per month are presented in Figure 4.35.

Although the cooling season 2014 was colder than usual, it can be stated that the 2nd floor of the pilot building has higher heating thermal needs than cooling thermal needs. Table 4.7 summarizes the main data about the three studied seasons.

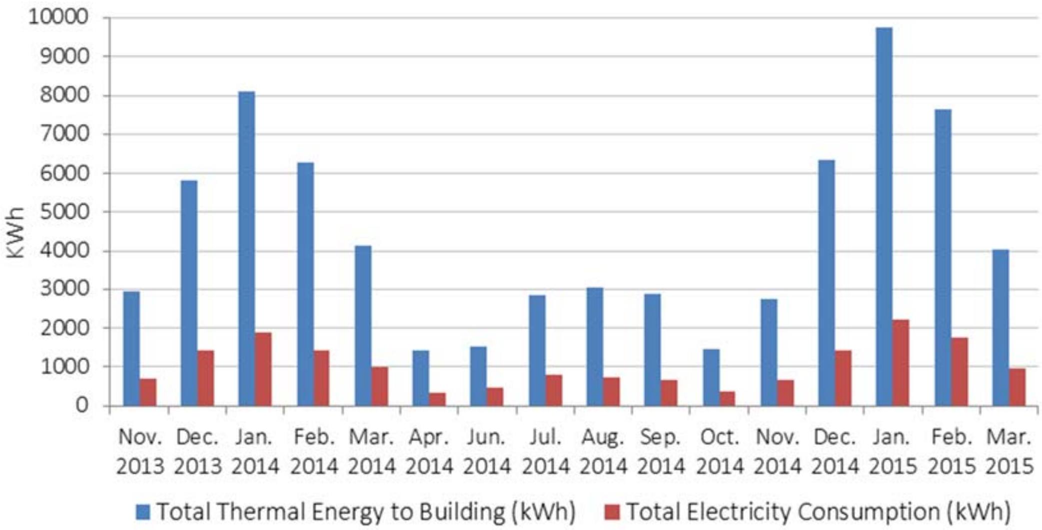


Figure 4.35: Total thermal energy and total electric energy consumption of the GSHP system during the three studied seasons.

Table 4.7: Summary data of the three studied seasons.

Season	SPF 1	SPF 2	SPF 3	SPF 4	Thermal Energy [kWh]	Eletric Energy [kWh]	Tmin Amb [ºC]	Tavg Amb [ºC]	Tmax Amb [ºC]	Tavg out BHE [ºC]	Tavg in BHE [ºC]
Heating 2013/2014	5.40	5.09	4.50	4.25	27184	6250	10.7	14.3	17.6	14.3	11.0
Cooling 2014	6.4	5.8	4.2	3.9	11221	2807	21.2	25.1	29.8	19.9	23.2
Heating 2014/2015	5.40	5.14	4.60	4.33	28204	6472	10.4	13.6	17.2	13.5	11.6

The comparison of the SPFs obtained in the two winter seasons allows concluding that the SPFs in this last winter season 2014/2015 were slightly improved. This can be justified by a higher HP load factor and by a better adjustment of the circulation pumps set points. The breakdown of the electricity consumed by each component of the GSHP system for each season is presented in Table 4.8.

Table 4.8: Breakdown of GSHP system electric consumption by equipments.

	Electricity consumption breakdown			
Season	HP (%)	CPints (%)	CPext (%)	Fancoils (%)
Heating 2013/2014	78	11	5	6
Cooling 2014	60	25	7	8
Heating 2014/2015	80	10	4	6

Lower the thermal needs of the building are, greater the contribution of the internal circulation pumps will be in the total electricity consumption, due to its continuous operation during the operating period of the GSHP system. This effect can be observed in Figure 4.36 by the larger difference between the HP SPF and the global SPF during the cooling season, where the thermal needs were very low.

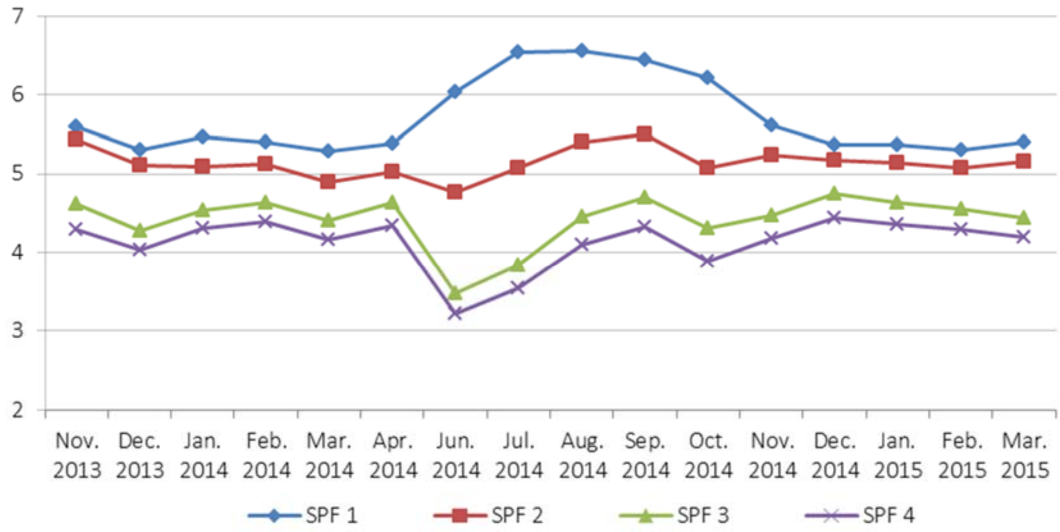


Figure 4.36: Monthly SPFs obtained during the three studied seasons.

The variation of the ground loop water temperature along these last two years is presented in Figure 4.37. The temperature amplitude of the water supplied by the borehole heat exchangers between summer and winter suggests that there is some influence of the outdoor temperature, which can be justified by the length (more than 50 m) of the horizontal pipes that connects the BHEs to the building.

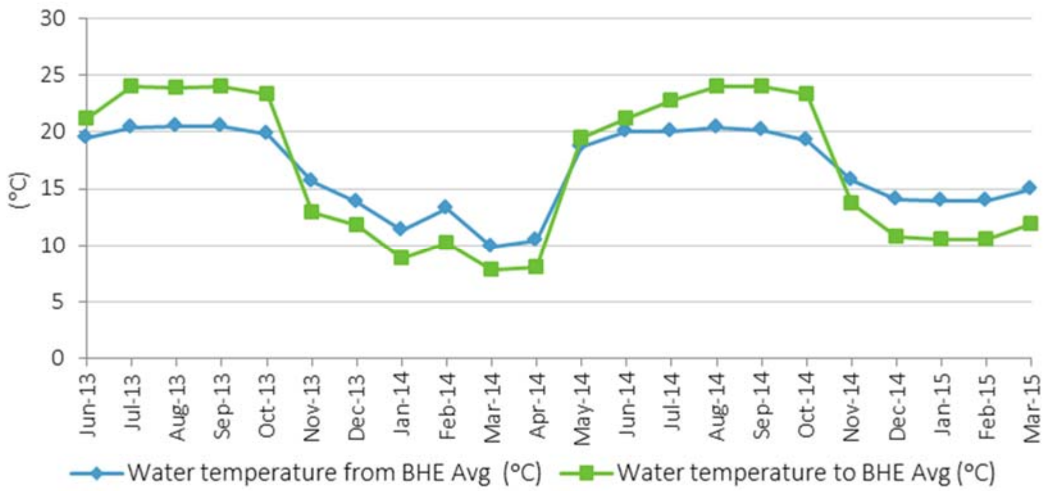


Figure 4.37: Ground loop water evolution during the three studied seasons.

4.6 - CONCLUDING REMARKS

The results of the pilot installation demonstrated the high efficiency of the GSHP systems, taking into account the obtained HP SPF of 5.4 for both heating seasons and 6.4 for the cooling season. Even considering the overall GSHP system performance, which includes the electric consumption of the long piping building thermal energy distribution system, a SPF of 4.4 and 4.5 for the two heating seasons and a SPF of 3.9 for cooling were obtained.

Through the tests performed during summer and winter, it was concluded that the control strategies of adjusting the set point temperature of the GSHP and the circulation pumps velocities are very important. The decreasing or increasing of 1 ˚C in the water supply temperature, depending on the season (heating or cooling), improved in 2.5% the HP efficiency (SPF1). It was also concluded that to ensure thermal comfort to the users, in winter the HP set point can vary from 40 ˚C in cold days to 38 ˚C in the less cold days and in summer can vary from 10 ˚C to 13 ˚C, also depending on the cooling requirements related to the outside climate conditions.

It was also concluded that the circulation pumps velocities have a higher influence in the overall system performance (SPF4) during days with lower thermal needs, since less thermal energy is delivered to the building, but the electricity consumption of the circulation pumps remains almost the same, especially in circuits with no load variation (ground loop and internal loop between HP and the decoupling tank).

For each season the best combination of the circulation pumps velocities (at the condenser side and the evaporator side) which maximizes the GSHP system performance was determined. Analyzing the daily performance factors it was possible to verify that a higher circulation pump velocity (higher flow rate) increases the HP performance (DPF1), but the corresponding increase of the circulation pumps energy consumption penalize the overall GSHP system performance (DPF4).

Applying these setting velocities to the circulation pumps, the overall performance factor (DPF4) has improved 8% in summer and 2% in winter, when compared with the maximum velocity pump settings tested.

CHAPTER 5

GSHPs COMBINED WITH BUILDING THERMAL MASS FOR ELECTRIC LOAD MANAGEMENT

The main purpose of this chapter is to assess the benefits of Ground Source Heat Pump (GSHP) space conditioning systems, based in the results achieved in the pilot GSHP installation. Strategies like preheating the building during cheaper electricity periods and turning off the GSHP system during the electricity peak periods, taking advantage of the thermal mass of the building, are presented and assessed in terms of cost savings to the user. The lumped capacitance method is presented as a possible simple model to be used to calculate the required preheating time of a building. An extrapolation of the application of these strategies to the office buildings at national level is also presented to assess its impact in the electrical grid. The impact of preheating the office buildings as a way to integrate the renewable generation surplus is studied, as well as the use of office buildings space heating as a flexible load to reduce the mismatch between the forecasted and real wind power generation.

This chapter was used as base to the study presented in the following paper submitted to the Energy Conversion and Management Journal: Anabela D. Carvalho, Pedro Moura, Gilberto C. Vaz e Aníbal T. de Almeida, “Ground source heat pumps as high efficient solutions for building space conditioning and for integration in smart grids” (Carvalho et al., 2015a).

5.1 - INTRODUCTION

A typical strategy of Demand Side Management (DSM) is to provide incentives to the consumer to shift flexible loads from peak demand periods to periods with lower energy consumption through very high electricity prices and power penalties in the peak period, when compared with off peak periods (Sun et al., 2013). Arteconi et al. (2013) demonstrated that heat pumps (HPs) have potential for the implementation of DSM strategies and that buildings temperature during the heat pump off periods is strongly affected by the thermal mass of the system considered (underfloor heating maintains room comfort during more time).

In existing buildings, where thermal storage systems are difficult or too expensive to install, the building thermal mass can be used as a way to store energy. Ellerbrok (2014) assessed the available balancing power potential of the thermal energy stored in the thermal mass of two different buildings, concluding that by using the thermal mass of the building, the heat supply can be shifted from the heat demand for a certain period.

The use of the thermal mass in DSM strategies depends on the thermal capacity of the building envelope and on the occupancy period. For example, in buildings not occupied during night, as most service buildings (e.g., offices, schools, administrative services and some health entities), a strategy of preheating the building during night and early morning, and switching off the space conditioning system during peak periods can be used. Using such strategy consumers can profit of cheaper electricity tariffs. A way to increase the potential of structural thermal mass for DSM purpose is preheating the building with higher temperature set points in cheaper electricity periods (off peak periods) and lower the set points during electricity price peaks (Reynders et al., 2013).

Other option is to shut down the system earlier and during off peak periods, using the energy stored in the walls and in other materials to maintain a comfortable temperature, still acceptable for the user. The preheating and the shutting down periods depend on the thermal mass of the building. Hewitt (2012) also examined the ability of a typical local housing stock to retain heat after the heating system is turned off and found that 3 hours of storage is possible, considering a variation of ±2 °C relative to the comfort temperature of 21 °C.

The DSM strategies enabled by the thermal mass can also be used to ensure benefits for the electrical grid management. In fact, one consequence of the large scale use of renewable energy generation is the high penetration level of intermittent energy resources, and this pose a significant challenge in balancing supply and demand. Most of the energy generation by renewable resources (wind and solar) are dependent on weather conditions and present a high variability, being its generation profile very different from the energy demand profile (Moura and De Almeida, 2010a). Nowadays, wind power already covers a large part of the demand for electricity in many areas of Europe and, in the particular case of Portugal, it represents 26% of the total installed power and ensured 24% of the generation in 2014 (REN, 2015). Wind power often presents large variations of generation with extreme ramp rates and large forecasting errors (Moura and De Almeida, 2010b). Additionally, during the night, in rainy and windy winters, the electricity generation from renewable sources exceeds the electricity demand in Portugal.

Therefore, it is crucial to increase the system flexibility in order to ensure the integration of intermittent renewable resources. This can be achieved, for instance, by introducing decentralized energy storage systems and by increasing demand elasticity (shifting or curtailment of loads) using Demand Response (DR). In the context of smart grids, the consumers will evolve from a passive to an active role, providing services to the grid to shape electricity demand according to the available capacity of generation (Moura et al., 2013). In this context, heat pumps can have a key role since they can provide a flexible load by adjusting power consumption, taking advantage of the thermal inertia of the building. The preheating strategy brings benefits to the grid management since less energy will be consumed on the peak period and the surplus of renewable generation, which often occurs during winter nights, can be integrated (Schibuola et al., 2015). In the same way, using the building thermal mass, electric space heating loads can be used as a flexible load to a DR strategy, adjusting the power consumption to compensate the reductions of generation and forecasting errors in wind power (Hedegaard et al., 2013).

5.2 - BUILDING THERMAL RESPONSE MODEL

In this section, a model to assess the thermal response of the test building is presented, which takes into account the building characteristics (thermal capacity and heat loss coefficient), the heating system capacity and the indoor and outdoor temperatures. This model allows the determination of the necessary preheating time, in order to guarantee that the building temperature is already stabilized when the users arrive, ensuring their thermal comfort, and to enable the application of load shifting strategies.

5.2.1 - MODEL DEFINITION

In the case of buildings with significant unoccupied periods, the intermittent heating of the building reduces the thermal energy required, in comparison with the continuous plant operation. To calculate the preheating time it is always necessary to have a model for the building thermal response. Dynamic detailed simulation is complex and too time consuming, requiring detailed input data that is not easily available frequently. For this analysis, it is very important to have a simple robust model, requiring a small number of inputs. The lumped capacitance method (Incropera and Dewitt, 2001) is a simplified method that is normally used to model the transient building heating/cooling (Day et al., 2008). The essence of the lumped capacitance method is the assumption that the temperature of the system is spatially uniform at any instant during the transient process. Thus, the building is considered as an equivalent homogeneous body, with no internal thermal resistances.

Applying the lumped capacitance method to the building, with no internal heat input, from the energy conservation equation and using the Newton´s law of cooling, the following equation is obtained:

$$C\frac{dT_i}{dt} = -U'(T_i - T_o) \tag{5.1}$$

where:

- T_i – Indoor temperature (equivalent uniform temperature) [°C]
- T_o – Outdoor temperature [°C]
- C – Building thermal capacity [kJ/°C]
- U' – Building heat loss coefficient [kW/°C]

Considering that during the initial unoccupied period the building is cooling firstly between time instants t_1 and t_2, integrating Equation (5.1) gives:

$$\int_{t_1}^{t_2} dt = -\frac{C}{U'}\int_{T_{i1}}^{T_{i2}} \frac{dT_i}{T_i - T_o} \tag{5.2}$$

Therefore equation (5.2) can be expressed by equation (5.3) or equation (5.4) which gives the building temperature decay.

$$t_2 - t_1 = -\frac{C}{U'} \ln\left(\frac{T_{i2} - T_o}{T_{i1} - T_o}\right) \tag{5.3}$$

$$\frac{T_{i2} - T_o}{T_{i1} - T_o} = e^{\left[-\frac{U'}{C}(t_2 - t_1)\right]} \tag{5.4}$$

Thus, the temperature of the structure is obtained for any time instant. The time constant for the cooling phase (τ) is given by:

$$\tau = \frac{C}{U'} \tag{5.5}$$

For the preheating period, assuming that the building is heated at a constant rate of heat input (Q_p) and applying again the energy conservation equation and the Newton´s law of cooling, it results:

$$C\frac{dT_i}{dt} = Q_p - U'(T_i - T_o) \tag{5.6}$$

For the heating period (between time instants t_2 and t_3) integrating Equation (5.6) over this time period gives:

$$\int_{t_2}^{t_3} dt = C\int_{T_{i2}}^{T_{i3}} \frac{dT_i}{Q_p - U'\left(T_i - T_o\right)} \tag{5.7}$$

If $Q_p > U'(T_i - T_o)$ the building will heat up, and if $Q_p < U'(T_i - T_o)$ the building will cool down. Equation (5.7) may also be written as:

$$\int_{t_2}^{t_3} dt = -\frac{C}{U'}\int_{T_{i2}}^{T_{i3}} \frac{d\left(-U'T_i\right)}{Q_p - U'\left(T_i - T_o\right)} \tag{5.8}$$

Integrating, the duration of the preheat period is obtained:

$$t_3 - t_2 = -\frac{C}{U'}\ln\left[\frac{Q_p - U'\left(T_{i3} - T_o\right)}{Q_p - U'\left(T_{i2} - T_o\right)}\right] \tag{5.9}$$

For the heating period (between time instants t_2 and t_3, at a constant rate of heat input (Q_p), the temperature evolution given by Equation (5.9) can be written as:

$$\frac{T_{i3}-T_o}{T_{i2}-T_o}=e^{\left[-\frac{U'}{C}(t_3-t_2)\right]}+\frac{Q_p}{U'(T_{i2}-T_o)}\cdot\left\{1-e^{\left[-\frac{U'}{C}(t_3-t_2)\right]}\right\} \tag{5.10}$$

Obviously, Equation (5.10) reduces to Equation (5.4) when $Q_p = 0$. Also, from Equation (5.10) can be noted that as $t_3 \rightarrow \infty$, Equation (5.10) reduces to $T_3-T_0 = Q_p/U'$.

The total time period for cooling and preheating (t_3–t_1), corresponds to the building unoccupied period. Assuming that $T_{i3} = T_{i1}$ (set point temperature), Equations (5.3) and (5.9) can be summed, resulting in:

$$t_3-t_1=-\frac{C}{U'}\ln\left[\left(\frac{T_{i2}-T_o}{T_{i1}-T_o}\right)\cdot\frac{Q_p-U'(T_{i1}-T_o)}{Q_p-U'(T_{i2}-T_o)}\right] \tag{5.11}$$

Knowing the building unoccupied period, $t_{unocc} = (t_3-t_1)$, the theoretical building temperature T_{i2}, that corresponds to the HP switch-on, for a given HP size and outdoor temperature is obtained by Equation (5.12).

$$T_{i2}=\frac{(T_{i1}-T_o)\cdot Q_p\cdot e^{\left[-\frac{U'}{C}(t_3-t_1)\right]}}{Q_p-U'(T_{i1}-T_o)\cdot\left\{1-e^{\left[\frac{U'}{C}(t_3-t_1)\right]}\right\}} \tag{5.12}$$

Then, the corresponding duration of the preheat period (t_3-t_2) can be calculated by Equation (5.9).

5.2.2 - THERMAL PARAMETERS OF THE COIMBRA BUILDING

The thermal building parameters required to the analysis are the effective building thermal capacity (C) and the building heat loss coefficient (U'). The building heat loss coefficient can be obtained in a quasi-steady state test. This test must be done at night to avoid solar gains, and during quasi steady weather conditions to minimize the interference from heat storage effects. The building heat loss coefficient U' is calculated as the heating rate divided by the indoor/outdoor temperature difference.

Several tests of this type were made, including the test with a larger indoor/outdoor temperature difference (12 °C) and a smaller outdoor temperature variation during the night

(1.2 °C). From this test, the value U' = 1.97 kW/°C was obtained.

The effective building thermal capacity can be obtained through a cool-down test. In this test, the building is heated and maintained above the outdoor temperature, and heating is shut off at a certain time. If the test is performed in the absence of solar (during the night) and internal gains, the variations in infiltration and in ambient temperature are small enough, and therefore the temperature decay is a characteristic of the building thermal capacity. By fitting the decay to an exponential, a time constant can be determined. Knowing the heat loss coefficient (U'), the effective thermal capacity (C) can be determined. There are several limitations to this approach, as referred by Subbarao et al. (1985). One is that, except for relatively light buildings with well-distributed masses, the decay cannot be represented by a single exponential, but by a sum of exponentials. As a result, the single time constant obtained by a fit depends on the choice of the region of the decay curve.

Several cool-down tests were performed during the night, measuring the decay of the indoor air temperature. As an example, Figure 5.1 shows the calculation of the building time constant, plotting $\ln[(T_i – T_o)/(T_{i0} – T_o)]$ versus time and making a simple linear regression, for a cool-down test performed during the night, measuring the indoor air temperature. For this test, and using the complete cooling period, it was obtained a building time constant of 1/0.0008 min = 1250 min = 20.8 h. As expected, the temperature decay curve cannot be represented by a single exponential (Figure 5.1). At the beginning of the cooling process, the air (that has a small thermal capacity) is hotter than the interior walls, so the air temperature decays more rapidly, until reaching the interior thermal equilibrium. Thus, at the calculation of the building time constant for the simplified thermal model it is more appropriate to exclude this initial period. Assuming 100 min for this initial stabilization period, a building time constant of 1/0.0006 min = 1667 min = 27.8h is obtained, as also shown in Figure 5.1.

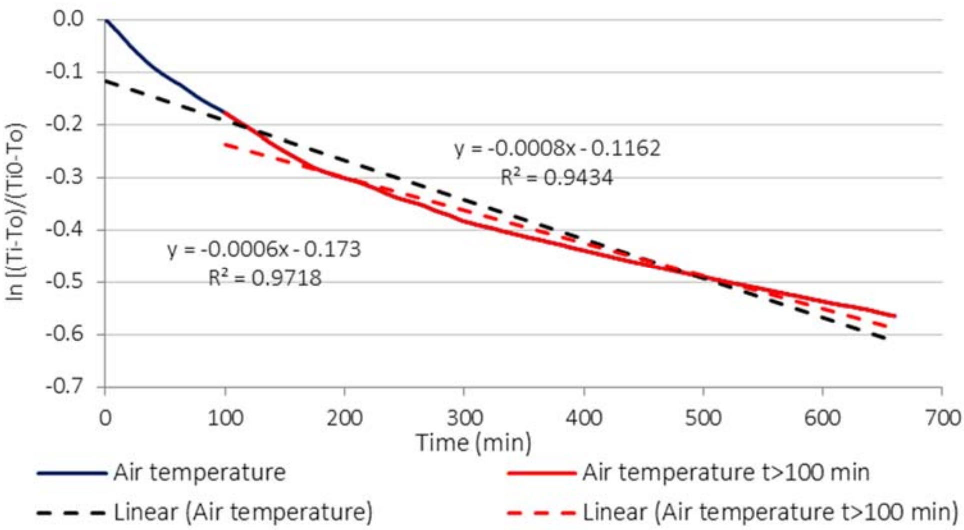


Figure 5.1: Building time constant determination.

Figure 5.2 shows the comparison between measured air temperature and the calculated temperature for the entire period using the simplified model with a time constant of 27.8 hours, and it can be concluded that for t>100 min, the temperature predictions are quite good. It is important to note that using this simplified model, the steady state building thermal mass equivalent temperature of 20 °C corresponds to an indoor air temperature of approximately 22 °C, as shown in Figure 5.2. Therefore, as the heat loss coefficient U’ is the heating rate divided by the indoor/outdoor temperature difference, this value (to be used in this model) must be corrected to U’ = 2.23 kW/°C.

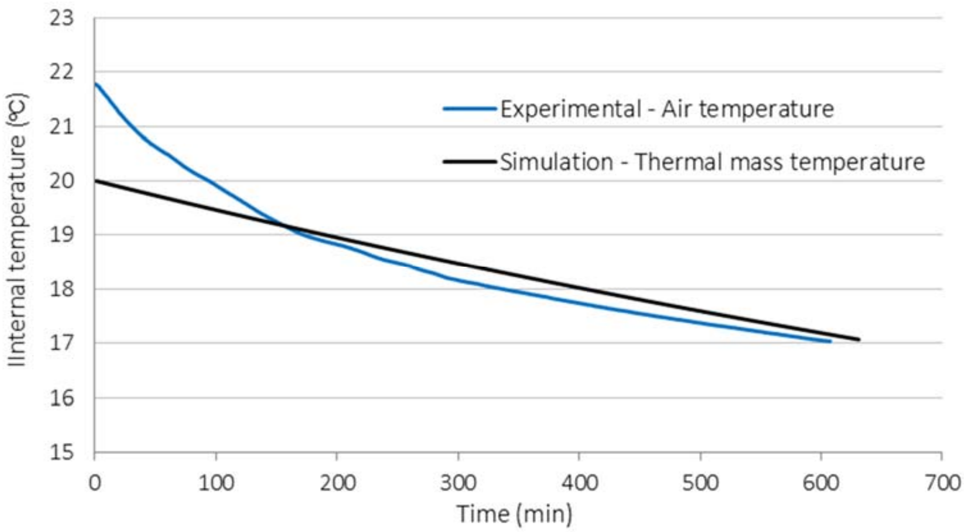


Figure 5.2: Comparison between air experimental and simulated temperature decay.

Finally, the equivalent building thermal capacity is calculated using Equation (5.13), giving:

$$C = \tau \cdot U' = 27.8 \times 3600 \times 2.23 = 223\,\mathrm{MJ/^oC} \tag{5.13}$$

5.2.3 - PREHEATING TIME ESTIMATION

The preheating time depends on the HP thermal power. Figure 5.3 shows this dependence for the Coimbra building, considering 15 hours as the building unoccupied period, a steady state building thermal mass equivalent temperature of 20 °C, and 11 °C for outdoor temperature. It is clear that increasing the HP output the required preheating time reduces. Also, by increasing the HP output, the preheating energy decreases, because the 24-hours mean building temperature is lower and the 24-hours average rate of building heat loss is lower. For the particular case of HP thermal power of 40 kW, the calculated preheating time is 6.5 hours.

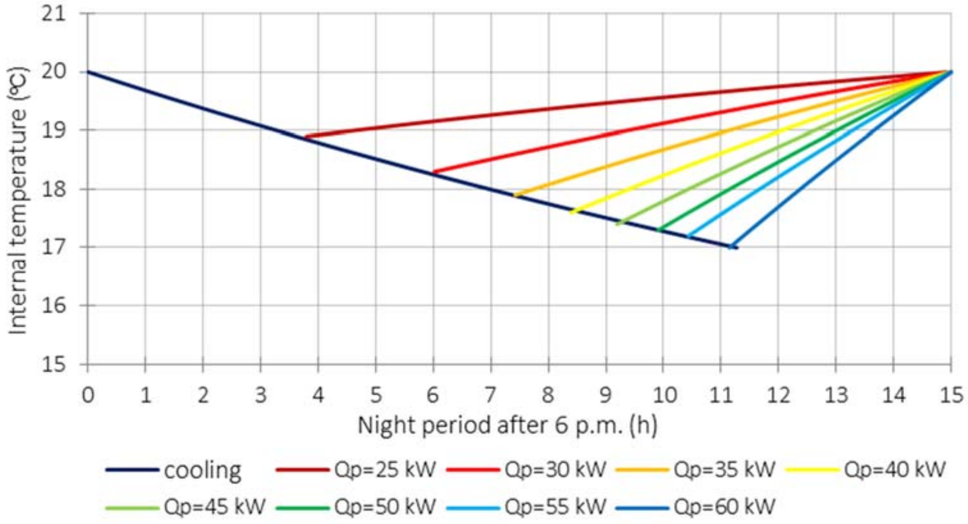


Figure 5.3: Influence of HP power on preheating time of the pilot building.

The preheating time also depends on the outdoor temperature. Figure 5.4 illustrates this dependence for the pilot building, considering a fixed HP thermal power of 40 kW. It can be observed that there is a range of HP switch-on points for different outdoor temperatures.

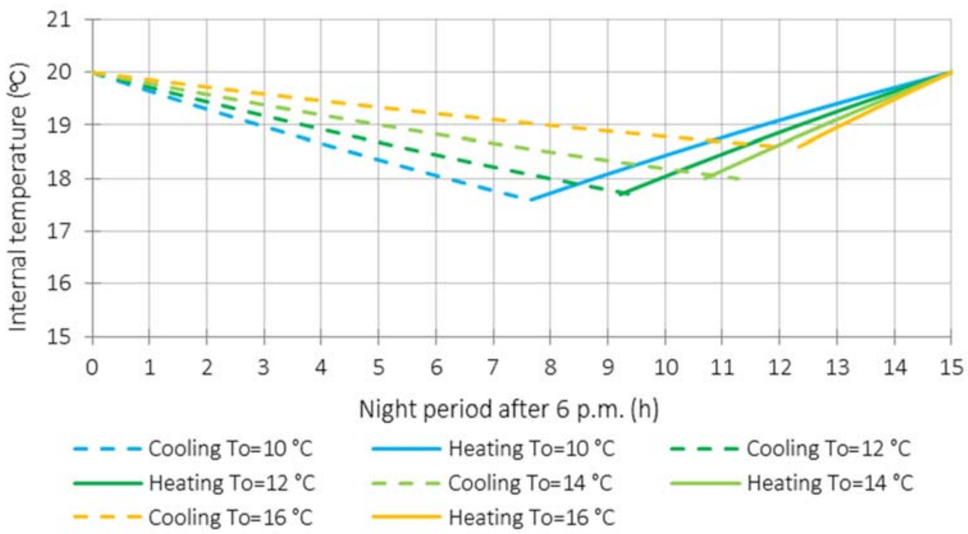


Figure 5.4: Influence of outdoor temperature on the building preheating time.

5.3 - DEMAND SIDE MANAGEMENT STRATEGIES

The aim of this section is to demonstrate that heat pumps combined with building thermal mass can be used as a flexible load and that heating load management strategies can bring significant costs savings at the consumer side. The associated impacts at the grid side are assessed in Section 5.4.

5.3.1 - DSM STRATEGIES DESCRIPTION

The DSM strategies proposed consist in shutting down the GSHP system earlier and during the electricity peak periods, and in preheating the building, taking advantage from building thermal mass and from lower electricity prices during off peak periods, while guaranteeing thermal comfort to the users.

The electricity prices contracted to this building during 2014 were used as base to estimate the costs savings associated to the application of the DSM strategies. Table 5.1 presents the variation of the electricity prices along the 24 hours in winter. The peak periods are very expensive when compared with off peak periods and have an additional penalty applied to the monthly average power demand used during the peak period of 0.4737 €/kW per day. This demand penalty corresponds to an additional cost of 0.1201 € for each kWh of electricity consumed during the peak period, increasing the total cost during peak hours to 0.3298 €/kWh.

Table 5.1: Contracted electricity prices by the facility in 2014 (winter).

Period type	Period hours	Prices €/kWh
Peak	9 a.m. to 10:30 a.m. 6 p.m. to 8:30 p.m.	0.3298
Middle Peak	8 a.m. to 9 a.m. 10:30 a.m. to 6 p.m. 8:30 p.m. to 10 p.m.	0.1211
Off Peak	10 p.m. to 2 a.m. 6 a.m. to 8 a.m.	0.0849
Special off Peak	2 a.m. to 6 a.m.	0.0747

To apply these strategies, the building thermal mass is a key factor to enable a long time period with the temperature near to the comfort level. Figure 5.5 shows the measured air temperature decay curves in the pilot building after shutting down the GSHP at 6 p.m., under different outdoor temperature conditions. As it can be seen, after one hour with the system turned off, the air temperature decreases about 1 °C and after 1.5 hours the decay is about 1.5 °C, even for the day with the lowest outdoor temperature (9.4 °C). These results show that the building has a good thermal inertia, which delays the indoor air temperature decrease.

For an air room temperature of 22 °C, it is assumed that a decrease of 1 to 1.5 °C of the indoor air is acceptable for most users. A comfortable room temperature not only depends on air temperature, but also on the mean radiant temperature of the inside surfaces whose decrease is much slower than the air temperature. By the thermal response model presented in Section 5.2.1, it was possible to determine that, after shutting down the heating system during 1.5 hours, the building inside surfaces temperature decreased around 0.5 °C for an outdoor temperature of 10 °C (Figure 5.4).

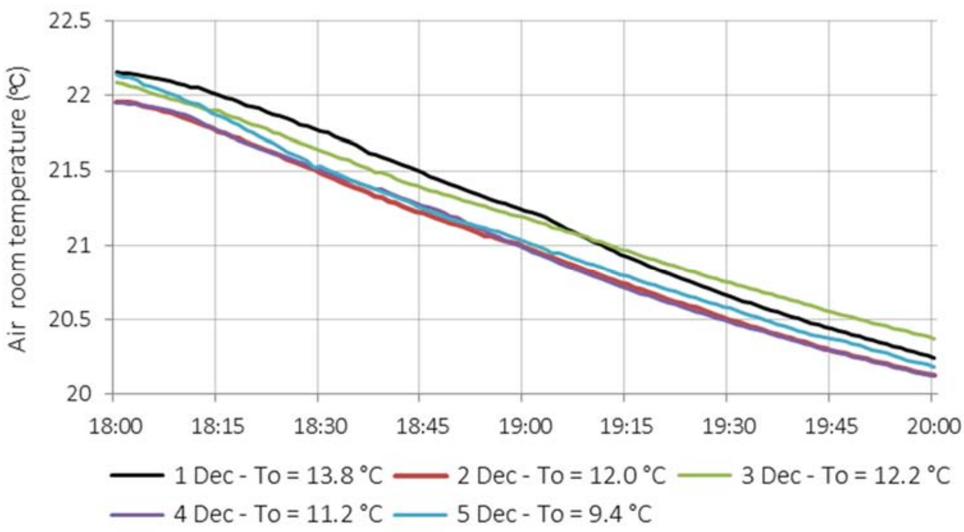


Figure 5.5: Air room temperature decay after turn off the system at 6 p.m.

Based on these results, the curtailment strategy can be used to avoid electricity consumption during peak electricity prices and to shutting down the GSHP earlier than the ending of the occupation period, in order to reduce the electricity costs.

Two different preheating strategies are proposed and assessed for the building. One strategy is to ensure a lower preheating level, depending on the outdoor temperature in winter period, just to ensure that room's temperature is near to the comfort level, when the users arrive. In such strategy, the preheating time depends on the outdoor temperature, a larger preheating time for days with lower outdoor temperatures being needed. This strategy ensures that the higher power demand occurs during the off-peak periods, reducing the electricity consumption during the first peak period (9 a.m. to 10:30 a.m.), therefore reducing costs.

Another strategy is to apply a longer preheating time to guarantee that in the first peak period (9 a.m.), the building temperature is already stabilized at a higher temperature (more 0.5 °C) than the required by users. In this situation it is possible to turn off the system during all the morning peak period (1.5 hours), avoiding the electricity consumption during the period with higher costs.

The strategies proposed and assessed in this section are a combination of the curtailment strategy with the preheating strategy.

5.3.2 - SAVINGS WITH DSM STRATEGIES

In this section, the impacts in terms of energy consumption and costs savings are assessed for the following combined strategies:

Strategy 1: Some preheating is carried out in order to avoid that the higher consumption happens during the morning peak period and shutting down the system 1 hour before the end of the occupation period, avoiding the second peak period.

Strategy 2: Longer preheating to turn off the GSHP during the morning peak period and shutting down the system 1 hour before the end of the occupation period.

The impacts of DSM Strategies 1 e 2 application are assessed using as baseline the system operation coincident with the users occupancy period (8 a.m. to 7 p.m.). Two different baselines were used, one for mild winter weeks (Baseline 1) and another for colder winter weeks (Baseline 2). To assess the costs savings, the daily (weekly average) electric load diagrams were used.

The first strategy was applied in two different weeks with different thermal needs. The week of 27-31 January was less cold than the week of 17-21 February. The preheating time in these situations was defined based in the analysis of the historic monitored data and of the outdoor temperature forecasts.

The daily average electric load diagrams of these two weeks, with the implementation of the DSM Strategy 1 and the corresponding baselines, are presented in Figure 5.6 (week 27-31 January) and in Figure 5.7 (week 17-21 February). These electric diagrams will be used to evaluate the associated grid impacts with the application of this first strategy in Section 5.4.2.

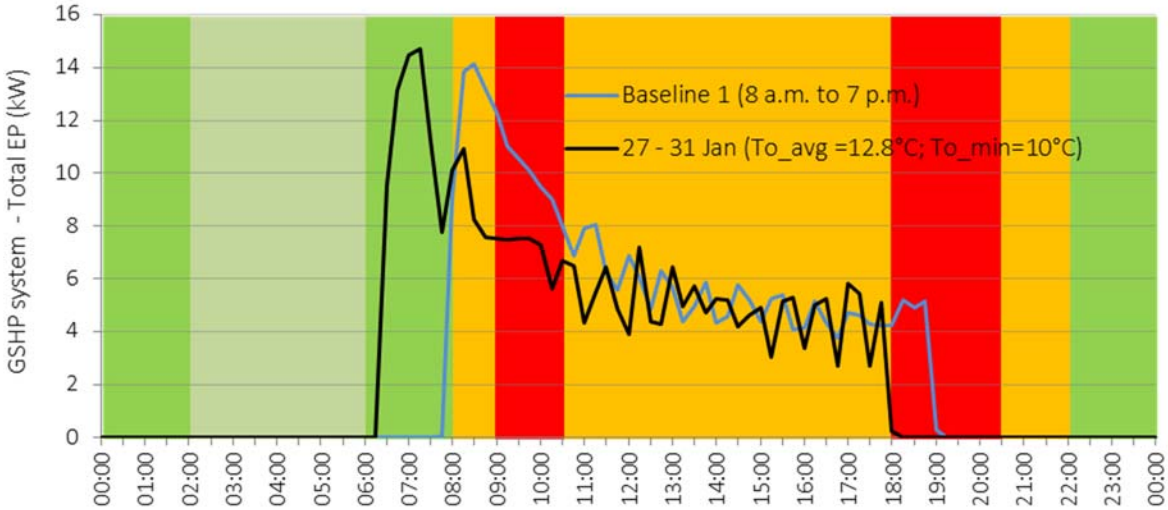


Figure 5.6: Average electric load diagrams with and without DSM Strategy 1 (mild winter week).

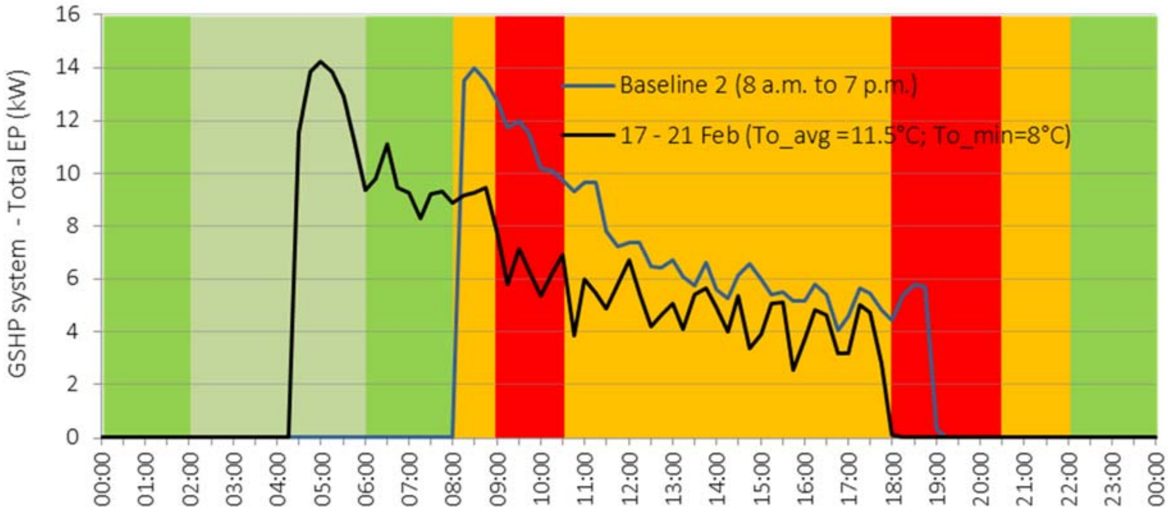


Figure 5.7: Average electric load diagrams with and without DSM Strategy 1 (cold winter week).

The daily energy consumption in Baseline 1 is about 74 kWh, which corresponds to daily costs of 13.2 €. The Baseline 2 is related to the colder week, when more energy is consumed (83 kWh) and the daily costs also increase (14.5 €). Applying the DSM Strategy 1, as represented in Figure 5.6 and Figure 5.7, although the energy consumption increases, costs savings of 17% (week 27-31 January) and 19% (week 17-21 February) are achieved.

To apply Strategy 2, the longer preheating time was estimated based on the building thermal response model (Section 5.2.3). For an average outdoor temperature of about 11 °C and turning off the system at 6 p.m. the day before, in order to have the building temperature stabilized at 9 a.m. with a higher set point (more 0.5 °C), the model estimates a preheating period of 7 hours. The application of this preheating time and curtailment of the GSHP system during the two peak periods of the day in a cold winter week is represented in Figure 5.8.

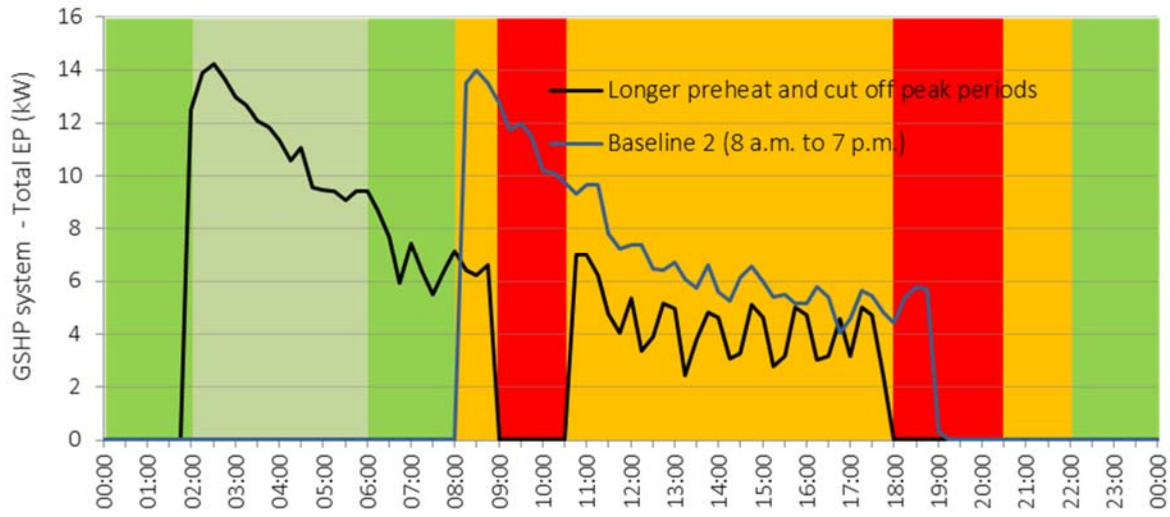


Figure 5.8: Average electric load diagrams with and without DSM Strategy 2 (cold winter week).

As expected, with a longer operation period and, consequently, with higher thermal energy losses, the energy consumption increases. In terms of costs savings, Strategy 2 is a very attractive strategy at the consumer side, since significant costs savings of about 34% can be achieved.

5.4 - HEAT PUMPS IN OFFICE BUILDINGS AS A FLEXIBLE LOAD

Office buildings, mainly its electric heating systems, present a high potential for electric load management and their potential integration as flexible loads in smart grids. In this section, two different scenarios of load management are assessed:

- The use of the preheating strategy to absorb the surplus of Renewable Energy Sources (RES) generation during winter nights.
- The use of the curtailment strategy, taking advantage of the thermal inertia of the building to implement a Demand Response action to compensate the mismatch between the forecasted and the generated wind power.

5.4.1 - ENERGY CONSUMPTION IN OFFICE BUILDINGS CHARACTERIZATION IN PORTUGAL

In order to assess the impact of load management strategies in the heating systems of office buildings on the Portuguese electric load diagram, the total energy consumption and the load diagram of space heating in such buildings was estimated. Based on data available in the Odyssee database (Odyssee, 2015) it was found that electricity is the main energy consumed in services buildings with about 75% of the total energy consumption, and that office buildings consume about 27% (Figure 5.9) of the total electricity consumption in service buildings and about 10% of the total electricity consumed at national level.

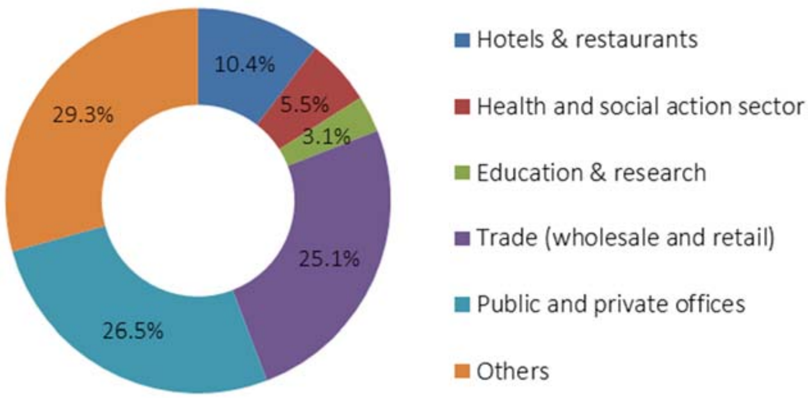


Figure 5.9: Electricity share by type of service buildings in Portugal, in 2012 (based on Odyssee (2015)).

In offices, the main loads are lighting, space conditioning (heating and cooling) and office equipment (computers, printers, network servers). Therefore, the average electricity breakdown by end-use for total services buildings considered in a study presented by Pardo et al. (2012) (excluding hot water, cooking and electric motors) was applied to office buildings (Figure 5.10). It was estimated that space heating represents a share of 19% in the total electricity consumption in office buildings, which corresponds to about 820 GWh *per annum*.

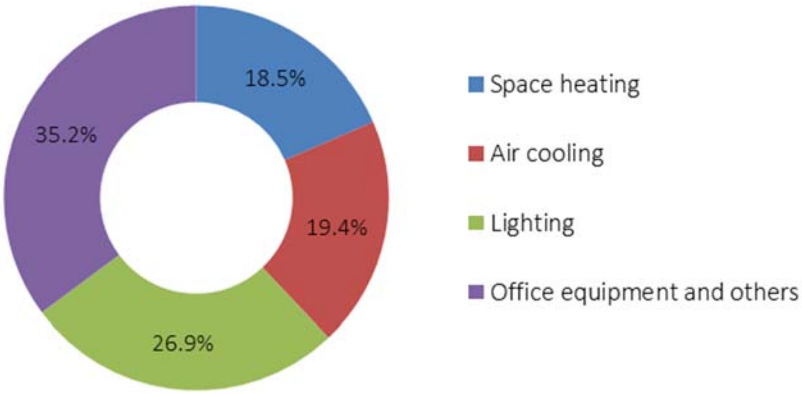


Figure 5.10: Electricity share by end use in office buildings in Portugal, in 2012 (based on Pardo et al. (2012)).

To guarantee a significant impact on the national load diagram, load management should be ensured by controlling the loads in a centralized way. In Portugal, thermal comfort only became a requirement for users since a few years. Therefore, in some of the office buildings, mainly in small buildings, the HVAC systems were gradually installed. In these situations the predominant technology is room air-to-air heat pumps, which cannot be easily controlled in a centralized way by the electrical grid. For this reason, it was considered that only 50% of the electric space heating has centralized control. Within this universe, it was assumed that half of the heating technologies have constant thermal power (one or two levels), like the test GSHP system presented in this thesis (Section 3.2). In this way, only 25% of the total universe of office buildings was considered. The extrapolation factor was calculated by comparing the monitored electric consumption of space heating during an entire winter season of the experimental installation with the total consumption of space heating in office buildings. This factor was applied to the average electric load diagram baselines of the experimental installation from the correspondent week, presented in Section 5.3.2, with an operation period between 8 a.m. and 7 p.m., without preheating. The average baselines electric load diagrams for space heating for the week of 17-21 February and for the week of 27-31 January are presented in Figure 5.11.

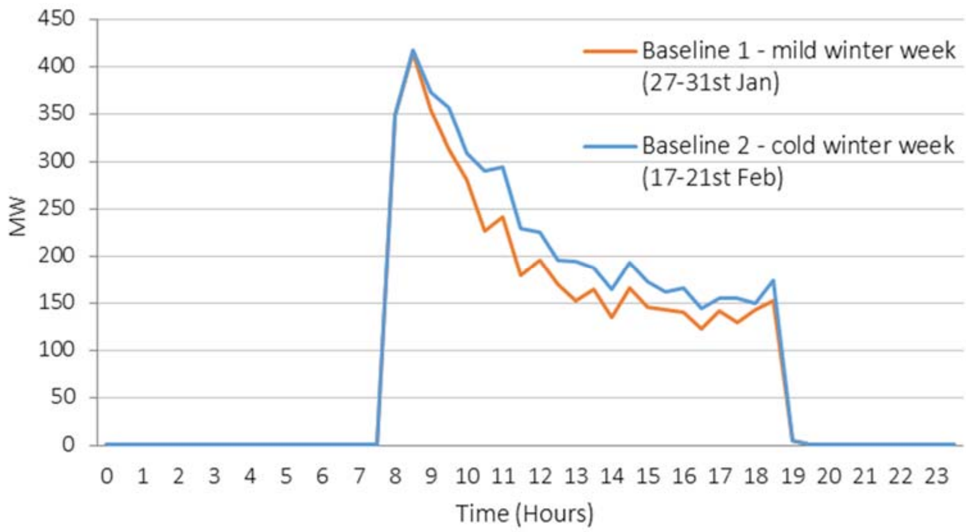


Figure 5.11: Electric load diagram baselines for office buildings space heating (cold and mild winter).

5.4.2 - Preheating to the Integration of the Surplus RES Generation

Renewable Energy Sources are already the main source of electricity in Portugal, with 64% of the total installed power, and ensuring 62.7% of the electricity demand in 2014. Table 5.2 presents the actual installed power and the generated energy during 2014 by source. As can be seen, hydro and wind power ensure together 57.3% of the installed power and 57% of the generated energy in 2014.

Table 5.2: Installed power and generation in Portugal - 2014.

	Power		Generation	
	[MW]	(%)	[GWh]	(%)
Coal	1756	9.8	11099	22.6
Natural Gas	3829	21.5	1468	3.0
Reservoirs	2681	15.0	6429	13.1
Run of River	2588	14.5	8268	16.8
Small Hydro	415	2.3	1508	3.1
Wind	4541	25.5	11813	24.0
Photovoltaic	393	2.2	588	1.2
Biomass and CHP	1637	9.2	7949	16.2
Total	**17840**		**49122**	

However, the output of hydro and wind power is driven by environmental conditions, inherently variable and outside the control of the generators or the system operators. Their output presents large seasonal variations between summer and winter and the variations are also present on shorter time scales, such as on hourly basis. Additionally, unlike conventional capacity, wind-generated electricity cannot be reliably dispatched or perfectly forecasted. Hydropower can be dispatched, but 53% of the installed power is run of river and small hydropower plants, which have a very limited degree of dispatchability.

This situation can lead to a high mismatching between the electricity demand and the available RES generation and one consequence is the excess of generation during the winter, mainly during the night. Due to the lower consumption during the night and the low degree of control over most of the renewable generation (reservoirs only represent 26% of the total of hydro and wind power), the generation level of wind and hydro power often exceeds the demand. This is also aggravated by the level of base generation, using coal power plants, that must be maintained and that cannot be turned on or off in response to the quick variations of renewable generation due to the characteristics of such power plants (large start-up time and low ramp-up rate). In such situations, the surplus of energy is traded in the Iberian Energy Market with a nearly zero price, leading to strong economic disadvantages.

Figure 5.12 and Figure 5.13 present two examples of recent typical situations of renewable generation surplus during the night. In the first case (February 19, 2014) there is excess of renewable generation during all the night (between 0 and 8 a.m.). In the second case (January 27, 2014) the excess of generation is first used in the pumping process of reservoirs. However, due to the limit of the pumping capacity, the use of the electricity surplus in pumping decreases after 5 a.m., leading to an excess of renewable generation in the period between 5 and 8 a.m.

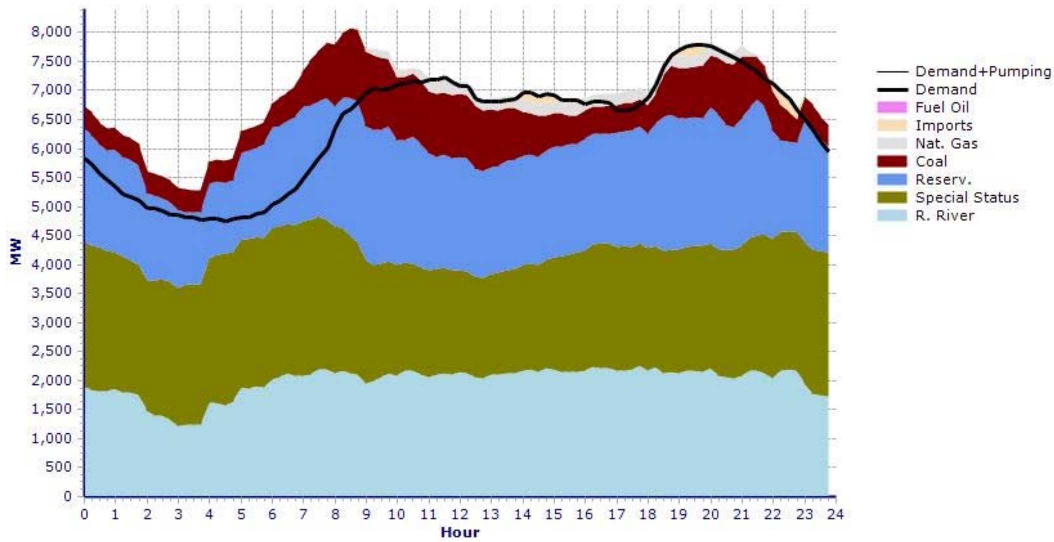


Figure 5.12: National load diagram on February 19, 2014 (REN, 2015).

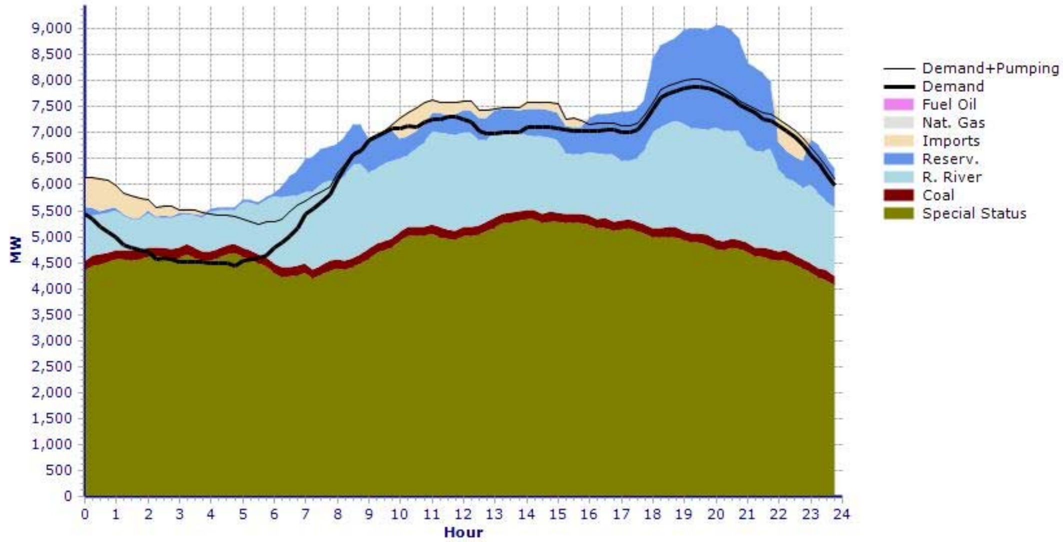


Figure 5.13: National load diagram on January 27, 2014 (REN, 2015).

The control of space conditioning in buildings can have a key role to minimize this situation of surplus of RES generation. The preheating of buildings can bring not only economic advantages to the consumer, taking advantages of lower tariffs, but it can also create positive impacts on the electrical grid management. Due to the increase of demand during valley periods, part of the surplus of renewable generation would be avoided, therefore reducing the problems in the grid management and simultaneously minimizing the negative economic impacts on the energy market. Additionally, in this situation, a substantial part of the heating electricity consumption is supplied only with renewable generation, therefore reducing greenhouse gases emissions.

To assess the potential impact of the preheating on the RES generation surplus, the experimental GSHP installation was controlled to ensure the preheating of the building with two different starting hours: one scenario starting at 4:30 a.m. and other scenario at 6:30 a.m. Each scenario was assessed during one week: the first, between 17 and 21 February 2014, and the second between 27 and 31 January 2014. The average load diagrams were then extrapolated to the office buildings in Portugal, using the same strategy as to the baseline (presented in Section 5.4.1). Figure 5.14 and Figure 5.15 present the extrapolated load diagrams for the preheating starting at 4:30 a.m. and 6:30 a.m., respectively.

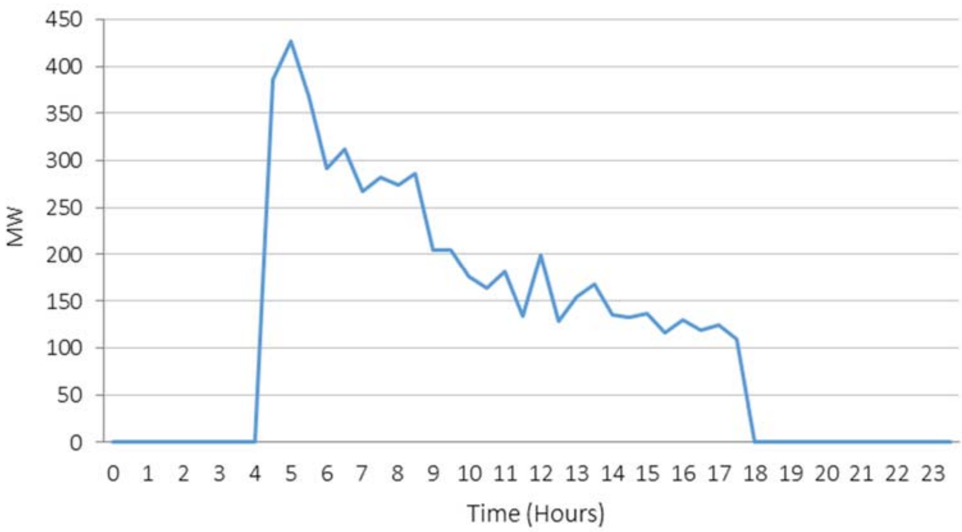


Figure 5.14: Load diagram (extrapolated to the office buildings in Portugal) – preheating starting at 4:30 a.m.

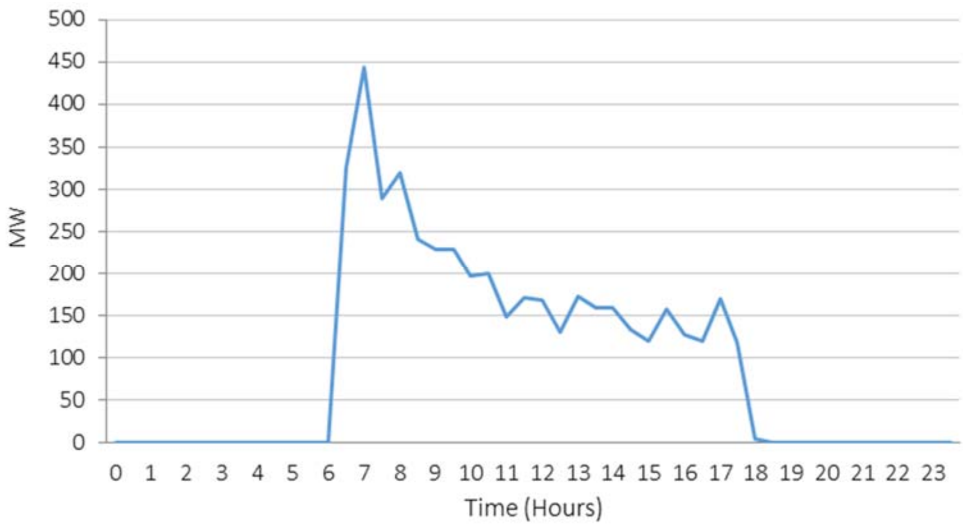


Figure 5.15: Load diagram (extrapolated to the office buildings in Portugal) – preheating starting at 6:30 a.m.

The impact of the preheating on the load diagram was then assessed, comparing the obtained diagrams with the baseline. Figure 5.16 and Figure 5.17 present the impact of the preheating starting at 4:30 a.m. and 6:30 a.m. As can be seen, there is an increase on the demand until 8 a.m. followed by a decrease.

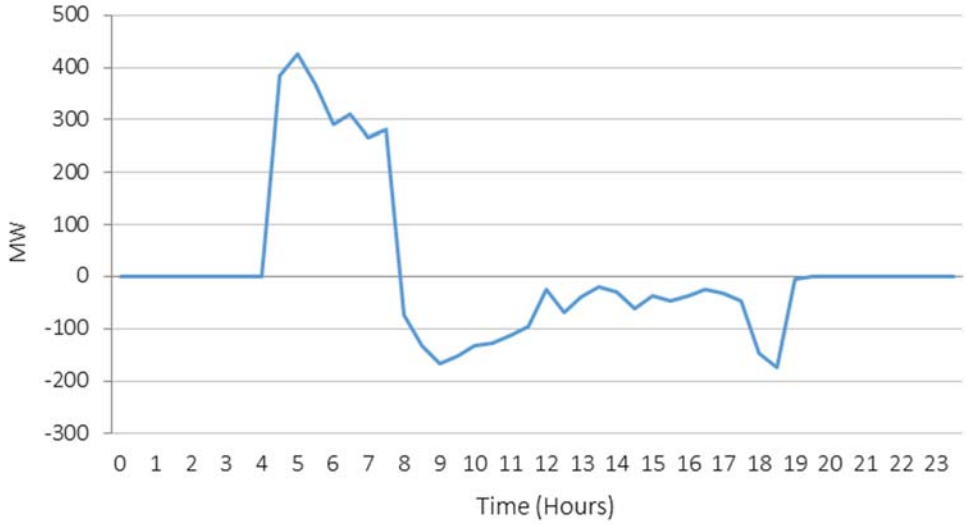


Figure 5.16: Impact on the extrapolated load diagram of the preheating – starting at 4:30 a.m.

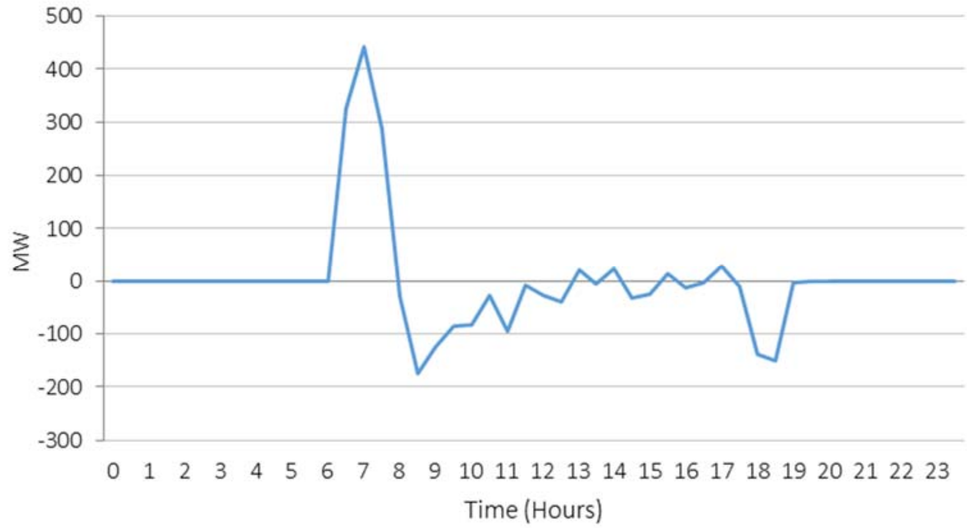


Figure 5.17: Impact on the extrapolated load diagram of the preheating – starting at 6:30 a.m.

These impacts on the load diagram were then applied on the total demand (including pumping) of the national load diagrams in the two examples of surplus on the renewable generation. The impact of the preheating starting at 4:30 a.m. was used on the diagram from February 19, 2014 (surplus of generation between 0 and 8 a.m.) and the preheating starting at 6:30 a.m. was used on the diagram from January 27, 2014 (surplus of generation between 5 and 8 a.m.). It is important to point out that the extrapolated load diagrams were obtained exactly in the same week as the used in the national load diagrams, therefore ensuring the same weather conditions.

Figure 5.18 and Figure 5.19 present the impact of the preheating on the national load diagram in February 19, 2014 and January 27, 2014, respectively. As can be seen, in both situations the demand (including pumping) increases during the periods with surplus of renewable generation, reducing such surplus, without a major impact on the demand during the other periods.

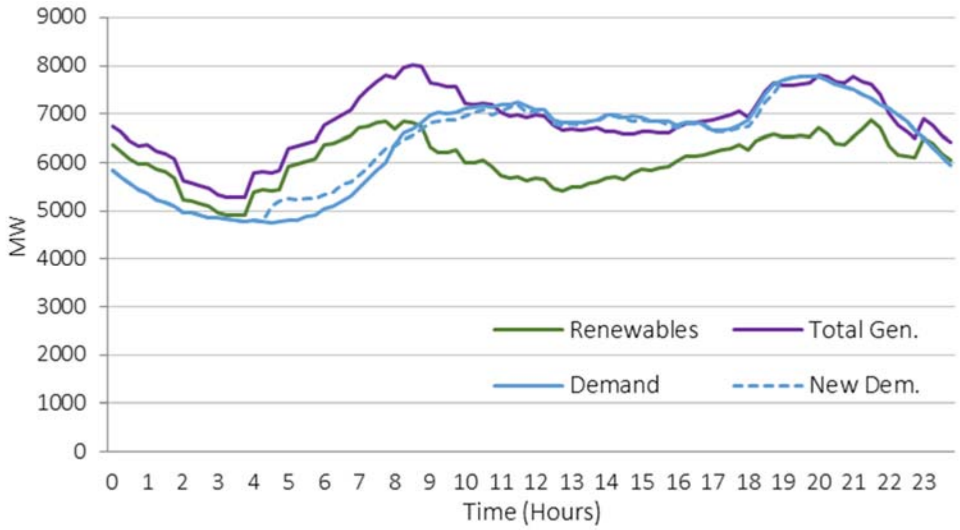


Figure 5.18: Impact of preheating on the national load diagram on February 19, 2014.

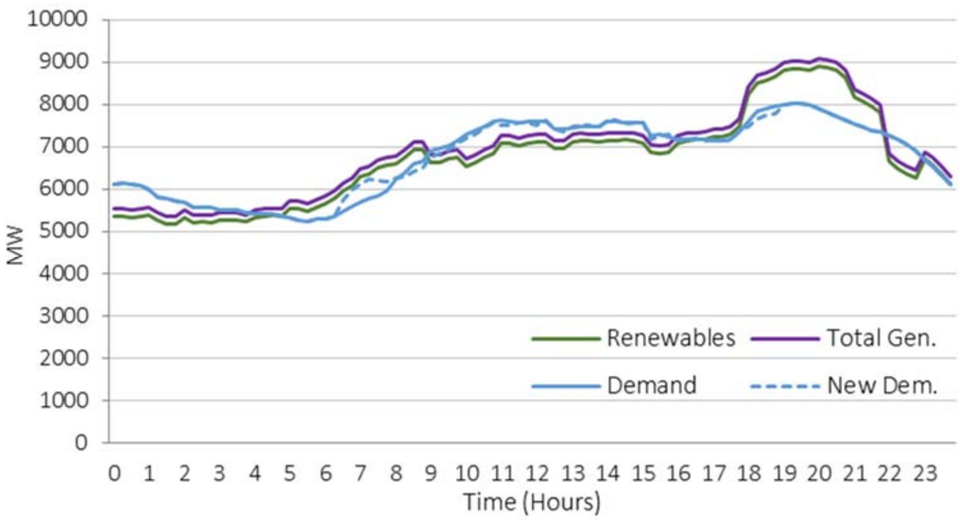


Figure 5.19: Impact of preheating on the national load diagram on January 27, 2014.

Table 5.3 quantifies the impact of the presented preheating strategies on the reduction of renewable energy surplus. As it can be seen, on February 19, 2014, the total surplus was reduced by 18.9% and during the period of preheating the surplus was reduced by 27.5%. On

January 27, 2014, the total surplus was reduced by 30% and during the period of preheating the surplus was reduced by 42.2%. Just by controlling the preheating in 25% of the office buildings it is possible to avoid an important part of the renewable generation surplus. Therefore, expanding this control strategy to more office buildings and other types of buildings could avoid such surplus. Additionally, in the context of smart grids, it is possible to remotely control the starting time of the preheating to match the renewable energy surplus by using the forecast of the renewable generation to define the starting time of the preheating and number of buildings to include in it.

Table 5.3: Impact of preheating on the renewable energy surplus.

		Baseline	Preheating	Reduction	
		[GWh]	**[GWh]**	**[GWh]**	**(%)**
February 19	0:00-8:30 a.m.	23027.7	18664.1	4363.6	18.9
	4:30-8:30 a.m.	15856.8	11494.2	4362.6	27.5
January 27	5:00-8:30 a.m.	6246.2	4370.2	1876.0	30.0
	6:30-8:30 a.m.	4442.3	2566.7	1875.6	42.2

5.4.3 - Demand Response for the Integration of Wind Power

Since the wind is determined by random meteorological processes, it is inherently variable and, therefore, wind power presents high variations not only seasonally, but also on shorter time scales, namely on hourly basis. Several extreme ramp rates of reduction on wind power generation were already recorded in Portugal and elsewhere. Additionally, the supply of power from wind turbines is stochastic in nature and the actual power is more or less proportional to the third power of the wind velocity. Therefore, wind power cannot be perfectly forecasted since a small error on the wind velocity leads to a high error on the forecast of the power.

Figure 5.20 presents one example of a recent event (January 6, 2014) with a large ramp rate of generation reduction and a large forecasting error. In this situation, there is a reduction of 1190 MW (40%) between 13 and 17 hours (average reduction of 300 MW/hour) and simultaneously there are large errors on the forecast, reaching a maximum error of 29.5% (476 MW).

As far as security of supply is concerned, the most severe problems due to the wind power intermittence occur in the peak load hours, since the largest part of the available system resources to deal with the intermittence is already used and a sudden reduction of the wind power production can have critical consequences on the system reliability. Thus, instead of acting in the supply side, to avoid the most severe intermittent situations, Demand Side Management measures can be promoted to achieve consumption reductions and, mainly, to balance supply and demand. Therefore, the use of efficient space conditioning solutions is very important to avoid such problems.

However, in cases of high wind power penetration, the energy consumption reduction during the peak hours may not be enough, due to the large generation variations, in situations with a large forecasting error. In such situations it will be very important to have Demand Response (DR) technologies to "force" consumption reductions at near real time, in the precise moment in which the critical situations occur. In a smart grid context this can be implemented by controlling the space conditioning loads, with a change in the temperature of the system or even with a temporary shutdown.

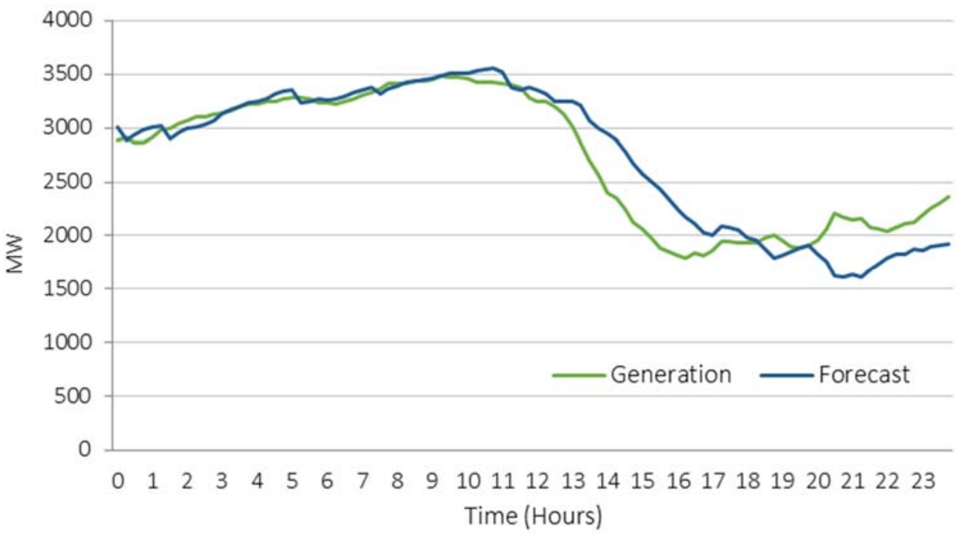


Figure 5.20: Wind generation and forecast on January 6, 2014.

As it was previously demonstrated for the experimental GSHP installation, the shutdown of the system space conditioning during one hour can be done without a relevant impact on the comfort level. Therefore, such strategy was implemented by considering the shutdown of the system between 1 and 2 p.m. The reduction of consumption achieved with the shutdown can therefore be considered as a virtual increase of the wind generation. Figure 5.21 presents the achieved impact and, as can be seen, during such period the difference between the real generation and the forecast can be substantially decreased (a reduction of 55.5% was achieved). This is achieved just by controlling 25% of the office buildings, and therefore by expanding this control strategy to all office buildings and other types of buildings a total compensation of the error would be possible.

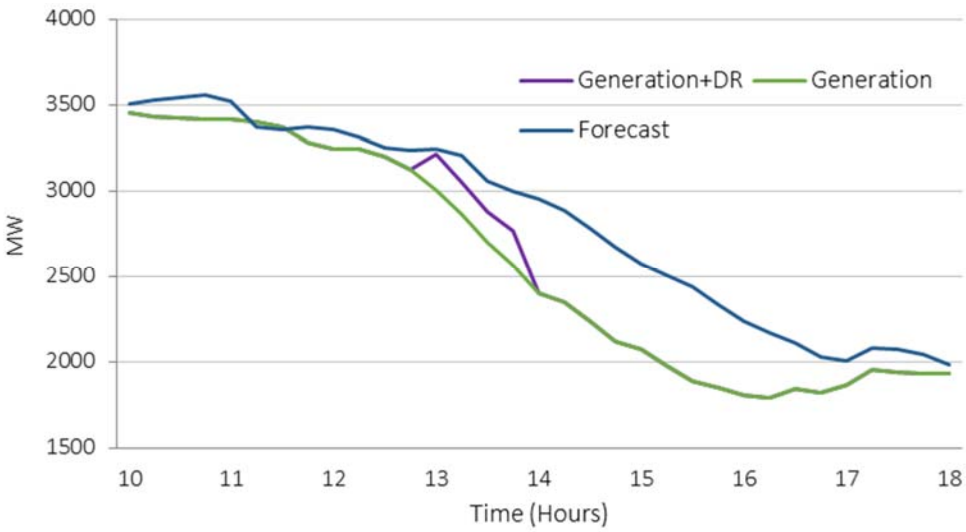


Figure 5.21: Impact of DR on the reduction of wind forecasting errors on January 6, 2014.

The DR action was implemented in the beginning of the variation and not in the moment of the highest error, since the main problems to the electrical grid occur in the beginning of the variation, due to the need of a fast answer from the system to compensate the lost generation capacity. If the system does not have enough flexibility to react immediately, e.g., with the increase of hydropower generation, it can have critical consequences on the system reliability. Therefore, concentrating the load reduction in the beginning of the variation, to give enough time to increase the generation ensured by other sources, is usually the best solution to avoid major reliability problems. The implementation of DR presents as a main benefit the increase on the electrical grid reliability, but it can also represent an economic benefit for the consumer, since the users that accept the implementation of a DR action are usually compensated with a reduction of tariffs or a payment by each DR action implemented.

5.5 - CONCLUSIONS

Using the experimental GSHP system, it was demonstrated that heat pumps, when combined with the building thermal mass, can be used as a flexible load to balance supply and demand, and to allow the reduction of operation costs at the consumer side. For this purpose, several strategies of heating loads management were proposed and tested.

To assess the thermal response of the building, a model based on the lumped capacitance method was applied. This model allowed determining the required preheating time as a function of the building parameters, building unoccupied period, desired indoor temperature, outdoor temperature and HP thermal power.

The indoor air temperature decay was analyzed after shutting down the heating system one hour before the ending of the occupation period. After turning off the system for one hour, the indoor air temperature only decreased 1 °C (for 1.5 hours the decrease is 1.5 °C). Assuming that this temperature decrease is acceptable for users, the curtailment strategy can be used for periods up to 1.5 hours.

To guarantee thermal comfort to the users when they arrive and to take advantage from the lower electricity prices, two preheating strategies were applied. The main purpose is that at 9 a.m. the building temperature is near to the steady state to minimize or even to avoid the energy consumption during the morning peak period. Different preheating times were tested, taking into account the outdoor temperature, associated to a curtailment strategy at the end of the day and costs savings of 16-19% were achieved. Higher costs savings can be achieved (about 34%) with longer preheating periods in order to be possible to avoid energy consumption during the morning peak, as well as by shutting off earlier the heating system, one hour before the ending of the working day, to avoid the second electricity peak period.

The impacts on the national electric diagram of applying the preheating strategy and the curtailment strategies were assessed. These strategies were extrapolated to other office buildings with similar occupancy periods, assuming that only 25% of the office buildings can be controlled. The preheating strategy of the office buildings was used as a way to integrate the surplus of renewable energy generation. Based on the two examples studied, it was found that significant part of the renewable energy generated during night and early morning (19-30%) can be absorbed. It was also demonstrated that by shutting down temporarily the space conditioning it is possible to substantially compensate the variations of wind power generation and forecasting errors during the first hour of variation (55.5% in the presented scenario), giving enough time to adapt the other resources of the electrical system.

The role of high efficient heat pumps combined with building thermal mass (as flexible load) can play a key role for sustainable and efficient building space heating, as well as contribute to integrate electricity generated by intermittent renewable resources and to improve grid management using load leveling and Demand Response strategies.

CHAPTER 6

GSHP CARBON EMISSIONS AND PRIMARY ENERGY REDUCTION POTENTIAL FOR BUILDINGS HEATING IN EU AND IN PORTUGAL

The main focus of this chapter is to assess the primary energy and CO_2 emissions savings by switching Natural Gas (NG) boilers to high efficient Heat Pumps (HPs) for space heating in the European Union (EU). A scenario where NG space heating will be progressively phased out in EU until 2050 is presented. The impacts of the large scale penetration of high efficiency heat pumps in EU are evaluated, in the following aspects: decrease in carbon emissions, decrease in primary energy consumption, decrease of NG dependency and increase of the Renewable Energy Sources (RES) share contribution. A similar scenario was developed for Portugal and the same impacts were also assessed.

The study presented in this chapter was published online in the following paper "Ground source heat pump carbon emissions and primary energy reduction potential for heating in buildings in Europe – results of a case study in Portugal", published online in the Renewable and Sustainable Energy Reviews journal (Carvalho et al., 2015b)

6.1 - INTRODUCTION

The European Union has limited oil and natural gas resources and the foreign dependency has been increasing fast, with a projected 70% natural gas imported share of the consumption by 2030. The EU energy dependency is at the top of the EU agenda, becoming more relevant due to the recent political turmoil in Eastern Europe, which has again highlighted the geopolitical risks of imports and the vulnerability of the EU economy. In 2012, the EU demand for natural gas was around 4570 TWh, with a 65% net import share (Eurostat, 2015). Around 30% of the natural gas consumption is imported from Russia and around half of it crosses Ukraine (Ecofys, 2014).

The building sector in EU is the main responsible for the increase in natural gas dependency, since it is responsible for 61% of the total imported NG. In 2012, building space heating was responsible for 45% of the total final NG consumption and for 29% of the total gross inland NG consumption in the EU (this total includes the NG used for electricity generation) (Eurostat, 2015). Therefore, the building sector assumes an important role, not only in the achievement of the EU sustainability targets for 2030 and 2050, but also in the decreasing of the natural gas dependency.

Denmark is the only EU country that exports natural gas, and it has one of the most ambitious programs towards sustainable energy and climate targets. Denmark aims to be fully independent of fossil fuels by 2050. In particular, gas and oil boilers are forbidden for new buildings since 2012 and a gradual phase out is planned for these technologies (Danish Government, 2011).

A fuel switching strategy by increasing the use of renewable energies and high-efficiency heat pumps to replace fossil fuels boilers is one of the promising strategies to be applied in the buildings sector. The latest generation of electric Ground Source Heat Pumps (GSHPs) has the advantage of being a very efficient technology (with one unit of electricity it is possible to produce 4 to 6 units of heating or cooling). Simultaneously, electricity is becoming a cleaner form of energy, due to the massive increase of renewable energies (mainly wind and solar) observed in EU in the last decade, leading to a lower fossil fuels consumption in electricity generation and to a progressively lower carbon content in electricity (Eurelectric, 2012).

Several studies have already presented heat pumps as a very promising technology to contribute to the building sector decarbonization for the heat demand in the EU, namely in the building sector (Morrone et al., 2014; Sarbu and Sebarchievici, 2014; Self et al., 2013; Ecofys, 2013a; Bayer et al., 2012). These studies were analyzed with more detail in Chapter 2. Within these related studies, the analysis presented in the Ecofys study (Ecofys, 2013a) is the most similar to the study presented in this thesis chapter, in terms of the type of impacts assessed. The Ecofys study (Ecofys, 2013a) also includes primary energy savings and the renewable energy sources contribution for the heat pumps implementation scenario, but the results presented are not only for NG space heating, but for total space heating, water heating and space cooling, using a forecast on heat pump implementation only until 2030 in eight EU countries.

The analysis carried out in this thesis chapter is focused in the replacement of the NG space heating boilers by state of the art high efficiency HPs (this is a realistic assumption based on the expected efficiency evolution of HPs in Europe and elsewhere over the next decades). In the timeframe to 2050, a more ambitious decarbonization of the electricity supply is assumed. The next sections address several key factors necessary to this type of fuel switching analysis as follows: Carbon emissions factor for the electricity sector in Europe for the year 2050; Characterization of space heating consumption and respective scenarios of evolution in Europe until the year 2050; Progressive penetration of HPs in Europe, to achieve full replacement of NG boilers by the year 2050.

6.2 - ELECTRICITY CO_2 EMISSIONS TREND IN EU

The electricity and heating sectors are responsible for nearly 30% of all greenhouse gas emissions in Europe, with CO_2 being the dominant polluter. Although electricity generation increased by roughly one third from 2004 to 2010, the corresponding increase of CO_2 has been less than 1% (Eurelectric, 2012).

Thus there is a clear decoupling between demand increase and carbon emissions. It can be stated that the strong investment of the EU to increase the share of RES (mostly wind and solar power) has been very important in the decrease of the CO_2 emissions related to electricity generation. The RES share in the EU electricity generation increased from 14.3% in 2004 to 23.5% in 2012 (Eurostat, 2015). Besides the increasing RES share, the fuel switching from coal to NG, and the power plants efficiency improvement have also contributed to decrease CO_2 emissions.

According to EURELECTRIC (Union of the Electricity Industry), by 2020, 44% of the total power plant capacity will be renewable based, generating around 31% of the electricity in Europe (Eurelectric, 2012). It is expected that the carbon intensity of electricity will be decreasing even more in the future due to the projected decarbonization of the EU energy system to at least 80% below 1990 levels by 2050 (European Comission, 2011b). Figure 6.1 shows the recent and projected electricity generation carbon intensity (g CO_2/kWh) decreasing over time. Until 2025 the data was adapted from the EURELECTRIC Power Statistics (Eurelectric, 2010); between 2025 and 2050 the carbon intensity for electricity generation was estimated considering the goal to reduce the carbon emissions by 80%, the same as defined for the whole EU energy system until 2050.

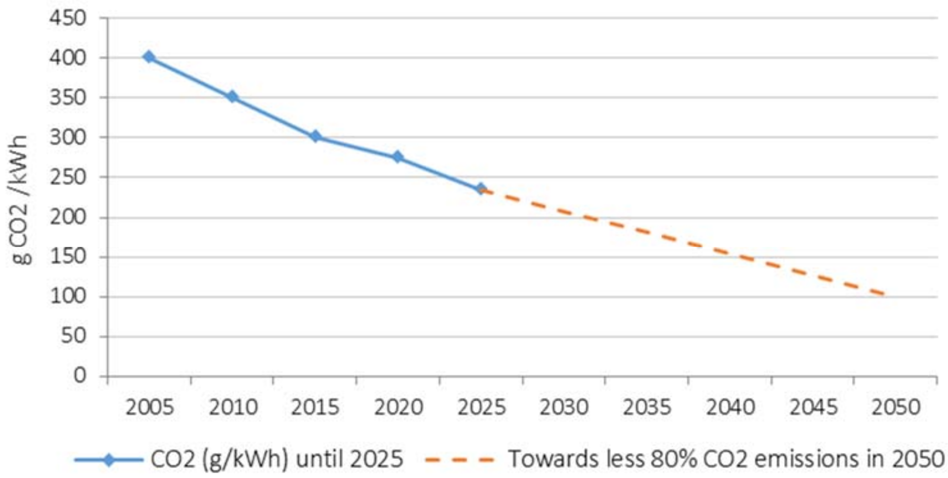


Figure 6.1: Recent evolution and projection of CO_2 emissions per unit of electricity produced in EU.

The carbon intensity of the electricity presented before is an average for EU as a whole, but it is also important to look to specific carbon intensity of each country, since this factor strongly depends on the energy mix (energy source and energy conversion technology efficiency) used for electricity generation. The CO_2 emissions savings from switching NG boilers by heat pumps will be much higher in countries where the electricity generation is based on nuclear or RES, such as hydro, solar or wind, which have near zero CO_2 emissions, than in countries where electricity generation is based on fossil sources, such as coal or fuel oil, which have the highest CO_2 emissions. For each country it was calculated the minimum Seasonal Performance Factor (SPF), from which the HPs bring benefits in terms of CO_2 emissions (Figure 6.2), compared to NG boilers. This SPF value was calculated dividing the electricity CO_2 emissions factor of each country, taken from International Energy Agency data for 2011 (IEA, 2013b), by a global emissions factor of 212.6 g CO_2/kWh for natural gas boiler, which was determined considering the emission factor for natural gas of 202 g CO_2/kWh (European Commission, 2007) and assuming a NG boiler efficiency of 95%.

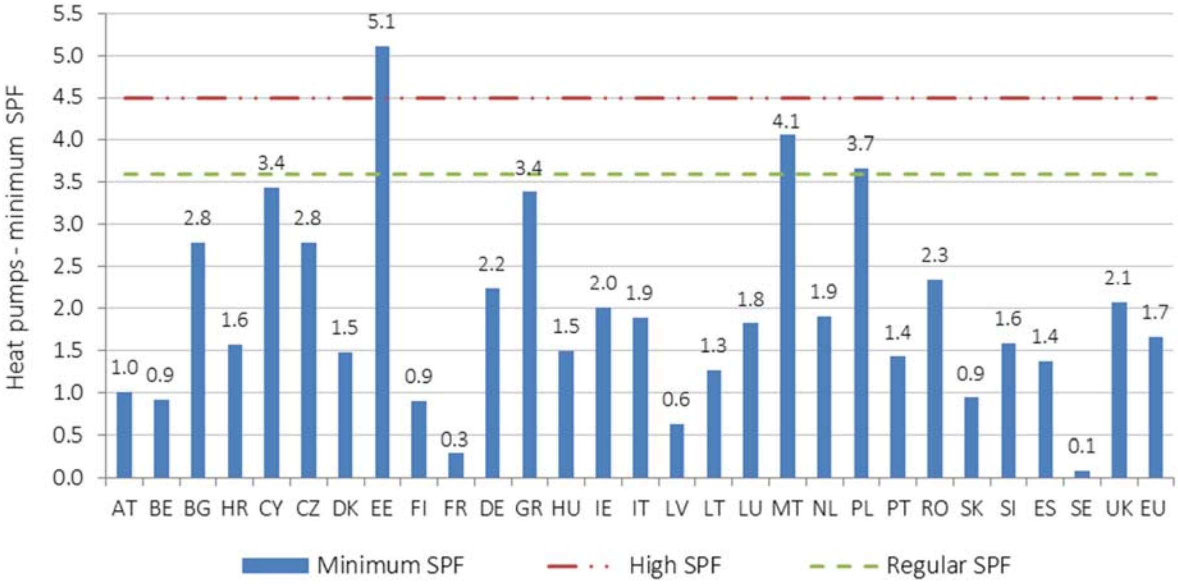


Figure 6.2: Minimum heat pump SPF per country to have CO_2 emissions savings in space heating when switching NG boilers by HPs.

Even considering a non-ambitious SPF of 3.6, as published by the European Heat Pump Association (EHPA) in 2009 (EHPA, 2009), significant emissions savings can be achieved for almost all European member states, except for a few countries (Estonia, Malta, Poland, Cyprus and Greece), due to their large coal contribution for electricity generation. However, even in these countries, due to the on-going decarbonization of the electricity generation sector, the situation has improved significantly in recent years, in terms of emissions saving expectations from heat pumps. In Greece, for example, lignite use in electricity generation mix has been reduced from 47.7% in 2012 to 40.4% in 2013, while renewables share has increased from 16.8% in 2012 to 27.1% in 2013, resulting in minimum SPF to drop from 2.45 in 2012 to 2.13 in 2013. In the remaining countries, CO_2 emissions savings can vary from 23% (Czech Republic) to 92% (France).

6.3 - SPACE HEATING CHARACTERIZATION AND SCENARIOS

The determination of the NG consumption for space heating in European buildings is required in order to assess the CO_2 emissions and primary energy reduction potential of switching NG boilers by HPs. In this section, the space heating characterization by energy type is presented for the European Union in 2012, based on the most recent data available from Eurostat

(Eurostat, 2015) and from Odyssee data base (Odyssee, 2015). Buildings heating demand scenarios for 2030 and for 2050 in EU, used in the Heat Roadmap studies (Connolly et al., 2012; Connolly et al., 2013), will be used to obtain the reference scenario for buildings space heating in this study.

6.3.1 - SPACE HEATING CHARACTERIZATION IN EU BUILDINGS

According to Eurostat (Eurostat, 2015), in 2012, the buildings sector (residential and non-residential buildings) consumed around 40% of the total final energy in the EU (non-residential buildings – 1730 TWh and residential buildings – 3360 TWh). The buildings consumption share of the total final consumption by energy type had the following values:

- 64% of the total derived heat (mostly district heating, which also includes heat from combined heat power plants);
- 61% of the final natural gas consumption;
- 60% of the total electricity available for final energy consumption in EU.

In 2012 buildings have consumed 40% of the total NG primary energy consumption in the EU (Eurostat, 2015). Considering the energy mix consumption, as presented in Figure 6.3, the largest shares in the building sectors are from natural gas (35%) and electricity (33%). The Liquefied Propane Gas (LPG) is included in the oil products.

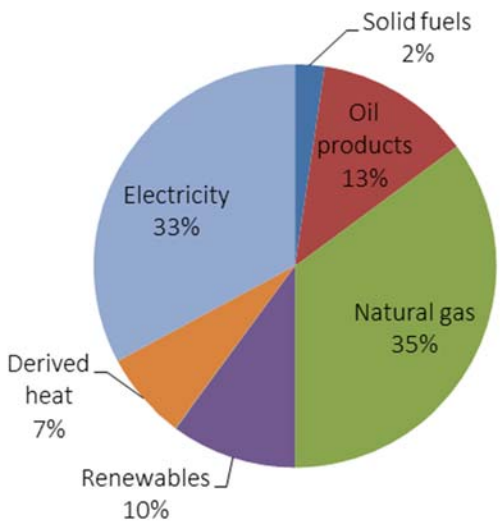


Figure 6.3: Contribution by energy type to the buildings total energy consumption in EU in 2012 (based on Eurostat data).

Based on the Odyssee data (Odyssee, 2015), the contribution of the NG for space heating in buildings in the EU as a whole, in 2012, was calculated for the residential and non-residential sectors. These shares were applied to the 2012 energy balance from Eurostat (Eurostat, 2015), as explained along this section. In Figure 6.4, the total space heating share and the NG space heating share, in the total final energy consumption in the residential sector for all each of the EU countries and for the EU, in 2012, are presented.

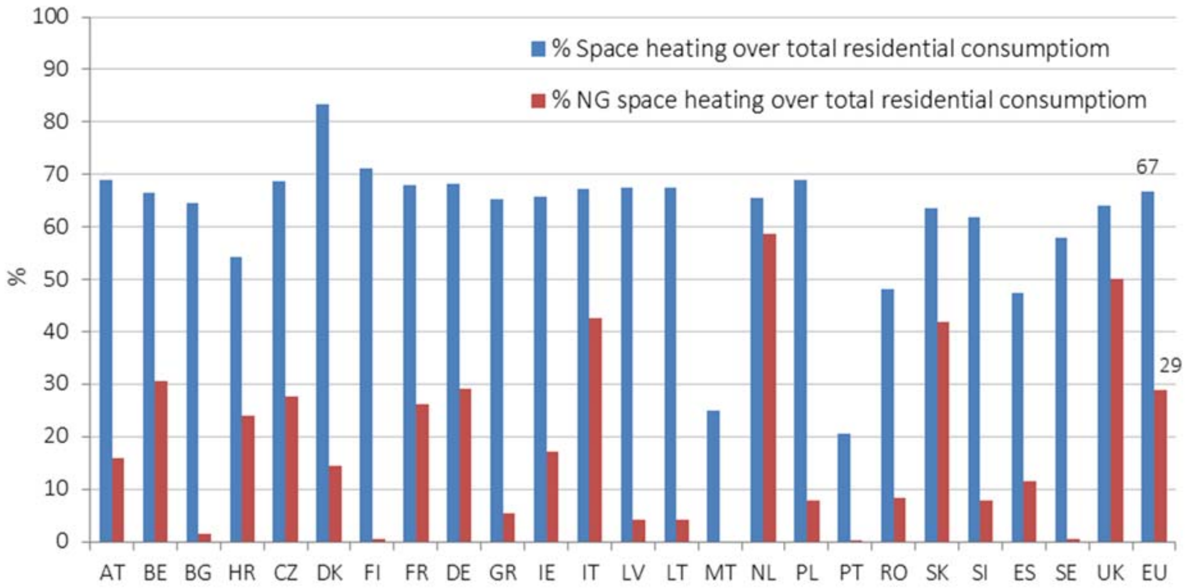


Figure 6.4: Space heating share in final energy consumption for residential sector by country in 2012 (based on Odyssee data).

Space heating is the most significant end use consumption in most of the European countries, with an average share of 67% of the total residential consumption in the EU. In 2012, NG space heating accounted for around 29% of the total residential sector consumption and 33% of the total final gas consumption in the EU. Four of the largest EU countries, namely United Kingdom, Germany, France and Italy, are responsible for around 70% of the total natural gas consumption used for space heating in the EU.

In Figure 6.5, it is possible to observe that the heating mix varies significantly from country to country over the EU. The dominant space heating source in United Kingdom, Netherlands, Italy, Germany, France, Slovakia, Belgium, Czech Republic and Croatia is NG. In Greece, Ireland, and Malta, oil based products dominate, whereas in Austria, Bulgaria, Latvia, Lithuania, Portugal, Finland, Romania and Slovenia wood is mostly used. In the remaining countries, Denmark and Sweden, most of the heating is supplied by district heating systems.

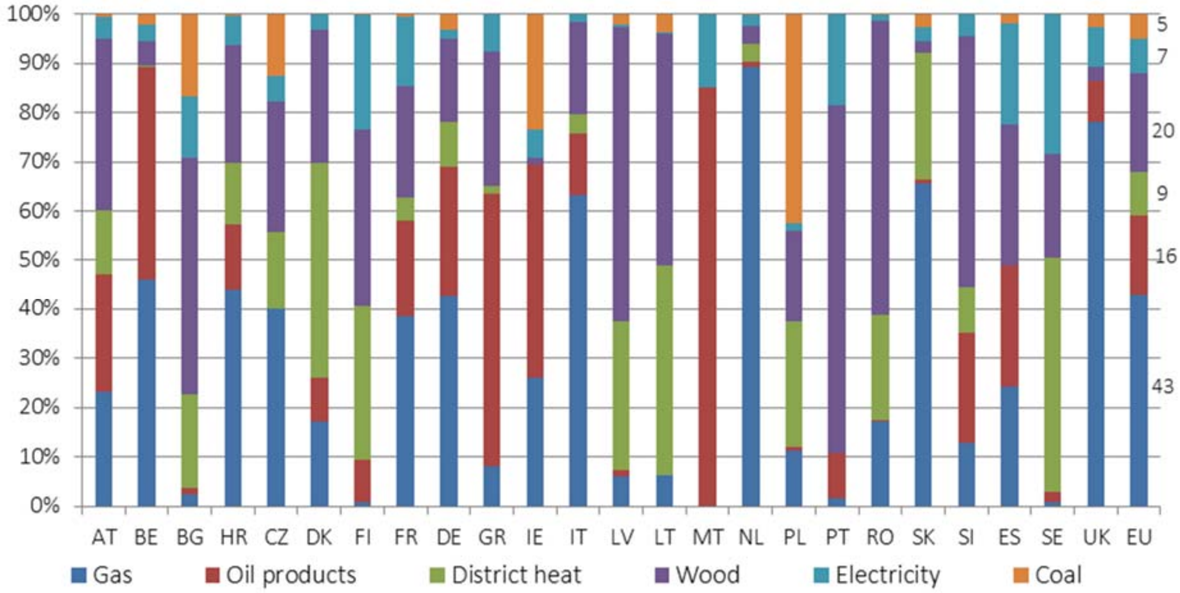


Figure 6.5: Space heating energy mix in the residential sector by EU country in 2012 (based on Odyssee data).

For the non-residential buildings, no detailed data for the EU space heating was found, but 8 countries are characterized in Odyssee data base (Odyssee, 2015). These 8 countries represent around 56.4% of the total population of the EU (Eurostat, 2015). The average space heating shares of these countries will be extrapolated to the EU, since it is considered that those countries are geographically representative of the existent space heating energy mix over the different European countries, as it can be observed in Figure 6.6.

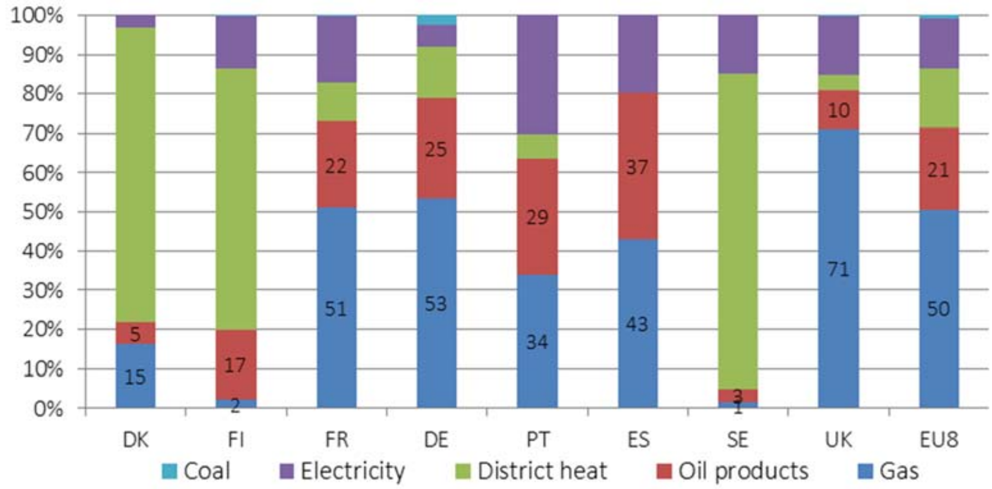


Figure 6.6: Space heating energy mix in non-residential buildings in 8 European countries in 2012 (based on Odyssee data).

The space heating in non-residential buildings in United Kingdom, Germany and France is dominated by natural gas, whereas in Denmark, Finland and Sweden is largely dominated by district heating. Finally, in Portugal and Spain, oil based products and NG dominate. In these 8 countries, in 2012, space heating accounted for around 43% of the total energy consumption in non-residential buildings. The NG space heating was responsible, on average, for 21% of the total energy consumption in non-residential buildings and for 13% of the total final NG consumption (Figure 6.7).

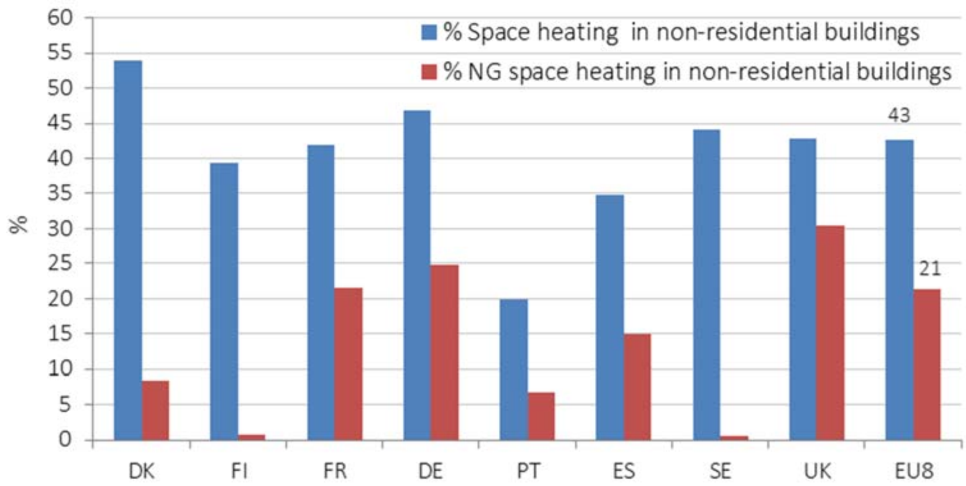


Figure 6.7: Space heating share in non-residential buildings total energy consumption in 8 European countries in 2012 (based on Odyssee data).

In 2012, space heating represented, on average, around 58% of the total consumption in the buildings sector (residential 67% and non-residential buildings 43%) and NG represented around 45% of the total space heating energy consumption in buildings in EU, as presented in Figure 6.8. The NG space heating has represented 29% (970 TWh) of the final energy consumption in residential buildings and 21% (380 TWh) of the total energy consumption in non-residential buildings.

According to the data from Eurostat energy balance (Eurostat, 2015), in 2012, the EU buildings sector has consumed a total final energy of around 5170 TWh. Applying the shares obtained previously, it was estimated that, in 2012, buildings space heating consumption was around 3015 TWh and NG space heating consumption was around 1350 TWh. This value will be used as the natural gas reference consumption for the primary energy and CO_2 emissions savings calculations in EU.

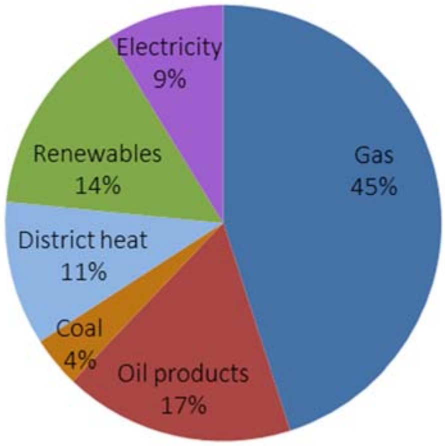


Figure 6.8: Buildings (residential and non-residential) space heating energy mix in EU in 2012 (based on Odyssee).

In conclusion, in 2012, buildings space heating accounts for around 23% of the total final energy consumption in EU, where two-thirds of the supplied heat is generated with fossil fuels (Figure 6.8), with the highest share corresponding to natural gas. This situation provides a huge opportunity for replacing fossil fuels by renewable energy resources coupled with the large scale application of high efficient heat pumps. This strategy is in line with studies that present renewable heating and cooling as a vital strategy to ensure Europe decarbonization (Connolly et al., 2012; Connolly et al., 2013; European Commission, 2011d; 2011e; 2011f).

6.3.2 - SPACE HEATING SCENARIO FOR 2030 AND 2050 IN EU

The reference space heating scenario that will be used in this study, to assess the CO_2 emissions and primary energy savings from switching natural gas boilers by HPs in EU buildings, is based on the two reference scenarios used in the two studies of the Heat Energy Roadmap 2050 (Connolly et al., 2012; Connolly et al., 2013). The reference scenarios used in these studies were adapted from two official scenarios defined and published by the European Commission in the Energy Roadmap 2050 report for the future of the EU energy system (European Commission, 2011c; 2011d; 2011f).

According to the Heat Roadmap second study (Connolly et al., 2013), the heat (space and water heating) demand in buildings will decrease 25% from 2015 to 2050 in the Current Policy Initiatives scenario (CPI scenario), which is a current trend scenario. On the other hand, water heat demand in buildings was forecasted to increase around 16% in the same period, even in an Energy Efficiency scenario (EE scenario).

The reference space heating scenario that will be used in this study results from the difference between the total heat demand scenario, including Hot Water (HW) demand (CPI heat scenario) (Connolly et al., 2013), and the hot water demand scenario (EE hot water scenario) (Connolly et al., 2013), corresponding to a total decrease of 35% in buildings space heating in the EU until 2050. This forecasted decrease is mainly due to the improvement in the buildings thermal insulation. Based in this new space heating scenario, and assuming that NG demand for space heating will decrease in the same proportion along the years, the reference NG space heating scenario was created, as represented in Figure 6.9. This scenario will be used to assess the primary energy and emissions savings in the EU. As a starting point, the NG consumption for space heating calculated in Section 6.3.1 was assumed for 2015.

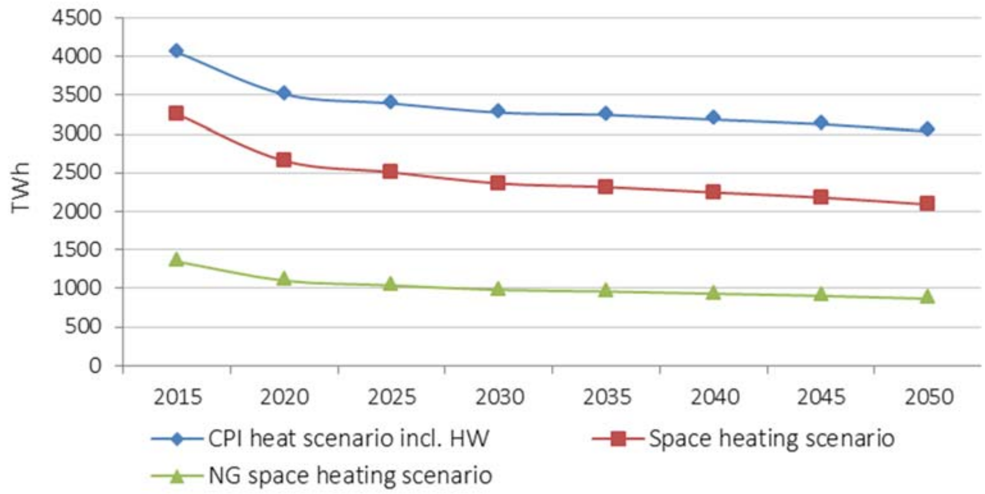


Figure 6.9: Buildings reference scenarios for heat demand (CPI heat scenario)(Connolly et al., 2013), forecasted total space heating and NG space heating in EU until 2050.

6.4 - RESULTS IN EU - PRIMARY ENERGY AND CO_2 EMISSIONS SAVINGS BY SWITCHING NG BOILERS TO HPS

In this study, an implementation scenario until 2050 will be used, where NG boilers will be gradually phased out and switched by electric HPs, to satisfy the same space heating demand corresponding to NG consumption scenario for buildings space heating that was defined in Section 6.3.2.

To assess the energy savings by switching NG boilers to GSHPs, the following conditions were considered:

- A reference NG space heating consumption as defined in Section 6.3.2 (Figure 6.9);
- The NG consumption for space heating will decrease linearly along the years till zero in 2050;
- The HPs and corresponding electric consumption will increase linearly in order to always compensate the space heating demand that would be supplied by NG boilers;
- A SPF of 4.5 for HPs was used, based on the Ground-Med Project results (Ground-Med, 2014) for all years up to 2050 (a conservative assumption);
- A constant gas boiler efficiency of 95% and the corresponding CO_2 emission factor based on CO_2 emissions of 202 g CO_2/kWh of NG (European Commission, 2007), were used for all years;
- Based on the predicted CO_2 emissions factor for electricity presented in (Figure 6.1), CO_2 emissions factors of around 200 g CO_2/kWh and 100 g CO_2/kWh were used for 2030 and 2050, respectively;
- A conversion factor of 2.5 for electricity primary energy calculations was used for 2015, based on the estimated 40% average EU generation efficiency, as established in Annex IV of Energy Efficiency Directive (2012/27/EU) (EU, 2012). For 2050, a conversion factor of 1.9 was estimated, based in the following assumption: to achieve a CO_2 emission factor of 100 g CO_2/kWh in 2050, only 20% of the electricity generation comes from fossil fuel with shares of 75% for GN (high efficiency combined cycle plants with an efficiency of around 60%) and 25% for coal. This combination will lead to an average efficiency of 53% for electricity generation from fossil fuels;
- A constant conversion factor of 1 for natural gas primary energy calculations was used for all years.

In Figure 6.10, the NG space heating consumption scenario, the decreasing NG consumption in space heating related to the replacement of NG gas boilers by HPs (NG replaced by HPs), the primary energy consumption associated to the increasing introduction of the HPs (HPs primary energy) and, finally, the decreasing of the primary consumption until 2050 associated to the replacement of the NG boilers by HPs (future Space Heating (SH) primary energy) are presented till 2050.

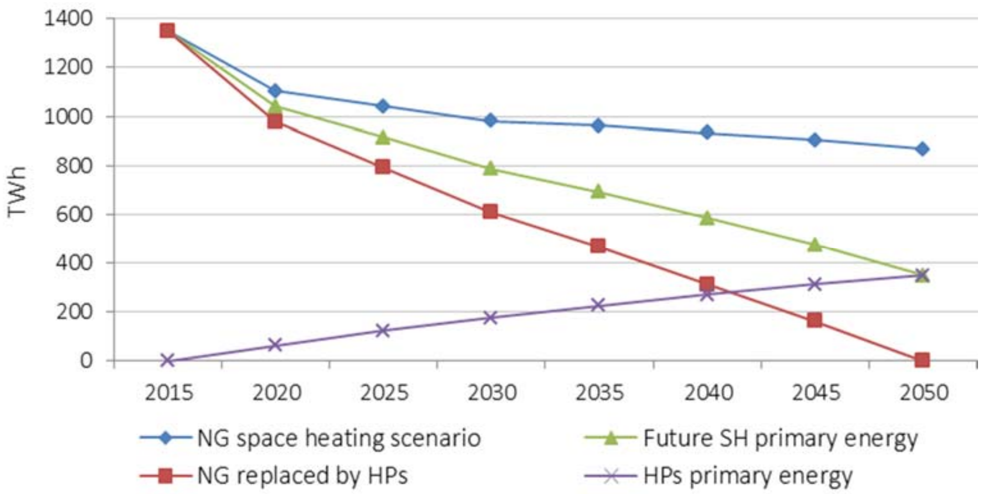


Figure 6.10: Heat pumps implementation scenario and energy impact of switching NG boilers by HPs to satisfy the same predicted space heating demand in EU.

According to the implementation scenario of switching NG boilers by HP in EU, as presented in Figure 6.10, the results in terms of primary energy for NG space heating are as follows:

- In 2030 – 38% of the NG space heating consumption will be replaced by HPs, which leads to a primary energy reduction of 196 TWh, corresponding to savings of 20%, in relation to the NG consumption that would be used without the HPs;
- In 2050 – 100% of the NG space heating consumption will be replaced by HPs, which leads to a primary energy reduction of 520 TWh, corresponding to savings of 60%, in relation to the NG consumption that would be used without the HPs.

In Table 6.1, the impact of the primary energy savings achieved in 2030 (196 TWh) and in 2050 (520 TWh), at global levels, are presented:

Table 6.1: Impacts in primary energy consumption of the replacement NG boilers by HPs.

Primary energy consumption	**2030**		**2050**	
	Energy [TWh]	**Savings (%)**	**Energy [TWh]**	**Savings (%)**
NG Space heating scenario (Figure 6.9)	980	20	870	60
Space heating scenario (Figure 6.9)	2 370	8	2 100	25
NG primary energy (CPI scenario) [1]	4 520	4	4 360	12
Total primary energy (CPI scenario) [1]	18 710	1	18 230	3

[1] Primary energy for the CPI scenario in the Heat Roadmap 2050 - 2nd pre-study (Connolly et al., 2013)

As presented in Figure 6.11, the corresponding emissions savings due to switching NG boilers by electric HPs in EU are as follows:

- In 2030 – 59 Mt CO_2, corresponding to a reduction of 30% of the expected emissions for the NG space heating. This value also corresponds to a reduction of 1.9 % in the total CO_2 emissions in EU presented in the CPI scenario (Connolly et al., 2013);
- In 2050 – 157 Mt CO_2, corresponding to a reduction of 90% of the expected emissions for the NG space heating. This value represents a reduction of 6% in the total CO_2 emissions in EU presented in the CPI scenario (Connolly et al., 2013). This large CO_2 emissions savings is, in a substantial part, due to the expectation that carbon intensity of electricity in 2050 will be only about 100 g CO_2/kWh, as presented in Section 6.2 (Figure 6.1).

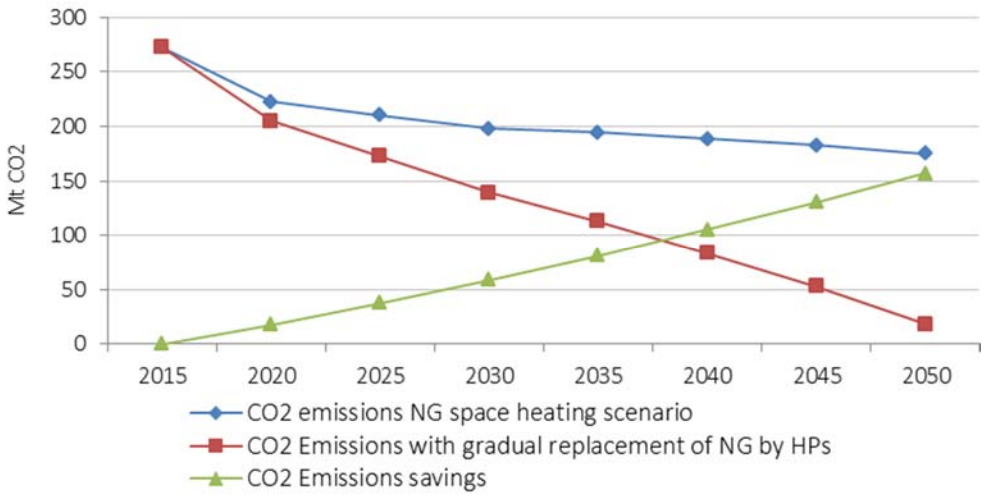


Figure 6.11: Emissions reduction in the NG space heating share, by switching NG boilers to HPs in EU.

The proposed fuel switching strategy (replacement of NG boilers by HPs) has a very positive impact towards the decarbonization of the buildings space heating at EU level. It can also help to achieve the EU reduction emissions target of reducing emissions by 80% in 2050, representing a contribution of around 3.5% of the global ambitious target.

The results obtained show larger percentage emissions and primary energy savings than other studies due to considering higher efficiency HPs and the lower carbon emissions expected by 2050. The savings projected for 2030 in the Ecofys analysis (Ecofys, 2013a) are higher than the obtained in this study, that only covers NG space heating, leading to absolute CO_2 equivalent emissions of 342 Mt and primary energy savings of 1260 TWh, which make senses, since total space heating, water heating and cooling were considered by Ecofys.

Additionally, the impact on the NG dependency and the contribution to increase the RES share in the total final energy consumption were calculated for 2030 and 2050. Assuming that the electricity consumed by HPs will not be generated by NG, the NG space heating fuel switching strategy in buildings per se can reduce NG dependency to 62% in 2030 and to 50% in 2050, as shown in Table 6.2.

Table 6.2: Impacts in NG dependency by switching NG boilers by HPs.

Primary energy consumption [TWh]	**2030**	**2050**
NG primary energy (CPI scenario)	4520	4360
NG imported (70% over total)	3160	3050
NG Space heating scenario (Figure 6.9)	980	870
NG Space heating with HP increase	610	0
NG replaced by HP	370	870
Future NG imported	2790	2180
Future NG dependency (%)	62	50

The EU RES Directive (EU, 2009b) imposes that the RES contribution of HPs is only taken into account if the SPF is greater than 1.15 × 1/η, where η represents the electricity generation efficiency. According to guidelines for calculating renewable energy from heat pumps (European Coommission, 2013c), an electricity generation efficiency of 45.5% must be used, leading to a minimum SPF of 2.5, which is the case by a large margin. Applying the methodology defined in the European Commission Decision 2013/114/EU (European Coommission, 2013c), the supplied renewable energy (ERES) from the installed HP is calculated by:

$$E_{RES} = Q_{usable} \times \left(1 - 1/SPF\right) \tag{6.1}$$

where the Q_{usable} is the total usable heat delivered by heat pumps which corresponds to the thermal energy they supply. The results are shown in Table 6.3.

Another important impact of the proposed strategy is its contribution to the increase of RES share in final energy consumption, which is one of the three targets of the EU energy policy. Heat pumps have two different renewable energy contributions, one from the heat source used (ground, water or air) and another one from the electricity, increasingly produced by renewable energies (expected RES share in electricity production of 80% by 2050). Considering these two effects, a RES share of 5.6% in final energy consumption can be achieved by 2050.

Table 6.3: RES contribution in total final energy and space heating consumption from HPs that replaced the NG boilers.

Energy [TWh]	2030	2050
Thermal energy supplied by HPs (Q_{usable})	355	825
Renewable energy supplied by HPs (E_{res})	275	640
Renewable energy supplied by HPs (%)	77.8	77.8
Renewable electricity consumed by HPs	35	148
Total final energy (CPI scenario)	13 100	14 600
RES share from HPs in total final energy (%)	2.4	5.6

6.5 - GSHP CARBON EMISSIONS AND PRIMARY ENERGY SAVINGS FOR BUILDINGS HEATING IN PORTUGAL

To assess the global impacts of switching boilers by HPs for space heating, as made for the EU case, the space heating characterization in Portugal is also presented. The efficiency results obtained in the test GSHP system are used as base for the Portuguese case.

6.5.1 - SPACE HEATING CHARACTERIZATION IN PORTUGAL

To estimate the buildings space heating consumption in Portugal, the same methodology previously applied for the whole EU was used. The space heating energy shares were calculated from the Odyssee data for 2012.

According to Eurostat, in 2012, the buildings sector (residential and non-residential buildings) consumed around 28% (53 TWh) of the total final energy consumed (total 190 TWh) in Portugal. The total final energy consumption, buildings final energy consumption and buildings space heating consumption by energy type in Portugal for 2012 are represented in Figure 6.12.

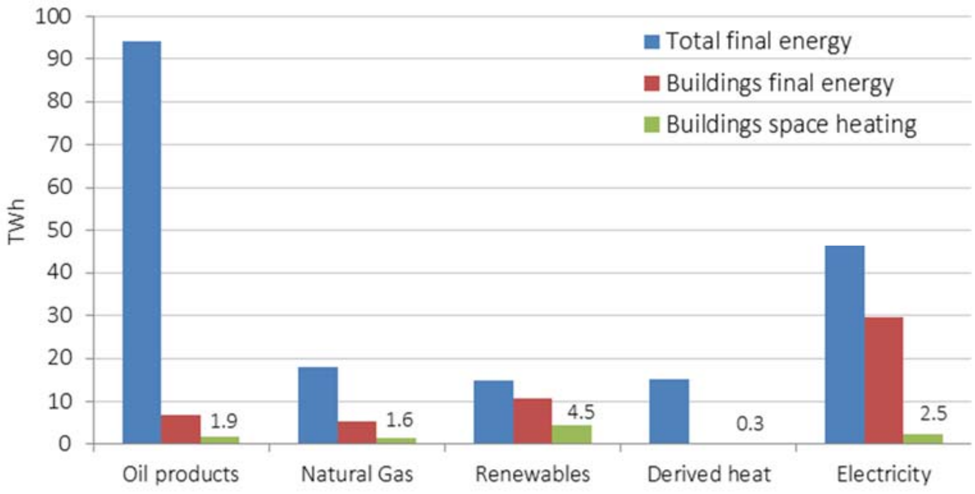


Figure 6.12: Final energy consumption by energy type in Portugal in 2012 (based on Eurostat and Odyssee data).

Electricity was the dominant energy consumed in the building sector (56%). The remaining energy sources had the following contributions: RES (20%), oil products (13%) and NG (10%). In the residential sector, the main loads are cooking appliances (40%), followed by space heating and water heating with similar shares, as shown in Figure 6.13.

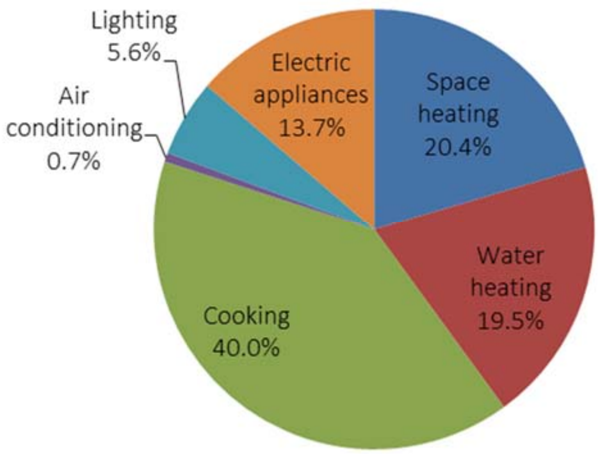


Figure 6.13: Energy share by end-use in the residential sector in Portugal in 2012 (based on Odyssee data).

In average, space heating represents 20% (10.7 TWh) of the total buildings consumption. This share was almost the same for residential and non-residential buildings. Portugal is the country with the lowest space heating share in EU (Figure 6.4 and Figure 6.7), which can be justified by the moderate climatic conditions, as well as by the lack of thermal comfort in many buildings. From the 10.7 TWh consumed for buildings space heating, 4.5 TWh are RES (mainly biomass in the residential sector), 2.5 TWh are electricity, 1.9 TWh are oil products and 1.6 TWh are from NG.

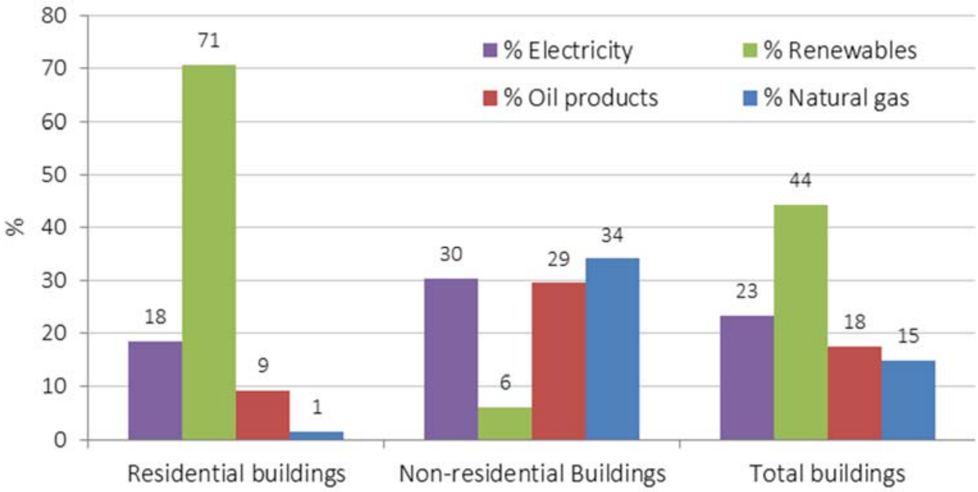


Figure 6.14: Space heating shares by energy type in buildings in Portugal in 2012 (based on Odyssee data).

Natural gas represents the lowest space heating share (15%) of the total buildings, as shown in Figure 6.14, contrary to the EU case, where NG has the highest contribution (58%). This is in part due to the limited availability of the NG network in the coldest parts of the country. Analyzing the amount of energy consumption for space heating, it can be concluded that a fuel switching strategy by HPs, even considering both fossil fuels (NG and oil products), will have a modest global impact in Portugal.

6.5.2 - RESULTS IN PORTUGAL - PRIMARY ENERGY AND CO_2 EMISSIONS SAVINGS BY HPS

For the Portuguese case study, the oil products, which have a larger contribution to space heating in buildings (23%) than NG, are also considered in this fuel switching strategy analysis. To assess the energy and emissions saving in the Portuguese case, some of the same considerations of the EU study were assumed and it was also considered a scenario in which primary energy will remain almost constant along the years. The main differences are:

- A scenario where space heating consumption will not decrease along the years was assumed, because of projected increased comfort levels;

- A SPF2 of 5.1, obtained in the experimental GSHP installation, was used (Section 4.1);
- An efficiency of 85% was assumed for all oil based heating appliances;
- The electricity CO_2 emission factor for Portugal was estimated based on the increasing of the RES share in electricity generation. In 2020, the RES target is 60% in Portugal. For 2030 and for 2050 it was assumed that RES shares of 70% and 90% are planned. The remaining thermal electricity generation is now produced by coal and increasingly by combined cycle NG power plants with efficiencies close to 60%, with a planned phase out of coal by 2030. The estimated electricity CO_2 emission factor for Portugal is around 40 g CO_2/kWh for 2050, which is less than half of European average;
- A conversion factor of 2.5 for electricity primary energy calculations was used for 2015, as in the EU case study (EU, 2012). For 2030 and 2050 a conversion factor of 1.7 was used, according to the assumptions of the previous point (the only thermal power plants will be combined cycle NG with 60% of efficiency);
- The oil products for space heating were disaggregated in Liquefied Petroleum Gases (LPG), diesel oil and heavy fuel oil, since the corresponding emission factors are different. To obtain the corresponding shares, the total consumption by each oil product for residential and non-residential buildings was used (Eurostat, 2015). Diesel oil and heavy fuel oil are only used for water and space heating, whereas LPG is also used for cooking. It was estimated that in building space heating, 55% is LPG, 39% is diesel and 6% is heavy fuel oil. The corresponding emissions factors are 227 g CO_2/kWh, 266 g CO_2/kWh and 278 g CO_2/kWh, as defined in the Commission Decision 2007/589/EC (European Commission, 2007).

In Figure 6.15, the scenario evolution till 2050 for NG and oil consumption in space heating, being progressively replaced by HPs, is presented.

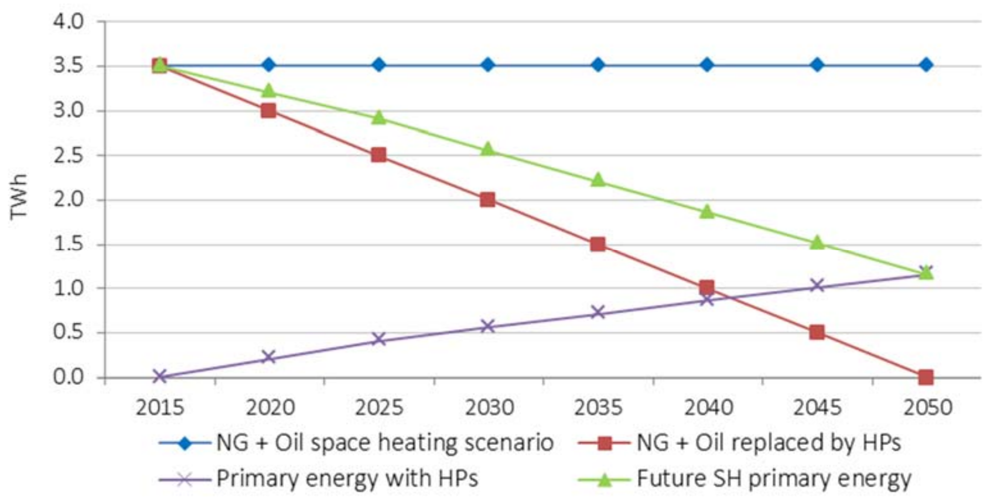


Figure 6.15: Implementation scenario and primary energy impact of switching NG and oil boilers by HPs to satisfy the same NG and oil space heating demand in Portugal.

According to the implementation scenario of switching NG and oil boilers by HPs in Portugal, as presented in Figure 6.15, the results in terms of primary energy for NG and oil space heating are as follows:

- In 2030 – 57% of the NG and oil space heating consumption will be replaced by HPs, leading to a primary energy reduction of 0.9 TWh (0.4 TWh of NG + 0.5 TWh of oil products), which corresponds to savings of 27% of primary energy, in relation to the NG and oil products consumption that would be used without the HPs;
- In 2050 – 100% of the NG and oil space heating consumption will be replaced by HPs leading to a primary energy reduction of 2.3 TWh (1 TWh of NG + 1.3 TWh of oil products), which corresponds to savings of 67% of primary energy, in relation to the NG and oil products consumption that would be used without the HPs.

In Table 6.4, the impacts of the primary energy savings achieved in 2030 and in 2050 at global levels, are presented. The NG and oil products dependency will decrease in Portugal 3.5% and 1.6%, respectively. These modest results are mainly due to the reasons presented before for the Portuguese case, mild winters and lack of thermal comfort in many buildings, leading to low energy consumption for space heating.

Table 6.4: Impacts in primary energy consumption in the case of the fossil fuel boilers replacement by HP in Portugal.

Primary energy consumption (TWh)	**2030**						**2050**					
	Energy [TWh]			**Savings (%)**			**Energy [TWh]**			**Savings (%)**		
	NG	**Oil**	**Total**	**NG**	**Oil**	**Total**	**NG**	**Oil**	**Total**	**NG**	**Oil**	**Total**
NG + Oil space heating scenario	1.6	1.9	3.5	26	28	27	1.6	1.9	3.5	65	69	67
Total space heating scenario			10.7	3.9	4.9	8.8			10.7	9.7	12	22
NG + Oil primary energy	46	117	162	0.9	0.5	0.6	46	117	162	2.3	1.1	1.4
Total primary energy			258	0.2	0.2	0.4			258	0.4	0.5	0.9
NG + Oil savings in space heating	0.4	0.5	0.9	-	-	-	1	1.3	2.3	-	-	-

As presented in Figure 6.16, the corresponding emissions savings of switching NG and oil boilers by electric HPs in EU are 309 kt CO_2 and 765 kt CO_2 in 2030 and 2050, respectively, leading to emissions reductions of 39% and 97% of the equivalent emissions using fossil fuels space heating.

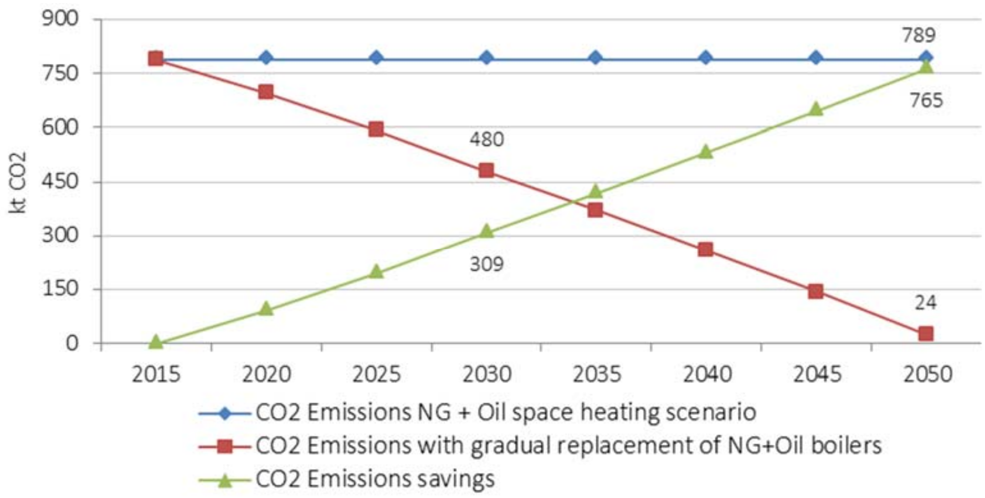


Figure 6.16: Emissions impact of switching fossil fuel boilers to electric HPs to satisfy the same NG and oil space heating demand in Portugal.

Comparing HPs with each fossil fuel technology for space heating and considering the electricity emissions factor of 40 g CO_2/kWh for 2050, it can be estimated that there are savings of 96% compared to NG boilers and 98% when compared with oil boilers.

The analysis carried out for Portugal demonstrates that HPs using electricity generated with mostly renewable sources can bring significant benefits when compared to NG and oil boilers, and seem an appropriate choice for new space heating systems.

6.6 - CONCLUSIONS

In 2012, buildings space heating accounted for 23% of the final energy consumption in EU, with a dominant contribution from fossil fuels, in which imported natural gas has the highest share. With the increasing decarbonization of the electricity generation mix, buildings space heating provides a huge opportunity for replacing fossil fuels by renewable energy sources coupled with the large scale application of high efficient heat pumps.

The analysis carried out is focused in the replacement of the NG space heating boilers by state of the art high efficiency HPs all over Europe in the period up to 2050. The proposed fuel switching strategy can save in European Union around 60% of the primary energy (520 TWh) and 90% of CO_2 emissions (157 Mt CO_2) for space heating in 2050, when compared with NG boilers. This strategy can also contribute to achieve the EU target of reducing emissions by 80% in 2050, representing a contribution of around 3.5% of that global ambitious target, as well as to decrease the European NG dependency on external suppliers to 50%. Another important impact of the proposed strategy is its contribution to increase of RES share, representing 5.6% in final energy consumption by 2050.

In the particular case of Portugal, with a very ambitious RES program and with the high SPF obtained experimentally, 67% of the primary energy consumed by existent NG and oil boilers and 97% of the corresponding CO_2 emissions can be saved in 2050.

The analysis of previous related studies showed that high efficient heat pumps have, in most cases, very positive impacts in the emissions and primary energy savings. The results can vary substantially depending on the considered key factors in the analysis, such as: HPs efficiency, the CO_2 emissions factors, the heating and electricity generation mix and the thermal loads that will be replaced by heat pumps. The results of this study show larger percentage emissions and primary energy savings than other studies due to considering higher efficiency HPs and the lower carbon emissions expected by 2050.

The current policy framework in place in the EU with relevance for the high efficiency HP technology implementation, includes the RES Directive (EU, 2009b) (which officially recognizes that HPs use renewable energy sources) and the Energy Performance of Buildings Directive (EU, 2010b) (which imposes to assess the techno-economic impact of alternative high efficient heating technologies for new buildings, like heat pumps). The Ecodesign Directive (EU, 2009a) is also promoting innovation in design and marketing of high efficiency HPs, leading to increasingly ambitious minimum energy performance standards. Financial and tax incentives are already being used in some countries and deserve to be expanded, since they can play a major role to speed up the large scale implementation of high efficiency HPs, particularly to retrofit existing buildings.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

7.1 - CONCLUSIONS

In this thesis it was proposed to assess the cross benefits of using Ground Source Heat Pump (GSHP) systems for space conditioning in buildings, in an integrated perspective, in terms of potential impacts such as increasing energy efficiency, increasing the use of Renewable Energy Sources (RES), reducing carbon emissions, as well as their use as a flexible load with thermal storage for balancing demand and supply, namely for the integration of intermittent renewable energies. Part of these impacts were assessed based on the experimental results obtained in the GSHP system installed in a public service building, under the European Project Ground-Med, from the Seventh Research Framework Programme - FP7.

Higher efficiency solutions with less carbon contents for space conditioning need to be implemented, namely for space heating, since space heating is the end-use with the highest share of the final energy consumption in buildings at the EU level. In the particular case of Portugal, due to the moderate climate conditions and due to the lack of thermal comfort in many buildings, space heating is not so significant, but together with space cooling, they already represent one of the most significant end-uses energy consumption in buildings. GSHP technologies are widely used in Northern and Central Europe but the corresponding market in Southern Europe is at early developing stages, although there is the additional advantage that they can be reversible and used for both space heating and space cooling. In the current context, in which EU legislation imposes the improvement energy efficiency in buildings, R&D and demonstration studies based on high performance field installations are very helpful to promote advanced technologies in existent and new buildings.

The efficiency of an advanced ground source heat pump system, which integrates state-of-the-art components, was assessed, considering the Seasonal Performance Factors (SPFs) for the heating and cooling seasons, which corresponds to the average system performance for the entire season. Four different SPFs were calculated for each season based on the monitored data of the experimental GSHP system installed in the service building in Coimbra. The results of the pilot installation demonstrated the feasibility of high efficiency GSHPs, with an obtained SPF of 5.4 for the heating season and 6.4 for the cooling season. Even considering the overall GSHP system performance, which includes the electric consumption of the building thermal energy distribution system, a SPF of 4.3 for heating and of 3.9 for cooling was obtained.

A control system was developed using the state-of-the-art measuring technology (integrated systems with real-time controller) that allows the installation of I/O modules to control external relays that command the start/stop of the equipment such as the Heat Pump (HP), the circulation pumps and the fancoil units. The advanced control system had as main purposes: to allow programing the daily operation period of the overall system, in order to guarantee temperature comfort when users arrive and to implement load shifting strategies, and to allow a daily parameterization in a remote way of the HP temperature operation and circulation pumps velocities, in order to optimize the system performance.

By the tests performed during summer and winter, it was concluded that the control strategies of adjusting the set point temperature of the GSHP and the circulation pumps velocities are very important to improve the system efficiency. The decreasing or increasing of 1 °C in the water supply temperature, depending on the season (heating or cooling), improved in 2.5% the HP efficiency.

It was also concluded that the circulation pumps velocities have a higher influence in the overall system performance during days with lower thermal needs, since less thermal energy is delivered to the building, but the electricity consumption of the circulation pumps remains almost the same, especially in circuits with no load variation (ground loop and internal loop between HP and the decoupling tank). In order to reduce energy consumption of auxiliary equipments, the external circulation pump operation was linked to the HP compressors operation. This strategy cannot be applied in the building internal loops since water should be always in circulation to feed fancoil units and minimum flow rate must be guaranteed for that HP works without damage risks, being more critical at the evaporator side.

For each season, the best combination of the circulation pumps velocities was determined (at the condenser side and the evaporator side), to maximize the GSHP system performance. Analyzing the daily performance factors, it was possible to verify that a higher circulation pump velocity (higher flow rate) increases the HP performance, but the corresponding increasing of the circulation pumps energy consumption penalizes the overall GSHP system performance.

The combination of the circulation pumps velocities that optimizes the system performance in the cooling mode was investigated. In the heating season fewer combinations were possible to test, since the HP evaporator is in the ground loop side, which has a higher pressure drop, and the minimum flow rate supported by the HP corresponds to 80% of the maximum head position of the external circulation pump. Applying these optimal velocities to the circulation pumps, the overall performance factor has improved 8% in summer and 2% in winter, when compared with the maximum velocity pump settings tested.

The building thermal response was analyzed in order to be possible to implement space heating load shifting strategies without creating major disturbances in the users thermal comfort. The necessary preheating period to guarantee that at a desired time the building temperature is already stabilized, has been determined using a model based on the lumped capacitance method, in which the building is considered as an equivalent homogeneous body. In this model, the inside building temperature corresponds to the temperature of the building thermal mass as a whole and this temperature is considered spatially uniform at any instant during the transient process. This model has several limitations, since it cannot predict specifically the temperature of any building component (indoor air temperature, walls temperature, etc.) as the building temperature decay is represented by a single exponential decay, corresponding to a single time constant. This simple method can be easily implemented in other buildings and it can be used as a first approach to define the best start operation time based on the indoor temperature decrease of the building. The key parameters of the building (building heat loss coefficient and building thermal capacity), which are needed to apply this model, were determined based on the monitored data of the outdoor temperature, indoor temperature and thermal power consumption. A good agreement between the model and the experimental data of the building temperature evolution was found.

Besides the analyses of the building thermal mass temperature decay, the indoor air temperature decay after turning off the GSHP system was also analyzed, in order to verify during how much time load curtailment strategies could be applied in this building. It was verified that, even for an outdoor temperature of around 10 °C at 6 p.m., the indoor air temperature decreased 1 °C after 1 hour and 1.5 °C after 1.5 hours. These results show that the building has a good thermal inertia, which delays the indoor air temperature decrease. It was assumed that this temperature decrease is acceptable for users and that the curtailment strategy can be used for periods up to 1.5 hours. Based on these results, it was concluded that the curtailment strategy can be used to avoid electricity consumption during peak electricity prices and to shut down the GSHP system earlier than the ending of the occupation period.

The application of load shifting strategies for space heating, such as preheating and curtailment strategies, were analyzed and assessed in terms of the savings costs at the consumer side and of the impacts on the national electric diagram. Only space heating was analyzed since most of the hydro and wind electricity is generated during winter and in this particular building the heating demand is 70% higher than the cooling demand. The main purpose of these strategies is to profit from the off peak electricity prices (from 2 a.m. to 6 a.m. – 0.0747 €/kWh and from 6 a.m. to 8 a.m. – 0.0849 €/kWh) and to avoid peak electricity prices, since they are much more expensive (0.2097 €/kWh) and has an additional penalty applied to the monthly average power demand used during the peak period, increasing the total cost during peak hours to 0.3298 €/kWh. Another objective was to take advantage of the use of the existent thermal storage capacity in buildings with medium and high thermal inertia. Although active thermal energy storage systems can increase the space conditioning flexibility, the existent thermal capacity of the building can also be used, without significant investments.

To guarantee thermal comfort to the users when they arrive and to take advantage from the lower electricity prices, two different preheating strategies were applied. The main purpose was that at 9 a.m. the building temperature is near to the steady state comfort temperature to minimize or even to avoid the energy consumption during the morning peak period. Different preheating times were tested associated to a curtailment strategy at the end of the day to avoid the second electricity peak period costs. The results showed that, applying these preheating strategies associated to the shutting off of the heating system one hour before the end of the working day, savings between 16 to 19% can be achieved depending on the necessary preheating time. Higher costs savings can be achieved (about 34%) with longer preheating periods and higher indoor temperature set point, in order to be possible to avoid energy consumption during the morning peak (shutting off the system between 9 a.m. and 10:30 a.m.).

The assessment of potential flexible loads to be integrated in smart grids, as a way to balance supply and demand, is also very important, namely in a fast increasing intermittent renewable electricity context. Nowadays, wind power already covers a large part of the demand for electricity in many areas of Europe and in the particular case of Portugal it represents 26% of the total installed power, ensuring 24% of the electricity generation in 2014.

The impact of preheating the office buildings as a way to integrate the renewable generation surplus was studied, as well as the use of office buildings space heating as a flexible load to reduce the mismatch between the forecasted and real wind power generation. Based in the results achieved in this particular building, the load shifting strategies were extrapolated to other office buildings, considering the same occupancy periods and assuming that 25% of the total office buildings space heating load in Portugal can be controlled in a centralized way. Office buildings represent about 10% of the total electricity consumed at national level and space heating represents around 19% of the total offices electricity consumption.

Based on the two examples studied, starting the building preheating at 6:30 a.m. in less cold days and at 4:30 a.m. in colder days, it was found that significant part of the renewable energy generated during night and early morning (19-30%) can be absorbed. Due to the increase of demand during valley periods, part of the surplus of renewable generation would be avoided, therefore reducing the problems in the grid management and simultaneously minimizing the negative economic impacts on the energy market. Additionally, in this situation, a substantial part of the heating electricity consumption is supplied only with renewable generation, therefore reducing greenhouse gases emissions.

The impact of shutting down the space heating systems of 25% of the office buildings during one hour was assessed using a day where large ramp rates of wind generation reduction and a large forecasting error occurred. It was also demonstrated that by shutting down temporarily the space heating it is possible to substantially compensate the variations of wind power generation and forecasting errors during the first hour of variation (55.5% in the presented scenario), giving enough time to adapt the other resources of the electrical system.

Finally, the impacts of the large scale penetration of high efficiency heat pumps, in a scenario where Natural Gas (NG) space heating will be progressively phased out in EU and in Portugal until 2050, were assessed in the following aspects: decrease in the carbon emissions, decrease in the primary energy consumption, decrease of NG dependency and the increase of the RES share contribution.

In 2012, buildings space heating accounted for 23% of the final energy consumption in EU, with a dominant contribution from fossil fuels in which imported natural gas has the highest share, providing a huge opportunity for replacing fossil fuels by renewable energy sources coupled with a large scale application of high efficient heat pumps.

The analysis carried out was focused in the replacement of the NG space heating boilers by state of the art high efficiency HPs all over Europe in the period up to 2050. A reference space heating and natural gas space heating scenario was created based on European studies for 2050. It was assumed that the NG consumption for space heating will decrease linearly along the years till zero in 2050 and that HPs and the corresponding electric consumption will increase linearly in order to always compensate the space heating demand that would be supplied by NG boilers.

The proposed fuel switching strategy can save in European Union around 60% of the primary energy (520 TWh) and 90% of CO_2 emissions (157 Mt CO_2) for space heating in 2050, when compared with NG boilers. This strategy can also contribute to achieve the EU target of reducing emissions by 80% in 2050, representing a contribution of around 3.5% of that global ambitious objective, as well as to decrease the European NG dependency on external suppliers to 50%. Another important impact of the proposed strategy is its contribution to increase the RES share, representing 5.6% of the final energy consumption by 2050.

A similar scenario was developed for Portugal and the same impacts were also assessed, considering the full replacement of the oil products and NG until 2050. In the particular case of Portugal, with a very ambitious RES program, 67% of the primary energy consumed by existent NG and oil boilers and 97% of the corresponding CO_2 emissions can be saved in 2050. The results of this study showed larger percentage emissions and primary energy savings than other studies due to considering higher efficiency HPs and the lower carbon emissions expected by 2050.

In summary, high efficient heat pumps can have an important role towards exploiting the increasing penetration of renewable electricity generation, effectively contributing to the replacement of fossil fuels and to decrease the growing natural gas dependency of Europe from risky and unstable countries. Additionally, high efficient heat pumps, when combined with the building thermal storage capacity (available in many buildings), can be used as a flexible load to balance supply and demand, applying load leveling and Demand Response strategies, and to reduce significantly operation costs at the consumer side.

7.2 - RECOMMENDATIONS FOR FUTURE WORK

In this thesis, the optimization of the GSHP system in terms of its performance, including auxiliary equipment like circulation pumps, and in terms of period operation to guarantee users thermal comfort and minimize operation costs, was proposed. Of course, this optimization work is not concluded and, at this phase, it is important firstly to assess the use of a thermal storage tank with PCMs and optimize the control of the GSHP, circulating pumps and fancoils.

Concerning the thermal storage tank with PCMs, the experimental installation is already prepared with the necessary devices to its implementation and some research was also done about this technology. The most suitable PCM technology to be used is already selected based in the main criteria factors. As there is a lack of demonstration studies with the application of PCMs in storage tanks for space heating in daily cycles, the PCMs tank evaluation is important in terms of response time (charge and discharge), losses, impacts on the system efficiency and stability of the heat storage capacity.

It is also important to study different tanks configurations, namely vertical tanks with other thermal energy storage technologies, in order to overcome space restrictions, which was the main reason that prevented the tank installation (nearly half of the room would be occupied with the horizontal tank required by salt hydrates modules).

Another important research line is the control mode of the HP and it would be interesting to study other control modes, besides the water temperature return mode, such as temperature compensation based on outdoor temperature or in buffer tank water temperature (this requires the manufacturer collaboration since HP control is a closed system). This will allow setting a range for the HP set point variation according to the thermal needs, which will improve performance factors.

As the energy consumption of the circulation pumps is not negligible, another control improvement is the adjustment of the circulation pumps velocity to the thermal needs requirements, increasing the systems efficiency. A range of velocities variation can be defined between the minimum velocity, which corresponds to the minimum flow rate affordable by the HP, and the maximum velocity, which will also improve performance factors.

To simplify the implementation of Demand Response strategies it would be very important to establish communication between the centralized control system and the fancoil units, in order to allow the change of the room temperature set point. This feature will facilitate the implementation of load shifting strategies, increasing and decreasing set point depending on the electricity prices and on the grid signals.

Finally, to optimize the daily operation period control of the GSHP system, it would be necessary to implement other control strategies such as an auto-adapt control to define the optimum start time operation of the system taking into account the real time temperatures (outdoor and indoor temperatures), the electricity prices and the users thermal comfort. The preheating times achieved by the thermal response model presented, or by a more complex and detailed model, could be used as a start point. The preheating times could be updated if the initial ones do not guarantee the desired conditions.

References

Agyenim, F., Hewitt, N., Eames, P. and Smyth, M. (2010). A review of materials, heat transfer and phase change problem formulation for latent heat thermal energy storage systems (LHTESS). *Renewable and Sustainable Energy Reviews*, 14, 615-628.

Ali, M., Jokisalo, J., Siren, K. and Lehtonen, M. (2014). Combining the Demand Response of direct electric space heating and partial thermal storage using LP optimization. *Electric Power Systems Research,* 106, 160-167. doi:10.1016/j.epsr.2013.08.017

Arteconi, A., Hewitt, N.J., and Polonara, F. (2013). Domestic demand-side management (DSM): Role of heat pumps and thermal energy storage (TES) systems. *Applied Thermal Engineering*, 51, 155-165. doi:10.1016/j.applthermaleng.2012.09.023

Balijepalli, V.S.K.M., Pradhan, V., Khaparde, S.A. and Shereef, R.M. (2011). Review of demand response under smart grid paradigm. In: *2011 IEEE PES International Conference on Innovative Smart Grid Technologies – India (ISGT India 2011)*, 1-3 December 2011, Kollam, Kerala, India. Piscataway, NJ: IEEE, pp. 236-243. doi: 10.1109/ISET-India.2011.6145388

Ban, M., Krajačić, G., Grozdek, M., Ćurko, T. and Duić, N. (2012). The role of cool thermal energy storage (CTES) in the integration of renewable energy sources (RES) and peak load reduction. *Energy*, 48, 108-117. doi:10.1016/j.energy.2012.06.070.

Barba, A. and Spiga, M. (2003). Discharge mode for encapsulated PCMs in storage tanks. *Solar Energy*, 74, 141-148.

BASF (2015). *BASF Micronal PCM – Intelligent Temperature Management for Buildings.* [Online]. Ludwigshafen, Germany: BASF SE, Business Management Micronal PCM. Available from: http://www.micronal.de/portal/load/fid443847/BASF_Micronal_PCM_Brochure%202009_English.pdf [Accessed April 2015].

Bayer, P., Saner, D., Bolay, S., Rybach, L. and Blum, P. (2012). Greenhouse gas emission savings of ground source heat pump systems in Europe: A review. *Renewable and Sustainable Energy Reviews*, 16, 1256-1267. doi:10.1016/j.rser.2011.09.027

Blum, P., Campillo, G., Münch, W. and Kölbel, T. (2010). CO_2 savings of ground source heat pump systems – A regional analysis. *Reneweable Energy*, 35, 122-127. doi:10.1016/j.renene.2009.03.034

Cabeza, L.F., Castell, A., Barreneche, C., De Gracia, A. and Fernández, A.I. (2011). Materials used as PCM in thermal energy storage in buildings: A review. *Renewable and Sustainable Energy Reviews*, 15(3), 1675-1695.

Carvalho, A.D., Moura, P., Vaz, G.C. and De Almeida, A.T. (2015a). Ground source heat pumps as high efficient solutions for building space conditioning and for integration in smart grids. (Submitted to the *Energy Conversion and Management* journal on 23 March 2015 and conditionally accepted on 3 May, depending on the second round revision of the suggestions carried out by the reviewers, Paper No. ECM-D-15-01125).

Carvalho, A.D., Mendrinos, D. and De Almeida, A.T. (2015b). Ground source heat pump carbon emissions and primary energy reduction potential for heating in buildings in Europe – Results of a case study in Portugal. *Renewable and Sustainable Energy Reviews*, 45, 755-768. doi:10.1016/ j.rser.2015.02.034

Çengel, Y.A. and Boles, M. A. (2004). *Thermodynamics: An Engineering Approach.* 5th edition. New York: McGraw-Hill.

Chiasson, A.D. and Yavuzturk, C. (2009). A design tool for hybrid geothermal heat pump systems in cooling-dominated buildings. *ASHRAE Transactions*, 115, 74-87.

Chua, K.J., Chou, S.K. and Yang, W.M. (2010). Advances in heat pump systems: A review. *Applied Energy*, 87, 3611–3624. doi:10.1016/j.apenergy.2010.06.014

CIAT - Compagnie Industrielle d'Applications Thermiques (2010a). *Connect 2: Microchip and LCD display electronic regulation and signaling electronic module for liquid coolers and heat pumps.* Culoz, France: Compagnie Industrielle d'Applications Thermiques.

CIAT - Compagnie Industrielle d'Applications Thermiques (2010b). *CIAT Cassette Coadis 2: Coanda Air Distribution.* Culoz, France: Compagnie Industrielle d'Applications Thermiques.

CIEPQPF - Centro de Investigação em Engenharia dos Processos Químicos e dos Produtos da Floresta (2010). *Relatório de análise: Sal hidratado – PCM.* Coimbra: Centro de Investigação em Engenharia dos Processos Químicos e dos Produtos da Floresta, Departamento de Engenharia Química da Faculdade de Ciências e Tecnologia da Universidade de Coimbra.

Connolly, D., Mathiesen, B.V., Østergaard, P.A., Möller, B., Nielsen, S., Lund, H., Persson, U., Werner, S., Grözinger, J., Boermans, T., Bosquet, M. and Trier, D. (2013). *Heat Roadmap Europe 2050 – Second pre-study for the EU 27.* Aalborg, Denmark: Department of Development and Planning, Aalborg University. ISBN: 978-87-91404-48-1

Connolly, D., Mathiesen, B.V., Østergaard, P.A., Möller, B., Nielsen, S., Lund, H., Persson, U., Nilsson, D., Werner, S. and Trier, D. (2012). *Heat Roadmap Europe 2050 – First pre-study for EU 27.* Aalborg, Denmark: Department of Development and Planning, Aalborg University.

Covrig, C.F., Ardelean, M., Vasiljevska, J., Mengolini, A., Fulli, G. and Amoiralis, E. (2014). *Smart Grid Projects Outlook 2014.* European Commission, Joint Research Centre, Institute for Energy and Transport. Luxembourg: Publications Office of the European Union. ISBN 978-92-79-38374-8

Daikin (2014). *Catálogo Geral 2014.* Oostende, Belgium: Daikin Europe NV.

Danish Government (2011). *Energy strategy 2050 – From coal, oil and gas to green energy.* Copenhagen: The Danish Ministry of Climate and Energy. ISBN: 978-87-92727-14-5

Day, A.R., Ratcliffe, M.S. and Shepherd, K. (2008). *Heating Systems, Plant and Control*. Wiley-Blacwell. ISBN: 978-1-4051-4763-7

De Almeida, A.T., Lopes, A., Carvalho, A., Mariano, J. and Nunes, C. (2004). Evaluation of fuel-switching opportunities in the residential sector. *Energy and Buildings*, 36, 195-203. doi:10.1016/j.enbuild.2003.11.003

Demirbas, M.F. (2006). Thermal energy storage and phase change materials: An overview. *Energy Sources, Part B*, 1, 85-95. doi:10.1080/009083190881481

Dincer, I. (2002). On thermal energy storage systems and applications in buildings. *Energy and Buildings*, 34, 377-388.

Dumas P. (2013). Geothermal Market in Europe. Presented at: *GeoPower and GeoHeat & Cool Global Congress*, *3rd edition*, 3-4 December 2013, Amsterdam, Netherlands. Available from: http://egec.info/wp-content/uploads/2013/12/Geothermal-MR-2013-2014.pdf [Accessed July 2014]

Ecofys (2013a). *Heat Pump Implementation Scenarios until 2030*. Ecofys study by order of European Heat Pump Association - EHPA. Köln: ECOFYS Germany GmBH.

Ecofys (2013b). *Appendix – Heat Pump Implementation Scenarios until 2030*. Ecofys study by order of European Heat Pump Association - EHPA. Köln: ECOFYS Germany GmBH.

Ecofys (2014). *Deep renovation of buildings – An effective way to decrease Europe's energy import dependency.* Ecofys study by order of Eurima. Köln: ECOFYS Germany GmBH.

EEA - European Environment Agency (2014). *Trends and projections in Europe 2014 – Tracking progress towards Europe's climate and energy targets for 2020*, *EEA Report No 6/2014*. Luxembourg: Publications Office of the European Union. ISBN 978-92-9213-491-4

EHPA - European Heat Pump Association (2009). *Outlook 2009 – European heat pump statistics – Outlook 2009*. Brussels: European Heat Pump Association.

EHPA - European Heat Pump Association (2014a)**.** *Heat Pumps Booklet 2014*. [Online]. Available from: http://www.ehpa.org/media/studies-and-reports/ [Accessed August 2014].

EHPA - European Heat Pump Association (2014b). *Heat pumps and EU targets.* [Online]. Available from: http://www.ehpa.org/technology/heat-pumps-and-eu-targets/eu-targets/ [Accessed August 2014].

EHPA - European Heat Pump Association (2015). *Energy labelling.* [Online]. Available from: http://www.ehpa.org/policy/energy-labelling/ [Accessed April 2015].

Ellerbrok, C. (2014). Potentials of demand side management using heat pumps with building mass as a thermal storage. *Energy Procedia*, 46, 214-219. doi:10.1016/j.egypro.2014.01.175

Energy Star (2014). *Energy Star Most Efficient 2014 – Geothermal Heat Pumps*. [Online]. Available from: http://www.energystar.gov/index.cfm?c=most_efficient.me_geothermal_heat_pumps [Accessed July 2014].

EU (2009a). *Directive 2009/125/EC of the European Parliament and of the Council of 21 October 2009 establishing a framework for the setting of ecodesign requirements for energy-related products*. OJ L 285, 31.10.2009, pp. 10-35.

EU (2009b). *Directive 2009/28/EC of the European Parliament and of the Council of 23 April 2009 on the promotion of the use of energy from renewable sources and amending and subsequently repealing Directives 2001/77/EC and 2003/30/EC*. OJ L 140, 5.6.2009, pp. 16-62.

EU (2010a). *Directive 2010/30/EU of the European Parliament and of the Council of 19 May 2010 on the indication by labelling and standard product information of the consumption of energy and other resources by energy-related products.* OJ L 153, 18.6.2010, pp. 1-12.

EU (2010b). *Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings (recast)*. OJ L 153, 18.6.2010, pp. 13-35.

EU (2012). *Directive 2012/27/EU of the European Parliament and of the Council of 25 October 2012 on energy efficiency, amending Directives 2009/125/EC and 2010/30/EU and repealing Directives.* OJ L 315, 14.11.2012, pp. 1-56.

Eurelectric (2010). *Power Statistics – 2010 Edition, Synopsis.* Brussels: Union of the Electricity Industry - EURELECTRIC.

Eurelectric (2012). *Power Statistics & Trends 2012 – Full Report*. Brussels: Union of the Electricity Industry - EURELECTRIC.

EurObserv'ER (2013). *Heat pumps barometer*. [Online]. Paris: EurObserv'ER. Available from: http://www.energies-renouvelables.org/observ-er/stat_baro/observ/baro218.pdf [Accessed July 2014].

European Commission (2007). *Commission Decision 2007/589/EC of 18 July 2007, establishing the guidelines for the monitoring and reporting of greenhouse gas emissions pursuant to Directive 2003/87/EC of the European Parliament and of the Council.* OJ L 229, 31.8.2007, pp. 1–85.

European Commission (2011a). *The Commission's new Energy Efficiency Plan*, *MEMO/11/149*. [Online]. Brussels, 8 March 2011. Available from: http://europa.eu/rapid/press-release_MEMO-11-149_en.htm?locale=en [Accessed April 2015].

European Commission (2011b). *A Roadmap for moving to a competitive low carbon economy in 2050*, COM(2011) 112 final/2. Brussels: European Commission.

European Commission (2011c). *Commission Delegated Regulation (EU) No 626/2011 of 4 May 2011 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to energy labelling of air conditioners*. OJ L 178, pp. 1-72.

European Commission (2011d). *Energy Roadmap 2050, COM (2011) 885 final.* Brussels: European Commission.

European Commission (2011e). *Impact Assessment, SEC (2011) 1566 final (Part 1/2). Accompanying the document Energy Roadmap 2050, COM(2011) 885 final.* Commission staff working paper. Brussels: European Commission.

European Commission (2011f). *Impact Assessment, SEC (2011) 1569 final (Part 2/2). Accompanying the document Energy Roadmap 2050, COM(2011) 885 final.* Brussels: European Commission.

European Commission (2012). *Commission Regulation (EU) No 206/2012 of 6 March 2012 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for air conditioners and comfort fans*. OJ L 72, 10.3.2012, pp. 7-27.

European Commission (2013a). *Commission Regulation (EU) No 813/2013 of 2 August 2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters.* OJ L 239, 6.9.2013, pp. 136-161.

European Commission (2013b). *Commission Delegated Regulation (EU) No 811/2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of space heaters, combination heaters, packages of space heater, temperature control and solar device and packages of combination heater, temperature control and solar device*. OJ L 239, 6.9.2013, pp. 1-82.

European Commission (2013c). *Commission Decision 2013/114/EU of 1 March 2013, establishing the guidelines for Member States on calculating renewable energy from heat pumps from different heat pump technologies pursuant to Article 5 of Directive 2009/28/EC of the European Parliament and of the Council.* OJ L 62, 6.3.2013, pp. 27-35.

European Commission (2014a). *Energy efficiency and its contribution to energy security and the 2030 framework for climate and energy policy, COM(2014) 520 final.* Brussels: European Commission.

European Commission (2014b). *Annual report from the European Commission to the European Parliament and the Council on progress towards meeting the Kyoto and EU 2020 objectives (as required under Article 21 of the EU Monitoring Mechanism Regulation), COM(2014) 689 final.* Brussels: European Commission.

European Commission (2015a). *2030 framework for climate and energy policies*. [Online]. Available from: http://ec.europa.eu/clima/policies/2030/index_en.htm [Accessed February 2015].

European Commission (2015b). *Energy topics - Energy efficiency - Buildings*. [Online]. Available from: http://ec.europa.eu/energy/en/topics/energy-efficiency/buildings [Accessed April 2015].

Eurostat (2015). *Eurostat Database*. [Online]. Available from: http://ec.europa.eu/eurostat/data/database [Accessed January 2015]

FERC - Federal Energy Regulatory Commission (2014). *Assessment of Demand Response and Advanced Metering - Staff Report*. [Online]. Available from: http://www.ferc.gov/industries/electric/indus-act/demand-response.asp [Accessed April 2015].

Ferrari, S. (2007). Building envelope and heat capacity: re-discovering the thermal mass for winter energy saving. In: *Proceedings of the 2nd PALENC Conference and 28th AIVC Conference on Building Low Energy Cooling and Advanced Ventilation Technologies in the 21st Century*, 27-29 September 2007, Crete Island, Greece, Vol. 1, pp. 346-351.

Fisher, D.E., Rees, S.J., Padhmanabhan, S.K. and A. Murugappan, A. (2006). Implementation and validation of ground-source heat pump system models in an integrated building and system simulation environment. *HVAC&R Research*, 12(3a), 693-710.

Gellings, C.W. (1985). The concept of demand-side management for electric utilities. *Proceedings of the IEEE*, 73(10), 1468-1470.

Geominho (2010). *Relatório de sonda piloto geotérmica - Fábrica dos Mirandas, Projecto GroundMed*. Póvoa de Lanhoso, Portugal: Geominho - Perfurações Geológicas do Minho, Lda.

Ground-Med (2014). *GROUND-MED Project - Advanced ground source heat pump systems for heating and cooling in Mediterranean climate*. [Online]. Supported by the European Commission within the Seventh Framework Programme, Grant Agreement No TREN/FP7EN/218895/. Available from: http://www.groundmed.eu/about_ground_med/ [Accessed July 2014].

Grundfos (2009). *Grundfos Data Booklet: MAGNA, UPE Series 2000 circulator pumps*. Bjerringbro, Denmark: Grundfos A/S.

Hamada, Y. and Fukai, J. (2005). Latent heat thermal energy storage tanks for space heating of buildings: Comparison between calculations and experiments. *Energy Conversion and Management*, 46, 3221-3235. doi:10.1016/j.enconman.2005.03.009

Hedegaard, K. and Münster, M. (2013). Influence of individual heat pumps on wind power integration – Energy system investments and operation. *Energy Conversion and Management*, 75, 673-684. doi:10.1016/j.enconman.2013.08.015

Heier, J., Bales, C. and Martin, V. (2015). Combining thermal energy storage with buildings – A review. *Renewable and Sustainable Energy Reviews*, 42, 1305-1325. doi:10.1016/j.rser.2014.11.031

Hesaraki, A., Holmberg, S. and Haghighat, F. (2015). Seasonal thermal energy storage with heat pumps and low temperatures in building projects – A comparative review. *Renewable and Sustainable Energy Reviews*, 43, 1199-1213. doi:10.1016/j.rser.2014.12.002

Hewitt, N.J. (2012). Heat pumps and energy storage – The challenges of implementation. *Applied Energy*, 89, 37-44. doi:10.1016/j.apenergy.2010.12.028

Huchtemann, K. and Müller, D. (2012). Evaluation of a field test with retrofit heat pumps. Building *Environment*, 53, 100-106. doi:10.1016/j.buildenv.2012.01.013

IEA - International Energy Agency (2011). *Technology Roadmap – Energy-efficient Buildings: Heating and Cooling Equipment.* Paris: International Energy Agency.

IEA - International Energy Agency (2013a.) *Energy Technology Initiatives 2013: Implementation through Multilateral Co-operation*. Paris: International Energy Agency.

IEA - International Energy Agency (2013b). *CO_2 Emissions From Fuel Combustion Highlights.* Paris: International Energy Agency.

IEA - International Energy Agency (2014). *Technology Roadmap – Energy storage.* Paris: International Energy Agency.

Incropera, F.P. and Dewitt, D.P. (2001). *Fundamentals of Heat and Mass Transfer*. 5th Edition. New York: John Wiley & Sons. ISBN: 978-0-471-38650-6

Jegadheeswaran, S. and Pohekar, S.D. (2009). Performance enhancement in latent heat thermal storage system: A review. *Renewable and Sustainable Energy Reviews*, 13, 2225-2244.

Kalnæs, S.E. and Jelle, B.P. (2015). Phase change materials for building applications: A state-of-the-art review and future research opportunities. *Energy and Buildings*, 94, 150-176. doi:10.1016/j.enbuild.2015.02.023

Kanhong, W., Jingjing, X., Keying, L. and Yuyang, S. (2010). Experimental research on solar assisted GSHP heating system with a latent heat storage tank. *Proceedings of the 2010 International Conference on E-Product E-Service and E-Entertainment (ICEEE 2010)*, 7-9 November 2010, Henan, China.

Kenisarin, M. and Mahkamov, K. (2007). Solar energy storage using phase change materials. *Renewable and Sustainable Energy Reviews*, 11, 1913-1965.

Kennedy, J., Fox, B. and Flynn, D. (2009). Use of electricity price to match heat load with wind power generation. In: *Proceedings of the 1st International Conference on Sustainable Power Generation and Supply (SUPERGEN '09)*, 6-7 April 2009, Nanjing, China.

Lennox (2014). *Lennox Product Catalogue 2013-2014.* [Online]. Available from: http://www.lennoxemeia.com/sites/default/files/download/catalogues/Catalogue-E.pdf [Accessed July 2014].

Luo, J., Rohn, J., Bayer, M., Priess, A., Wilkmann, L. and Xiang, W. (2014). Heating and cooling performance analysis of a ground source heat pump system in Southern Germany. *Geothermics*, 53, 57-66. doi:10.1016/j.geothermics.2014.04.004

Mehling H. and Cabeza L. (2008). *Heat and cold storage with PCM - An up to date introduction into basics and applications*. Berlin: Springer. ISBN: 978-3-540-68556-2

Moreno, P., Solé, C., Castell, A. and Cabeza, L.F. (2014). The use of phase change materials in domestic heat pump and air-conditioning systems for short term storage: A review. *Renewable and Sustainable Energy Reviews*, 39, 1-13. doi:10.1016/j.rser.2014.07.062

Morrone, B., Coppola, G. and Raucci, V. (2014). Energy and economic savings using geothermal heat pumps in different climates. *Energy Conversion and Management*, 88, 189–198. doi:10.1016/j.enconman.2014.08.007

Moura, P.S. and De Almeida, A.T. (2010a). Multi-objective optimization of a mixed renewable system with demand-side management. *Renewable and Sustainable Energy Reviews*, 14, 1461-1468. doi:10.1016/j.rser.2010.01.004

Moura, P.S. and De Almeida, A.T. (2010b). The role of demand-side management in the grid integration of wind power. *Applied Energy*, 87, 2581-2588. doi:10.1016/j.apenergy.2010.03.019

Moura, P.S., López, G.L., Moreno, J.I. and De Almeida, A.T. (2013). The role of Smart Grids to foster energy efficiency. *Energy Efficiency*, 6, 621-639. doi:10.1007/s12053-013-9205-y

Mustafa Omer, A. (2008). Ground-source heat pumps systems and applications. *Renewable and Sustainable Energy Reviews*, 12, 344-371. doi:10.1016/j.rser.2006.10.003

Nishio, K. and Hoshino, Y. (2010). *Impacts of Electrification on CO_2 Emission Reduction Potentials in the G7 Countries.* Tokyo: Socio-economic Research Center, CRIEPI - Central Research Institute of Electric Power Industry of Japan. Available from: http://criepi.denken.or.jp/en/serc/research_re/download/10004dp.pdf [Accessed July 2014].

Nowak, T. (2013). Heat Pump and GSHP market aspects and economic assessment. Presentation at: *GroundMed Workshop*, 26 September 2013, Coimbra, Portugal. Available from: http://www.groundmed.eu/training/course_6_coimbra/ [Accessed July 2014].

Odyssee (2015). *Odyssee database on energy efficiency and CO_2 indicators*. [Online]. Available from: http://www.indicators.odyssee-mure.eu/ [Accessed January 2015].

Omagari, Y., Sugihara, H. & Tsuji, K. (2008). An Economic Impact of Thermal Storage Air-conditioning Systems in Consideration of Electricity Market Prices. In: *Proceedings of the 2008 IEEE Power and Energy Society General Meeting – Conversion and Delivery of Electrical Energy in the 21st Century*, 20-24 July 2008, Pittsburgh, USA.

Osterman, E., Tyagi, V.V., Butala, V., Rahim, N.A. and Stritih, U. (2012). Review of PCM based cooling technologies for buildings. *Energy and Buildings*, 49, 37-49. doi:10.1016/j.enbuild.2012.03.022

Parameshwaran, R., Kalaiselvam, S., Harikrishnan, S. and Elayaperumal, A. (2012). Sustainable thermal energy storage technologies for buildings: A review. *Renewable and Sustainable Energy Reviews*, 16(5), 2394-2433.

Pardo, N., Vatopoulos, K., Krook-Riekkola, A., Moya, J.A. and Perez, A. (2012). *Heat and cooling demand and market perspective.* European Commission, Joint Research Centre, Institute for Energy and Transport. Luxembourg: Publications Office of the European Union. ISBN 978-92-79-25310-2

PCM Energy P. (2015). PCM Energy Private Ltd, India. [Online]. Available from: http://www.pcmenergy.com [Accessed April 2015].

PCM Products (2011). *Plus-ICE Phase Change Materials (PCM) Thermal Energy Storage (TES) Design Guide.* Yaxley, UK: Phase Change Material Products Ltd.

PCM Products (2015). Phase Change Material Products Ltd, UK. [Online]. Available from: http://www.pcmproducts.net [Accessed April 2015].

Pinel, P., Cruickshank, C.A., Beausoleil-Morrison, I. and Wills, A. (2011). A review of available methods for seasonal storage of solar thermal energy in residential applications. *Renewable and Sustainable Energy Reviews*, 15, 3341-3359. doi:10.1016/j.rser.2011.04.013

Pomianowski, M., Heiselberg, P. and Zhang, Y. (2013). Review of thermal energy storage technologies based on PCM application in buildings. *Energy and Buildings*, 67, 56-69. doi:10.1016/ j.enbuild.2013.08.006

Qi, Q., Deng, S. and Jiang, Y. (2008). A simulation study on a solar heat pump heating system with seasonal latent heat storage. *Solar Energy*, 82, 669-675.

Qureshi, W.A., Nair, N.C. and Farid, M.M. (2008). Demand side management through efficient thermal energy storage using phase change material. In: *Proceedings of the 2008 Australasian Universities Power Engineering Conference (AUPEC '08)*, 14-17 December 2008, Sydney, Australia.

Rasmussen P. (2011). *Energy labelling and ecodesign requirements for heat pumps.* Prepared for the Danish Energy Agency. Danish Technological Institute. Available from: http://www.eceee.org/ ecodesign/products/boilers/resolveuid/a11dcc2a42440061ace1f40aaf961251 [Accessed July 2014].

Rathod, M.K. and Banerjee, J. (2013). Thermal stability of phase change materials used in latent heat energy storage systems: A review. *Renewable and Sustainable Energy Reviews*, 18, 246-258.

REN - Rede Elétrica Nacional (2015). *Electricidade - Centro de Informação*. [Online]. Available from: http://www.centrodeinformacao.ren.pt/ [Accessed February 2015].

Reynders, G., Nuytten, T. and Saelens, D. (2013). Potential of structural thermal mass for demand-side management in dwellings. *Building and Environment*, 64, 187-199. doi:10.1016/j.buildenv. 2013.03.010

RHC-Platform - European Technology Platform on Renewable Heating and Cooling (2011). *2020-2030-2050 Common Vision for the Renewable Heating & Cooling sector in Europe.* Luxembourg: Publications Office of the European Union. ISBN 978-92-79-19056-8

RHC-Platform - European Technology Platform on Renewable Heating and Cooling (2013). *Strategic Research and Innovation Agenda for Renewable Heating and Cooling*. Luxembourg: Publications Office of the European Union. ISBN 978-92-79-30657-0

Rousse, D.R., Salah N.B. and Lassue, S. (2009). An overview of phase change materials and their Implication on power demand. In: *Proceedings of the 9th IEEE Electrical Power and Energy Conference (EPEC 2009)*, 22-23 October 2009, Montreal, Canada.

Rubitherm (2015). *Rubitherm PCM's*. [Online]. Available from: http://www.rubitherm.de/english/ index.htm [Accessed April 2015].

Sandnes, B. and Rekstad, J. (2006). Supercooling salt hydrates: Stored enthalpy as a function of temperature. *Solar Energy*, 80, 616-625.

Saner, D., Juraske, R., Kübert, M., Blum, P., Hellweg, S. and Bayer, P. (2010). Is it only CO_2 that matters? A life cycle perspective on shallow geothermal systems. *Renewable and Sustainable Energy Reviews*, 14, 1798-1813. doi:10.1016/j.rser.2010.04.002

Sarbu, I. and Sebarchievici, C. (2014). General review of ground-source heat pump systems for heating and cooling of buildings. *Energy and Buildings*, 70, 441-454. doi:10.1016/j.enbuild.2013.11.068

Schibuola, L., Scarpa, M. and Tambani, C. (2015). Demand response management by means of heat pumps controlled via real time pricing. *Energy and Buildings*, 90, 15-28. doi:10.1016/j.enbuild.2014.12.047

Self, S.J., Reddy, B.V. and Rosen, M.A. (2013). Geothermal heat pump systems: Status review and comparison with other heating options. *Applied Energy*, 10, 341-348. doi:10.1016/j.apenergy.2012.01.048

Sharma, A., Tyagi, V., Chen, C. and Buddhi, D. (2009). Review on thermal energy storage with phase change materials and applications. *Renewable and Sustainable Energy Reviews*, 13(2), 318-345.

Smart Cities Stakeholder Platform (2013). *Key to Innovation – Integrated Solution: Heat pump and micro-CHP as complementary boiler alternatives.* [Online]. Available from: http://eu-smartcities.eu/sites/all/files/Heat pump 2014.pdf [Accessed July 2014]

Soares, N., Costa, J.J., Gaspar, A.R. and Santos, P. (2013). Review of passive PCM latent heat thermal energy storage systems towards buildings' energy efficiency. *Energy and Buildings*, 59, 82-103. doi:10.1016/j.enbuild.2012.12.042

Spitler, J.D. and Southard, L.E. (2014). *Performance of the HVAC Systems at the ASHRAE Headquarters Building - Final Report*. Springfield, IL: GEO - The Geothermal Exchange Organization

Stadler, I. (2008). Power grid balancing of energy systems with high renewable energy penetration by demand response. *Utilities Policy*, 16, 90-98.

Stevens, V., Colin, C. and Grunau, B. (2013) *Thermal Storage Technology Assesment. An introductory assessment of thermal storage in residential cold climate construction*. Alaska: CCHRC - Cold Climate Housing Research Center.

Stratego (2015). *STRATEGO Project - Multi level actions for enhanced Heating and Cooling plans*. [Online]. Supported by the European Commission within the Intelligent Energy Europe Programme. Available from: http://stratego-project.eu/ [Accessed March 2015].

Subbarao, K., Burch, J., Hancock, E. and Jeon, H. (1985). Measurement of effective thermal capacitance in buildings. In: *Proceedings of Thermal Performance of the Exterior Envelopes of Buildings III*, 2-5 December 1985, Clearwater Beach, Forida, USA. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, pp. 351-363.

Sun, Y., Wang, S., Xiao, F. and Gao, D. (2013). Peak load shifting control using different cold thermal energy storage facilities in commercial buildings: A review. *Energy Conversion and Management*, 71, 101-114. doi:10.1016/j.enconman.2013.03.026

Tatsidjodoung, P., Le Pierrès, N. and Luo, L. (2013). A review of potential materials for thermal energy storage in building applications. *Renewable and Sustainable Energy Review*s, 18, 327-349. doi:10.1016/j.rser.2012.10.025

Tyagi, V.V. and Buddhi, D. (2007). PCM thermal storage in buildings: A state of art. *Renewable and Sustainable Energy Reviews*, 11, 1146-1166.

Urchueguía, J.F., Zacarés, M., Corberán, J.M., Montero, Á., Martos, J. and Witte, H. (2008). Comparison between the energy performance of a ground coupled water to water heat pump system and an air to water heat pump system for heating and cooling in typical conditions of the European Mediterranean coast. *Energy Conversion and Management*, 49, 2917–2923. doi:10.1016/j.enconman.2008.03.001

Ure, Z. (2008). Application of PCM thermal energy storage. *Consulting Engineer Magazine*, May 2008, 35-36.

Wang, H., Qi, C., Wang, E. and Zhao, J. (2009). A case study of underground thermal storage in a solar-ground coupled heat pump system for residential buildings. *Renewable Energy*, 34, 307-314. doi:10.1016/j.renene.2008.04.024

Xu, J., Wang, R.Z. and Li, Y. (2014). A review of available technologies for seasonal thermal energy storage. *Solar Energy*, 103, 610-638. doi:10.1016/j.solener.2013.06.006

Yau, Y.H. and Rismanchi, B. (2012). A review on cool thermal storage technologies and operating strategies. *Renewable and Sustainable Energy Reviews*, 16, 787-797. doi:10.1016/j.rser.2011.09.004

Zalba, B., Marín, J.M., Cabeza, L.F. and Mehling, H. (2003). Review on thermal energy storage with phase change: materials, heat transfer analysis and applications. *Applied Thermal Engineering*, 23, 251-283.

Zhai, X.Q., Wang, X.L., Wang, T. and Wang, R.Z. (2013). A review on phase change cold storage in air-conditioning system: Materials and applications. *Renewable and Sustainable Energy Reviews*, 22, 108-120. doi:10.1016/j.rser.2013.02.013

Zhang, Y., Zhou, G., Lin, K., Zhang, Q. and Di, H. (2007). Application of latent heat thermal energy storage in buildings: State-of-the-art and outlook. *Building and Environment*, 42, 2197-2209.

Zhao, Z., Feng, W., Mi, H., Cheng, H., Yun, L. and Gong, X. (2014). Experimental study of a solar assisted ground source heat pump for heating. *Journal of Automation and Control Engineering*, 2, 146-149. doi:10.12720/joace.2.2.146-149

Zhou, D., Zhao, C.Y. and Tian, Y. (2012). Review on thermal energy storage with phase change materials (PCMs) in building applications. *Applied Energy*, 92, 593-605.

Zhu, N., Hu, P., Xu, L., Jiang, Z. and Lei, F. (2014). Recent research and applications of ground source heat pump integrated with thermal energy storage systems: A review. *Applied Thermal Engineering*, 71, 142-151. doi:10.1016/j.applthermaleng.2014.06.040

Zhu, N., Ma, Z. and Wang, S. (2009). Dynamic characteristics and energy performance of buildings using phase change materials: A review. *Energy Conversion and Management*, 50, 3169-3181. doi:10.1016/j.enconman.2009.08.019