UNIVERSITÀ DEGLI STUDI DI PADOVA

Simulation study of Demand Side Management with heat pumps installed in a typical single family house with low temperature radiators

Master thesis report

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LIST OF ABBREVIATIONS

Air Source Heat Pump: ASHP

Buffer Storage Tank: BST

Coefficient Of Performance: COP

Demand Side Management: DSM

Domestic Hot Water: DHW

Ground Source Heat Pump: GSHP

Heating Ventilation and Air Conditioning: HVAC

Over Degree Temperature Factor: ODTF

Seasonal Primary Energy Efficiency Factor: SPEEF

Seasonal Performance Factor: SPF

Under Degree Temperature Factor: UDTF

1 INTRODUCTION

The building sector is estimated to utilize about the 40% of the overall energy use and its Greenhouse Gas Emission represents approximately 24% of the global total (Mikk Maivel, 2014). Over recent years has completely changed the global attitude towards the ways to supply energy to the buildings. In fact the awareness and the comprehension of the effect due to the conventional energy generation has conducted to a shift in opinion regarding the energy sector both in the Governments and normal people. Currently 82% of primary energy need is supplied by non-renewable generating techniques such as fossil fuel combustion.
The problems related to the burning of these fossil fuels are mainly two:

- Environmental impact on environment in terms of pollution and greenhouse effect;
- Decrease of traditional fossil fuel stocks

For this reason it’s important to consider sustainable and renewable methods in order to satisfy electricity and heating demand.
This kind of renewable progression and the plan of significant diminution of carbon pollution due to energy production have been encouraged both at domestic level and international level. In order to achieve this goal Kyoto Protocol had determined several CO_2 targets to be met by 2012 and at the same time European Commission has set its own goals which have to be accomplished by 2020. In Germany this trend of renewable progression is more evident than in other countries.
In fact comparing data regarding the electricity production in Germany during the last 25 years is clear the rise of renewable energy production.
In 1990 the brown and hard coal were used to produce 57% of the overall power; nowadays these sources are still the most important ones but they account for 44% of the total electricity production (Destatis, 2015).
On the other hand the importance of renewable forms of energy is on the rise. While in 1990 renewable energy sources represented only 4% of total energy production, their contribute in 2014 is risen up to 26% (Destatis, 2015).
In the following list it’s possible to highlight percentages relative to the gross electricity production in Germany during 2014. These data are taken from Destatis, one of the most important German statistic database that gives information also about Economic sectors, Industry, Energy, Construction and other branches.
Distribution of gross electricity production in 2014 depending on the energy source:

- Brown Coal 25,6%;
- Nuclear Energy 15,9%;
- Hard Coal 18,0%;

- Natural Gas 9,6%;
- Mineral Oil products 0,8%;
- Renewable energy sources 25,8%;
- Other energy sources 4,3%.

1.1 Motivation

Since the building sector is one of the most important consuming sector it’s necessary to evaluate strategies for decreasing its consumption from non-renewable energies.
In order to achieve reductions of energy consumptions in the construction industry, it’s necessary to build up new buildings with improved envelope in terms of insulation and use energy systems and sources that achieve low CO_2 emissions.
The main consumptions in the building sector are certainly the energy used for space heating, ventilation, cooling, domestic hot water production and lighting (De Carli, 2014).
While in the last years the most common heating systems for supplying energy to the building were traditional heating boilers, nowadays Germany is incentivizing more efficient heating and cooling technologies (GermanyTrade&Invest, 2015).
These incentives and promotions have led to a significant market shift in the heating and cooling sector. In particular the number of renewable efficient technologies (such as CHP, heat pumps, solar thermal, condensing heating boilers and pellet heating systems) has increased enormously. For instance, on the basis of data published by the European Heat Pump Association, currently heat pumps account for 9% of the German heat generator market (EHPA, 2015). In particular in 2013 were sold in Germany 60000 heat pumps. About 37400 were air to water heat pumps, 19400 were brine to water heat pumps and only 2800 water ton water heat pumps (EHPA, 2015).
Therefore in this work will be considered the Heat Pump’s technology and its potential for reducing energy use for the benefit of the environment. As shown before, since the most common heat pumps installed in Germany are Air and Ground Source Heat Pumps, only these will be considered and the building typology analyzed will be a Single Family House.
In this project will be studied the behavior of low temperature radiators that work with a design value for the flow temperature around 45° C.
In fact in new and well insulated buildings the trend is to use heating distribution systems characterized by lower values of supply temperature if compared with conventional ones.
So there is no need to use radiator with flow temperature around 70° C also because they can’t be in combination with heat generators as heat pumps. Effectively in this case heat pumps would work in inefficient conditions or even would not reach these high level of temperature.

1.2 Aim of the study and Demand Side Management with heat pumps

In the recent years many studies were carried out regarding the Demand Side Management applied to electrical heat generators (Kelly, 2014). Usually heat pumps are coupled with underfloor heating distribution systems that work with lower temperatures than radiators and in the thermal plant is not considered a thermal storage.

However if heat pumps are coupled with buffer storage tanks it’s possible to investigate a heating system able to decouple the heat and electricity demand of a building. In fact using heat pumps as heat generators it’s possible to study and analyze the mismatch existing between heat demand of the building (in terms of space-heating and domestic hot water) and the electrical feeding.

This goal can be reached using the concept of thermal energy storage.

The thermal energy storage can be used in buildings as a temporary storage of thermal energy and heat. Thermal energy storage is widely recognized as a way to integrate more renewable energies and can play an important role also in the demand side.

Basically Demand Side Management applied to heat pumps for storing thermal energy is applied nowadays in accordance with two different approaches:

- The first solution is to store the energy activating thermal storage capacity in the structures of a building. For instance this approach is used in Thermally Activated Building Structures (De Carli, 2008);
- The second approach that will be examined in the present work is to store thermal energy using the water capacity of buffer storage tank.

Demand Side Management was defined in the 1980s as “the planning, implementation and monitoring of those utility activities designed to influence customer use of electricity in ways that will produce desired changes in the utility’s load shape, i.e., changes in the pattern and magnitude of a utility’s load” (A. Arteconi, 2012).

So all the strategies that have as aim the yearly decreasing of energy consumption, the moving of electrical consumption from high costs to low costs of electricity or the increasing of renewable energies use are considered Demand Side Management.

In recent years these types of system have demonstrated their capability to shift electrical load from high-peak to off-peak hours. In this way they can become a powerful tool in Demand Side Management (DSM).

The implementation of Demand Side Management in this work will be applied for retaining and storing thermal energy for a later use.

In particular the impact of DSM with the heat pump’s control strategies will be analyzed.

In fact electrically driven heat pumps facilitate demand flexibility because thermal storage water capacity provides the ability to shift operation time; moreover it will be clearer how to control heat pumps depending on the renewable energies production.
So the aim of this thesis is to investigate the potential decrease of the conventional energy consumption of the heat pump and, at the same time, the increased portion of energy consumption from renewable energies. Obviously it makes sense only if the total electrical consumption remains the same or increases a little bit.
Furthermore the effect of this Demand Side Management applied to heat pump will be shown in terms of indoor thermal behavior of the building inasmuch it's important to prevent nominal room temperature changes. In fact there is no sense to enhance renewable energy use for heat pumps if there are negative consequences for the thermal comfort.
It's important to remind that maintaining comfort conditions for building's occupants is one of the most important goals of HVAC (Heating Ventilation and Air Conditioning) systems.
Thermal comfort will be evaluated in terms of under-cooling and over-heating during the heating season from September to April compared to an optimal range of temperature that will be introduced at a later stage.

The following is a brief and indicative list of the most important steps made in order to reach the master thesis' goal:

- Explanation of electricity data;
- Building model;
- Simulation tools;
- Heating load calculation;
- Explanation of the heat pump systems;
- Control strategies and optimizations;
- Results.

2 ELECTRICITY DATA

As mentioned beforehand during the last years Germany has led to a quick expansion of the renewable energy market and in particular solar and wind power generation is increasing.

By the end of 2012, 32600 MW of solar power and 31300 MW wind power were connected to the German power grid. The peak consumption in Germany is round 70000 MW, instead the average value of the electricity consumption is nearly 50000 MW (Marek Miara, 2011).

These data illustrate how much high is the impact of renewable electricity generation if compared to the overall electricity generation in Germany; in fact as said before the electricity production from renewable energies is circa 26 % of the overall production in Germany (Destatis, 2015). Solar and wind power production changes during the year and its main characteristic is to be highly intermittent during the day because it depends on the hour of the day (only solar power production) and also on the weather condition.

So nowadays the main problem in this context is to match this intermittent renewable production from solar and wind source with the electrical demand.

In order to balance renewable energy production and demand it's necessary to move the electrical consumption from times with little to times with high renewable power production and consequently also shift operation of electric consumers from times with high electricity demand to times with low consumption. In this study, this load shifting concept is investigated based on the electricity data of a small city in Northern Hessen, Germany. The main characteristic of the energy generation in this city is to have a positive annual energy balance between energy consumption and renewable energy production, so this means that in one year the renewable energy production can cover totally the electricity demand.

In order to understand what was the potential of Demand Side Management applied to heat pumps it was necessary to make a preliminary analysis of electrical data relative to 2013 furnished by the electrical grid operator.

In the electricity data it's possible to find information about the electrical load required by household and industrial consumers, but also regarding the renewable energy production.

Actually data concerning the renewable energies production were real only for solar and biomass production, instead the wind energy production has been estimated with wind velocity measurements. In fact the wind farm will be completely installed not before the end of 2015. In particular have been considered 10 MW_P of installed solar capacity in 2013 and 12 MW_P of installed wind capacity carried out in 2015.

These electricity data were available with a time step of 15 minutes.

Figure 1 enables to see the annual profile of renewable energy production considering the three contributes of biomass, solar and wind respectively.

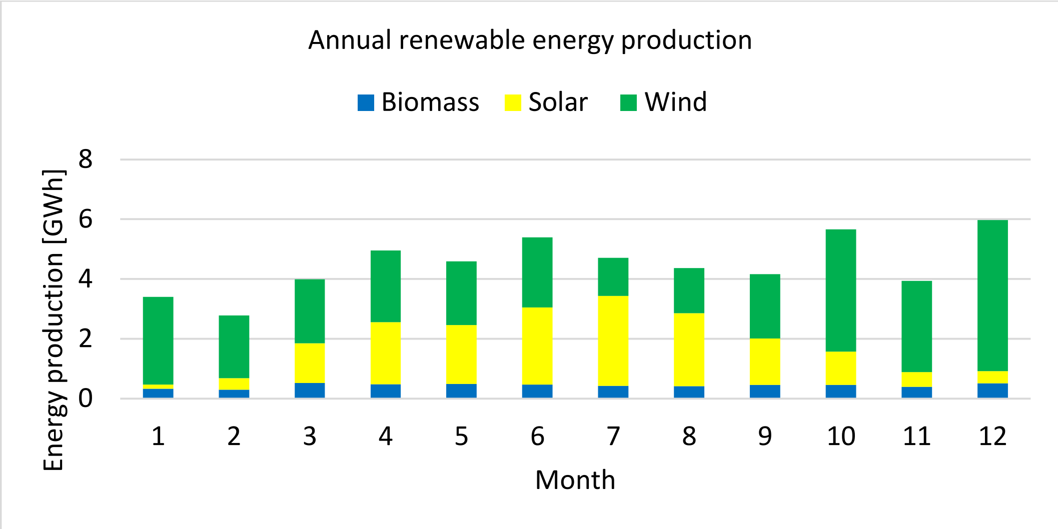

Figure 1 Annual renewable energy production

Figure 1 provides an overview of the renewable energy production that as said beforehand is intermittent during the months.

In fact while the biomass contribute is almost constant during the whole year, solar and wind contribute have very high fluctuations depending on the month considered.

For example solar production is higher during the summer time, instead the wind production reaches its peak during the winter month of December.

In the following list are summarized the renewable energy productions divided by energy source:

- Annual solar energy production: 17 GWh/year;
- Annual expected wind energy production: 31 GWh/year;
- Annual energy production from biomass: 5 GWh/year;
- Annual renewable energy production: 54 GWh/year.

It's important also to show the electrical demand required by the residential and commercial load during the whole year. It takes continually values included between 3,7 and 4,6 GWh/month and takes higher values during the winter period.

Overall the annual electricity demand in the household and commercial sector in 2013 was 49 GWh/year; so in the annual balance it could be completely covered by the renewable energy production. In fact the ratio between the annual renewable energy production and the annual electrical demand is 1,10.

Monthly energy demand is depicted in Figure 2.

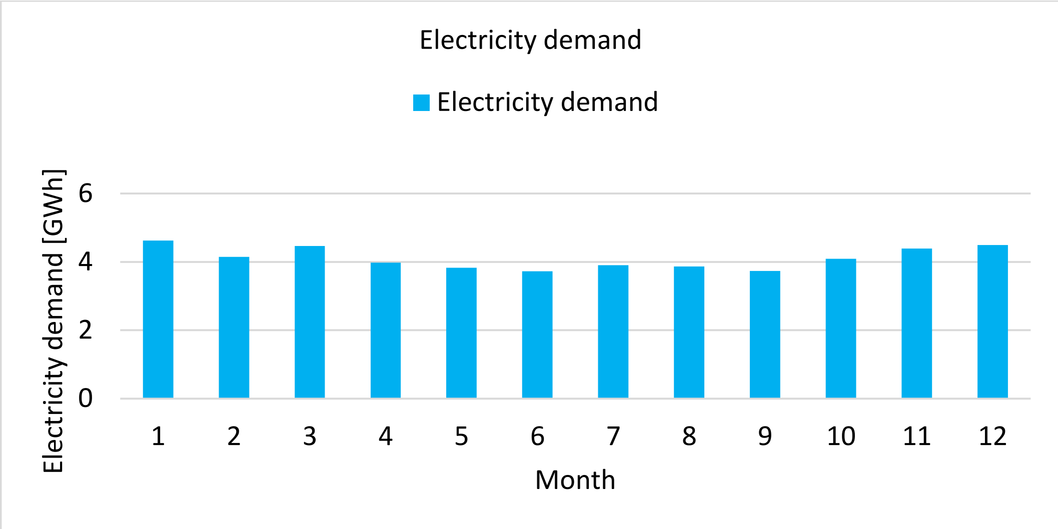

Figure 2 Electricity demand

Once renewable energy production and electricity demand have been considered it's important to define a parameter that correlates these two variables at any given time.

This parameter is the residual load and is defined as the instantaneous difference between the electricity household demand and the amount supplied by renewable energies.

Only the electrical load has been considered because in this work the aim is to evaluate the potential energy consequences due to the control of heat pumps in residential buildings as for instance single family houses.

Positive residual load means that the electricity demand in the considered time step is higher than the renewable energy production from solar source, wind source and biomass. On the other hand negative residual load means that variable renewable production not only covers the household electricity demand, but also provides surplus electricity.

Because of the higher production from renewable energies during the summer time and for instance also in the month of December it will take a monthly negative value.

On the other hand if it's considered a shorter time horizon the intermittence and unpredictability of renewable energy production cause that often the power demand can't be totally covered by renewable energy feeding also during the months characterized by negative residual load.

The following pictures illustrate the monthly value for the residual load (Figure 3) and its cumulative annual graph (Figure 4) (Yu, 2012). The latter represents the residual load annual evolution starting from its higher values to its lower ones.

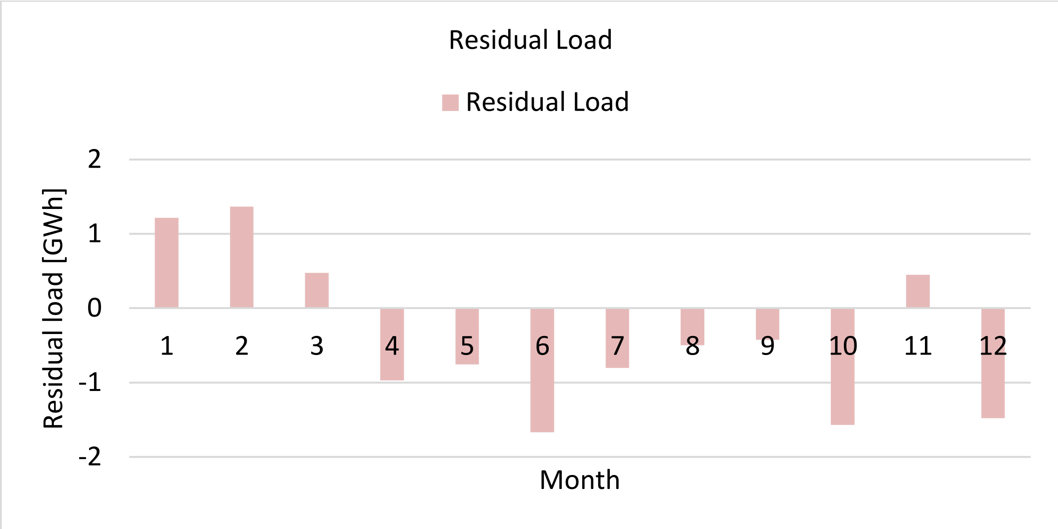

Figure 3 Residual load

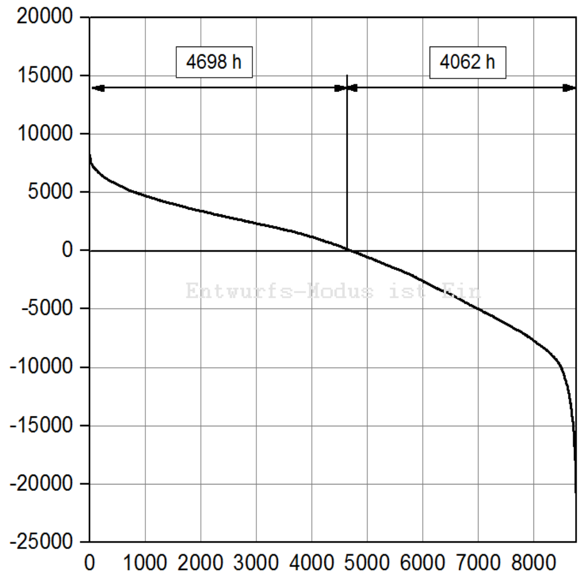

Figure 4 Cumulative graph of residual load (Yu, 2012)

Especially in the Figure *4* (Yu, 2012) is highlighted that for circa half a year there are negative values for the residual load. More specifically the residual load takes negative values for 4062 hours and positive values for 4698 hours during a year. The purpose of Demand Side Management applied to the control of heat pumps is to move electrical consumption that nowadays are in the left side of the graph to time intervals with higher production from renewable sources. That means make this graph more flat.

2.1 Real-time electricity price model

The concept of residual load has clearly consequences also in the electricity cost.
In fact when the residual load takes high positive values that means the electrical energy has to be supplied by conventional power generation systems that causes higher cost for the electricity. On the contrary when the residual load takes high negative values that means the electrical demand can be covered by renewable energies with resulting lower costs for the electricity.
In order to consider the real time value of the residual load it has been necessary to attribute a different Tariff depending on the residual load value (Yu, 2015). Basically this subdivision has been made considering the following residual load ranges:

- Tariff 1 if Residual Load takes lower value than -4300 kW;
- Tariff 2 if Residual Load takes value between -4300 kW and 0 kW;
- Tariff 3 if Residual Load takes value between 0 kW and 3000 kW;
- Tariff 4 if Residual Load takes higher value than 3000 kW.

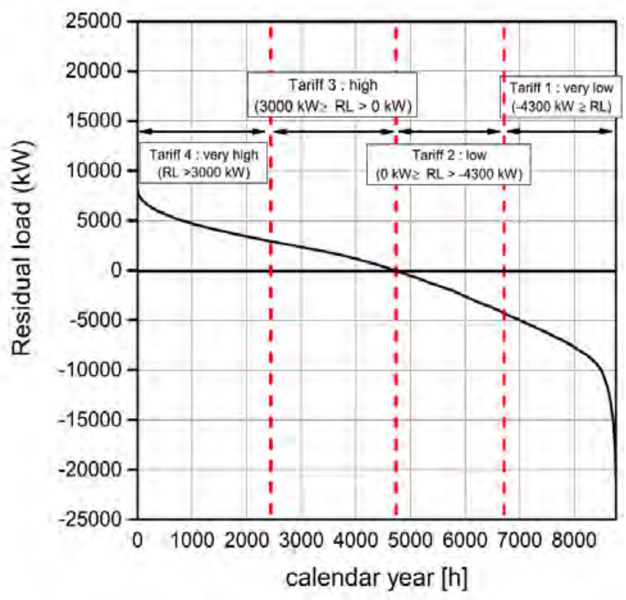

Figure 5 Tariff depending on the residual load (Yu, 2015)

This subdivision shown in Figure 5 (Yu, 2015) will be useful during the dynamic simulations because depending on the residual load's value will be analyzed different control strategies regarding the Demand Side Management applied to heat pumps. In order to make clearer the concept of intermittence of renewable energies production during the day and during the year it's important to depict the following figure (Figure 6). It represents the distribution of the electricity Tariff depending on the hour of the day considered. This graph shows the hourly Tariff of the electricity as a colored pixel. In particular have been defined different colors for different Tariffs. Tariff 4 (that means highest price for the electricity) is red colored, Tariff 3 is orange colored, Tariff 2 is yellow colored and finally Tariff 1 is green colored.

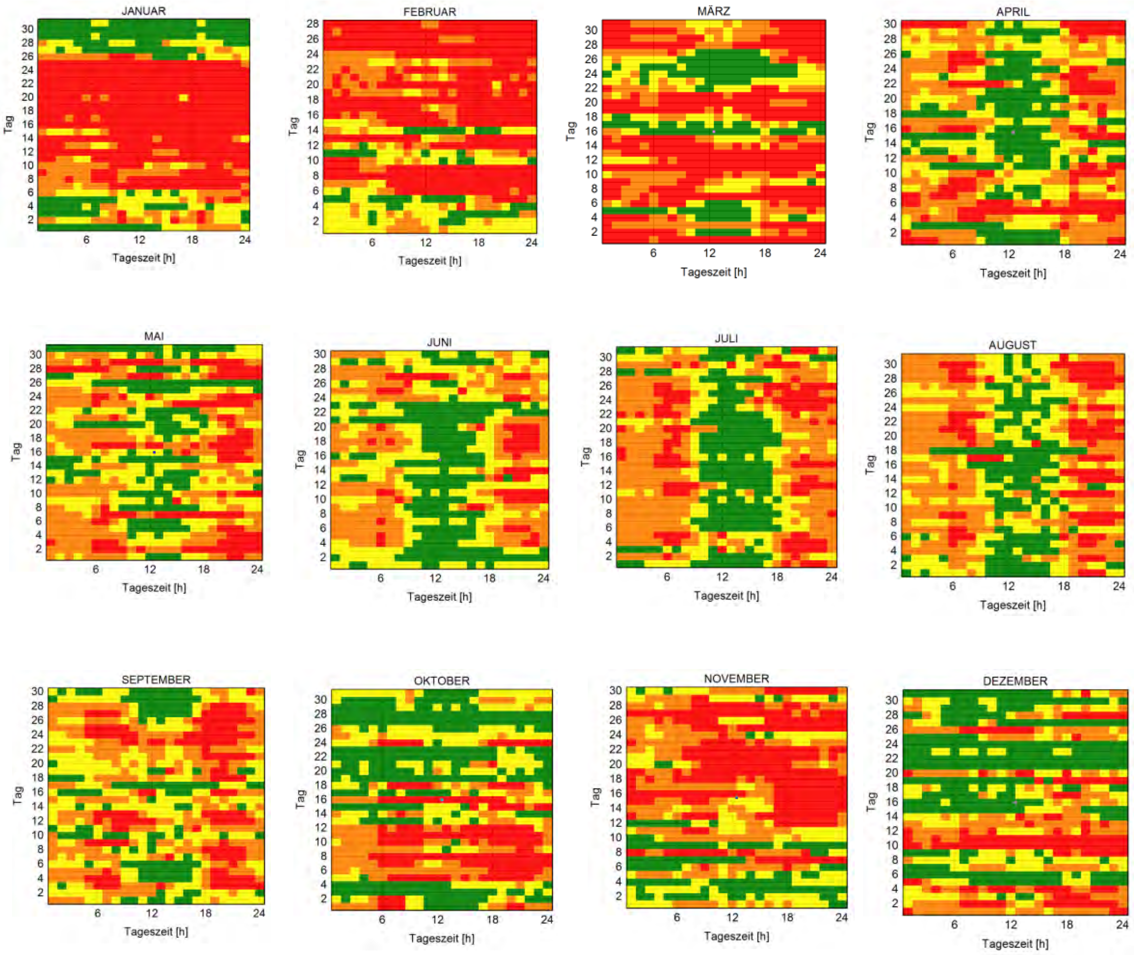

Figure 6 Tariff distribution during the year (Yu, 2015)

The Figure 6 (Yu, 2015) enables to see that the Tariff distribution during the whole year is strictly connected to the renewable energy production. In fact for example for most of January and February the electrical Tariff is 4 because the renewable production from wind, solar and biomass is not able to cover the electrical demand. So in this period that corresponds also to the coldest period of the year Demand Side Management has to be applied in order to exploit as much as possible the time intervals characterized by lower values for the Tariff.

Basically the goal is to move the electrical consumption of the heat pump to period with negative residual load and analyze the load shifting phenomena in the building.

Obviously this process doesn’t have to provoke any problem in terms of thermal comfort.

3 BUILDING MODEL

The building considered in this work is a typical single family house situated in a small German city. This building respects all the values imposed by the German Standard EnEV 2009. In fact this standard sets out thermal insulation values that have to be observed both in new buildings and property renovation in order to save energy costs regarding the space heating and to reduce Greenhouse effect due to CO_2 gas emissions (EnEV, 2009). Thermal insulation applied in the construction industry is an important factor for achieving also thermal comfort for building's occupants. The main aim of this Standard is to reduce the overall use of energy in the buildings combining the thermal insulation ordinance with the heating systems ordinance.

Basically the building evaluated is a single family house that has three floors.

The building model for the simulation study is explained below:

- Underground there is the basement. This indoor environment will be used for the installation of the main components of the heating system as heat pump, buffer storage tank and domestic hot water tank;
- In the first floor there are 7 rooms, that is Living room, Guest room, WC, Corridor 1, Kitchen, Air trap and Utility room 1;
- In the second floor there are 7 rooms too, that is Parent's room, Child 1 room, Utility room 2, Corridor 2, Work, Child 2 room and Bathroom;
- In the highest part of the building there is the attic;

Both basement and attic are considered unheated spaces, so there isn't need to evaluate heat losses for these two internal environments and there won't be installed any kind of heat distribution systems. So low temperature radiators will be installed only inside the rooms pertaining to first and second floor.

Figure 7 and Figure 8 show the plan of first and second floor, which are the indoor environments that must be heated during the heating period that in this case lasts from September to April according to the local Standards.

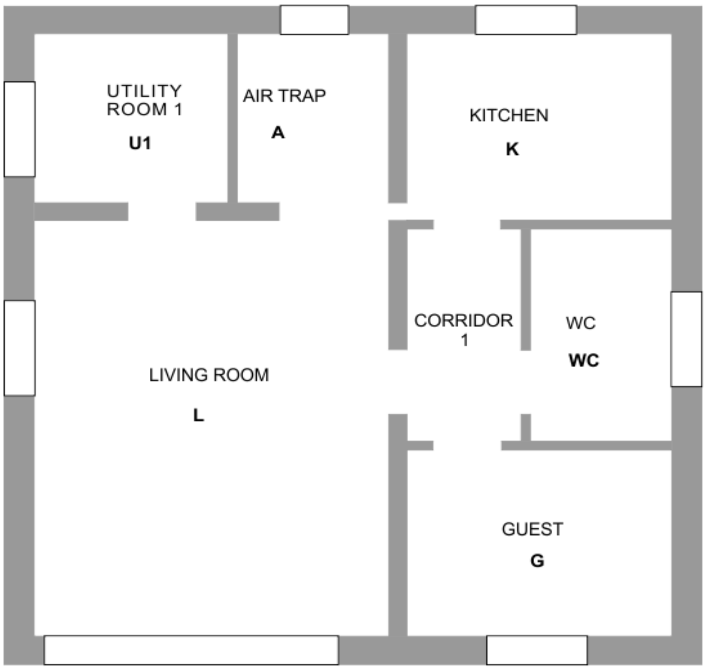

Figure 7 First floor plan

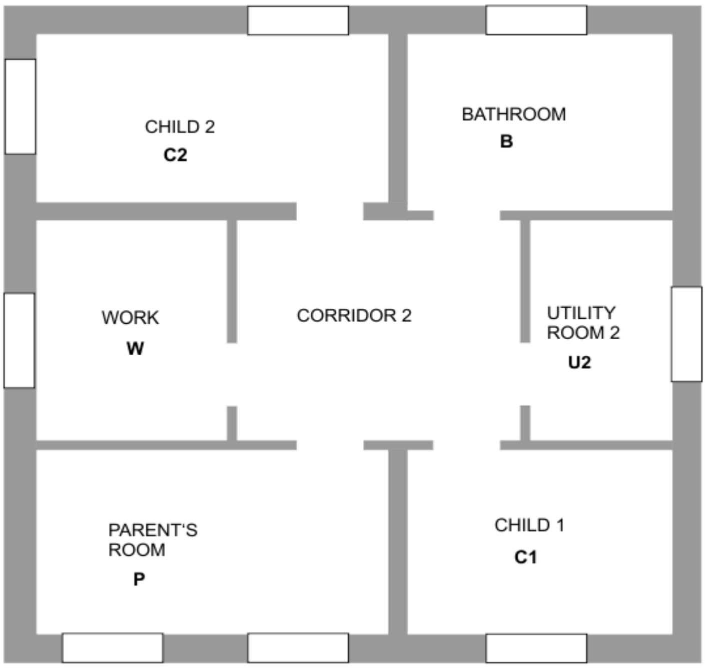

Figure 8 Second floor plan

In Table 1 are summarized the main geometrical properties of each room, in particular floor area, glazed area and orientation.

These characteristics are important to describe the thermal behavior of the building, since determine both the thermal losses through the windows and the solar radiation gain entering the room.

For this reason the glazed surfaces that are exposed to the South orientation on the one hand produce higher values for the solar gain during the spring and the autumn, on the other hand provoke higher thermal losses during the winter period.
That will be clearer introducing the thermal properties for windows.

Room	Floor area (m^2)	Glazed area (m^2)	Glazed area depending on the orientation (m^2)
Living room	33,51	10,91	2,15 West 8,76 South
Guest room	11,24	2,15	2,15 South
WC	7,17	2,15	2,15 East
Utility room 1	7,48	2,15	2,15 West
Air trap	5,48	-	-
Kitchen	10,28	2,15	2,15 North
Corridor 1	5,75	-	-
Parent's room	15,00	4,30	4,30 South
Child 1 room	11,24	2,15	2,15 South
Work	9,72	2,15	2,15 West
Utility room 2	7,17	2,15	2,15 East
Child 2 room	14,31	4,30	2,15 North 2,15 West
Bathroom	10,28	2,15	2,15 North
Corridor 2	14,31	-	-

Table 1 Geometrical properties of the building

3.1 Building envelope and thermal properties

This paragraph is useful to illustrate thermal properties regarding the main structures of the building such as external walls, internal walls, slabs etc. In fact in the following charts are shown the stratigraphy layouts of the building structures with information about their thermal properties as conductivity, capacity and density.
The thermal conductivity is the material's ability to conduct heat and can be defined as the amount of heat transmitted through a unit thickness of a material due to a unit temperature gradient under steady state conditions (Rossi, 2013). In the International System of Units its unit of measure would be W/mK but since in the Simulation is used TRNSYS Simulation tool its value is shown on the basis of the American System of Units. It takes low values for insulation materials if they are compared to normal construction materials such as bricks, plasters, concrete etc. On the other hand the capacity defines the amount of heat needed to raise the temperature of one kg mass 1 degree Celsius (Rossi, 2013). For most of the building materials its value is around 1 kJ/kgK. In closing density is defined as the building material mass per unit volume. Its unit of measure is kg/m^3.

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Layer A	0,015	2,52	1	1400
Layer B	0,175	3,564	1	1600
Insulation	0,167	0,173	1	60
Layer C	0,020	0,468	2,1	600

Table 2 External wall stratigraphy

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Plaster	0,010	1,26	1	1200
Ventilating brick	0,250	1,19	1	800
Plaster	0,010	1,26	1	1200

Table 3 Internal wall thick stratigraphy

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Plaster	0,010	1,26	1	1200
Ventilating brick	0,120	1,19	1	800
Plaster	0,010	1,26	1	1200

Table 4 Internal wall thin stratigraphy

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Floor pavement	0,050	5,04	1	2000
Insulation	0,080	0,126	0,84	80
Concrete	0,150	7,56	1	2400

Table 5 Slab on grade stratigraphy

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Wooden panel	0,010	0,36	1,7	300
Floor pavement	0,050	5,04	1	2000
Sound insulation	0,015	0,126	0,84	80
Concrete	0,220	7,56	1	2400
Insulation	0,090	0,144	1,03	80
Lime cement	0,010	3,13	0,869	1800

Table 6 First floor slab stratigraphy

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Wooden panel	0,010	0,36	1,7	300
Floor pavement	0,050	5,04	1	2000
Sound insulation	0,015	0,126	0,84	80
Concrete	0,220	7,56	1	2400
Plaster	0,010	1,26	1	1200

Table 7 Second floor slab *stratigraphy*

Layer	Thickness (m)	Conductivity (kJ/hmK)	Capacity (kJ/kgK)	Density (kg/m^3)
Wooden panel	0,010	0,36	1,7	300
Insulation	0,018	0,144	1,03	80
Concrete	0,220	7,56	1	2400
Plaster	0,010	1,26	1	1200

Table 8 Attic floor slab stratigraphy

All these information are useful to calculate one of the most important parameter that influences the thermal losses of a building that is the thermal transmittance.
On the basis of the European Standard EN ISO 6946 (UNIENISO-6946, 2008) has been considered this parameter, also known as U-value, which is by definition the amount of heat flow in watts per square meter of construction due to 1 Kelvin of temperature gradient (Rossi, 2013). So its unit of measure is W/m^{2}K. It takes into account and incorporates the thermal conductance of a structure and the heat transfer due to convection and radiation.

$$U = \frac{1}{R_T}$$

Equation 1

$$R_T = R_{si} + \sum_i R_i + R_{se}$$

Equation 2

$$R_i = \frac{d}{\lambda}$$

Equation 3

Where:

- U Thermal transmittance measured in W/m^{2}K;
- R_T Total thermal resistance measured in m^{2}K/W;
- R_{si} Thermal resistance of internal surface measured in m^{2}K/W;
- R_i Thermal resistance of internal surface measured in m^{2}K/W;
- R_{se} Thermal resistance of external surface measured in m^{2}K/W;
- d Thickness of the layer measured in m;
- λ Thermal conductivity of the layer measured in W/mK.

The values of R_{si} and R_{se} take into account the surface heat flux due to convection and radiation. Their values also depend on the heat flow direction as shown in the following chart (Table 9) taken from the European Standard EN ISO 6946 (UNIENISO-6946, 2008).

Thermal resistance (m^2K/W)	Heat flow direction		
	Ascending	Horizontal	Descending
R_{si}	0,10	0,13	0,17
R_{se}	0,04	0,04	0,04

Table 9 Thermal resistance of internal and external surface (UNIENISO-6946, 2008)

In accordance with the previous equations the thermal transmittance of each building structure has been calculated. These results are highlighted in Table 10.

Building structure	Thermal transmittance (W/m^2K)
External wall	0,275
Internal wall thin	1,694
Internal wall thick	1,017
Slab on grade	0,390
First floor slab	0,323
Second floor slab	1,153
Attic floor slab	0,204

Table 10 Thermal transmittances of walls and slabs

Table 10 permits to show that the lowest value, as is to be expected, is for the external wall because of its thermal insulation. In fact the external walls are a primary cause of overall thermal losses through the building envelope. Furthermore the building is characterized by different values for the thermal transmittance of the slab depending on the slab considered. In fact it's essential to obtain lower value for attic floor slab and first floor slab than second floor slab because they border two unheated spaces. In fact first floor is adjacent to the basement but the second floor is adjacent to the attic.

3.2 Windows and thermal properties

In the thermal behavior of a building also windows play an important role both considering the thermal losses and the solar heat gain entering the building.
Thereby it's fundamental to describe the two main parameters that influence on one hand the thermal losses of the windows (U-Value), on the other hand the total solar heat transmittance through the glazed surface.
The U-value has been already defined and the difference in its calculation as opposed to the opaque structure is that it's necessary to take into account of the thermal resistance of air

cavity; more information about its calculation can be found in the national standards (UNIENISO-10077-1, 2007). In the building considered the U-value for the windows is 1,43 W/m^{2}K.

The second parameter that defines the behavior of the building in terms of solar heat gain is the g-value (De Carli, 2013). In particular it describes the solar energy transmittance through windows. It can take values between 0 and 1; 1 means that the maximum amount of solar energy is passing through it, on the other hand 0 means that there is no solar energy transmittance through the glass. Hence it permits to estimate the solar heat gain into the space during sunny conditions. In cold climates solar heat gain can be a benefit during the winter season because it reduces the need for space heating. For this reason windows should be designed, sized and positioned for increasing the solar heat gain during the heating period. In the building considered, as shown via previous tables, the highest glazed surfaces are placed in the South orientation such as in living room and parent's room.

But then during the spring, summer and autumn season the solar heat gain can provoke phenomena of overheating because of its higher values than the winter period. This phenomenon will be also confirmed with results of this study in terms of thermal comfort for the building (Chapter 11).

The total solar heat gain through the window and entering the room is, by definition, the solar heat transmitted through the material directly plus the solar heat absorbed by the window and the frame and then re-emitted into the internal environment via convection and infrared radiation. So there is the need to use a parameter that permits to compare different kinds of windows; this parameter is the g-value and is defined in Equation 4 as the ratio between the total solar heat gain and the incident solar radiation (De Carli, 2013).

$$g = \frac{I_t + c \cdot I_a}{I}$$

Equation 4

Where:

- I_t Solar radiation transmitted through the window;
- c Fraction of the energy absorbed by the glass which is transferred in the room;
- I_a Solar radiation absorbed by the window;
- I Solar radiation impinging the window.

3.3 TRNSYS 3D and TRNBUILD

In order to investigate the Demand Side Management impact on heat pumps TRNSYS has been used; it is a simulation environment for the transient simulation of thermal systems and allows user to investigate the thermal behavior of multi zone buildings and validate new control strategies applied to heat generators.

In particular for defining the building model two programs have been utilized, that is Google SketchUp and TRNBuild. The first program has been used in order to define the three-dimensional geometric information of the building; in fact that is required for carrying out simulation which consider detailed radiation calculation (Klein, 2014).

Google SketchUp allows user to define only the geometrical shape of the building.

The most important thing before creating the building model is to have the energy model in mind because each zone created in this program corresponds to a thermal zone in the TRNBuild model. Since the purpose is to analyze also the thermal comfort of the building in Google SketchUp have been made 16 thermal zones, in other words 14 rooms for first and second floor and 2 zones more for attic and basement. Each thermal zone has been defined as a parallelepiped; its external surface includes external walls, internal walls, slabs, doors and windows.

Figure 9 shows the geometrical shape of the single family house which will be used during the simulations.

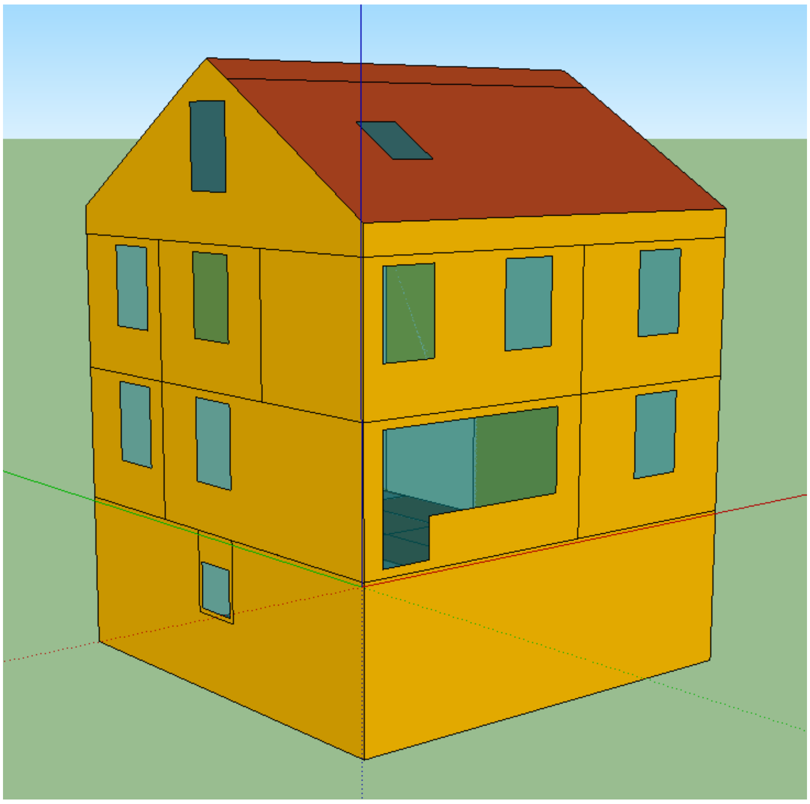

Figure 9 Tridimensional building model in Google SketchUp

Once the building's geometry has been planned, the IDF file has been imported in the TRNBuild software.

During the import the following steps have been performed:

- Partition of air-nodes and surfaces;
- Numeration of surfaces;
- Calculation of the volume;
- Generation of corresponding *_b17_IDF file that includes all the zones and the surfaces with the corresponding number.

Once the file has been imported to TRNBuild properties of the building model such as materials and constructions had to be put like input in the TRNBuild. Hence all the data introduced beforehand, such as thickness, conductivity, capacity and density, have been inserted in the program.

TRNBuild is a program that processes a file comprehending all the information about the building description and generates like output 2 files that will be used by the Type 56 component during a simulation.

This program has been developed to be user friendly for creating the building file.

3.4 Internal gains

Evaluating the thermal behavior and the energy demand of the building also internal gains have been taken into account because they provide a valuable source of heat contribution to space heating. Its value has been calculated according to the German Standard DIN V 18599 (DIN-V-18599, 2007) and considers the heat dissipated by persons, appliances, machinery, equipment and lightning in residential buildings. The design value suggested by the Standard is 50 Wh/m^{2}d. This value is related to the net floor area that may be assumed to be 1,1 times the heated living space.

4 HEATING LOAD CALCULATION

The heating load calculation has been done on the basis of the European Standard EN 12831 (UNIENISO-12831, 2006). This standard details how to calculate the energy need of a building in design conditions in order to achieve the design value for indoor temperature (Rossi, 2013).

This standard can be used in 2 different ways:

- If it is utilized for each room the result of the calculation is the design heat output of each distribution system that in this case is a low temperature radiator;
- On the other hand if it is used considering the entire building as one single thermal zone the result of the calculation is the design heat output of the heat generator that in this case is a heat pump.

In this work the first approach has been followed in order to size all the distribution systems that are installed in the single family house; at the same time the heat power of the heat pump will be evaluated as sum of the design heat output of each radiator.

This standard can be applied for buildings which have indoor spaces with height less than 5 meters and which are heated in design conditions. These type of buildings are for example residential buildings, offices, schools, libraries, hospitals etc (Rossi, 2013).

The fundamental hypothesis, on which is based this standard, are the following:

- Uniform temperature in each environment (air temperature corresponds to design temperature);
- Heat losses calculated in steady state conditions supposing constant values for temperatures, properties of building's structures;
- Solar and internal heat gains' contribute is not considered in this steady state model (protective condition);
- Air temperature coincides with operative temperature;
- Heated spaces are warmed up at the design indoor temperature;

The implementation of this method is founded on the identification of thermal losses from the building envelope to the external environment:

- First of all the thermal conduction through the building envelope and the heat losses from indoor environment to unheated zones have to be considered;
- After that the ventilation losses have to be calculated; they are the thermal losses due to ventilation or infiltration of air through the building envelope.

The steps of this method applied in the current project are shown below:

1) Identification of external design temperature depending on the locality;
2) Identification of heated and unheated spaces; in the first ones the design value for the indoor temperature must be maintained;
3) Within this step it's necessary to collect all the information about the building such as for instance:
   - ✓ V_i Air volume of each heated space (i), measured in m^3;
   - ✓ A_K Area of each building element (k), measured in m^2;
   - ✓ U_K Thermal transmittance of each building element (k), measured in W/m^2K;

The total design heat loss Φ_i for a heated space (i) is calculated as follow (Equation 5):

$$\Phi_i = \Phi_{T,i} + \Phi_{V,i}$$

Equation 5

Where:

- $\Phi_{T,i}$ Design transmission heat loss for heated space (i), measured in W;
- $\Phi_{V,i}$ Design ventilation heat loss for heated space (i), measured in W;

The first term is calculated as follow in Equation 6:

$$\Phi_{T,i} = \left(H_{T,ie} + H_{T,iue}\right) \cdot \left(\theta_{int,i} - \theta_{ext}\right)$$

Equation 6

Where:

- $H_{T,ie}$ Transmission heat loss coefficient from heated space (i) to the exterior (e) through the building envelope, measured in W/K;
- $H_{T,iue}$ Transmission heat loss coefficient from heated space (i) to the exterior (e) through unheated space (u), measured in W/K;
- $\Theta_{int,i}$ Internal design temperature of heated space (i), measured in °C. Its value depends on the room considered. It has been taken as 20°C in all of the rooms with the following exceptions: 24°C for the Bathroom and 15°C for the Utility rooms;
- Θ_{ext} External design temperature, measured in °C.

The first parameter of the previous correlation is calculated as (Equation 7):

$$H_{T,ie} = \sum_k A_K \cdot U_K$$

Equation 7

Where:

- A_K Area of building element (k) that separates heated space from the external environment (e), measured in m^2;
- U_K Thermal transmittance of the building element (k) calculated according to the European Standards UNIENISO 6946 for opaque elements (UNIENISO-6946, 2008) and UNIENISO 10077-1 for doors and windows (UNIENISO-10077-1, 2007); it is measured in W/m^{2}K;

Instead the second parameter is calculated as (Equation 8):

$$H_{T,iue} = \sum_k A_K \cdot U_K \cdot b_u$$

Equation 8

Where:

- A_K Area of the building element (k) that separates heated space (i) from unheated space (u), measured in m^2;
- U_K Thermal transmittance of the building element (k), measured in W/m^{2}K;
- b_U Temperature reduction factor. This parameter takes into account the difference between temperature of unheated space and external design temperature. Its value has been found in the Standard and is shown in the following chart (Table 11):

Unheated space	b_U
Basement with windows/external doors	0,8
Roof space (with insulation)	0,7

Table 11 Temperature reduction factor (UNIENISO-12831, 2006)

The design ventilation loss for a heated space (i) is calculated as follow (Equation 9):

$$\Phi_{V,i} = V_i \cdot \rho \cdot c_p \cdot (\theta_{int,i} - \theta_{ext})$$

Equation 9

Where:

- V_i Air flow rate of heated space (i), measured in m^3/s;
- ρ Density of air at $\Theta_{int,i}$ measured in kg/m^3;
- c_p Specific heat capacity of air at $\Theta_{int,I}$ measured in kJ/kgK.

The procedure for calculating the air flow rate V_i depends upon the case considered, i.e. with or without ventilation system (Rossi, 2013).

In the case of buildings devoid of ventilation system the ventilation flow rate equals volume flow rate due to the infiltration. On the other hand in a building with mechanical ventilation system the air flow rate is the sum between the flow rate due to the infiltration and the flow rate led into the building for improving the air quality. In this case the fresh air flow doesn't have necessarily the same characteristic of the external air. For example in the winter period its temperature will be higher than the external temperature if in the ventilation system is used a recovery heater that warms up external air to the detriment exhaust air flow. In the case under examination is installed a ventilation system with heat recovery. More details regarding the ventilation system will be shown in the chapter dedicated to the ventilation (Chapter 9).

Table 12 enables to explain the total design heat loss calculated for each room according to the guidelines of the European Standard EN 12831. Moreover this table will be used in order to size each low temperature radiator. The total heating load of the building, calculated as sum of heating loads of each room, is 5,33 kW.

Room	Design heat loss (W)
Living room	1230
Guest room	474
WC	322
Utility room 1	208
Air trap	140
Kitchen	432
Parent's room	608
Child 1 room	416
Work	292
Utility room 2	196
Child 2 room	462
Bathroom	551

Table 12 Design heat loss for each room

5 HEAT PUMP

Heat pumps are devices that extract heat from the heat source and make it usable to the heat sink. Heat pumps can be exploited in the building sector both for the space cooling and the space heating of the building.

A familiar example of a heat pump that is utilized for cooling is a domestic refrigerator or freezer. Effectively a heat pump and a refrigerator are the same from the thermodynamic point of view, because for example a refrigerator catches heat from indoor air for cooling its internal volume. There are many other familiar examples such as air conditioning units, and recently this type of technology has been exploited also for space and water heating (Livingstone, 2010). Nowadays heat pump is one of the most effective solution in order to achieve simultaneously real energetic savings, decreasing of pollution and Greenhouse effect and also limit costs relative to the buildings' air conditioning. The increased efficiency of these devices in recent years has led to prefer this kind of systems to the traditional boilers. Another aspect that has to be highlighted is that there aren't limitations during the heat pump's operation because for instance air source heat pump can work with very low values of the external temperature as -15°C (Rossi, 2013).

Obviously that will involve a proportional decrease of its performance.

5.1 Thermodynamic theory

A thermodynamic cycle consists of a sequence of thermodynamic processes that involve heat transfers and work into and out of the boundaries of the system.

In this process temperatures, pressures and other state variables within the system vary.

The second law of thermodynamics states that it's impossible to operate a thermodynamic cycle where the only consequence is to move heat from a colder source to a hotter source. But if the cycle gets an energy input this transfer of heat is possible.

If the cycle has the main goal to transfer heat from a colder body to a warmer body it is called heat pump. On the other hand if the desired effect is to remove heat from an environment that must be kept at a lower temperature then the surrounding the cycle is called refrigeration cycle. In both cases the operation of the device is the same: procuring mechanical work is possible to remove heat from an ambient with lower temperature and transfer it, increased also of the thermal equivalent of the mechanical work, to an ambient with higher temperature (Rossi, 2013).

Nowadays exist two types of heat pumps (vapor compression and absorption) but only the vapor compression heat pump will be considered. Schematically the main components in a vapor compression heat pump are the evaporator, the compressor, the condenser and the expansion valve. They are illustrated in Figure 10.

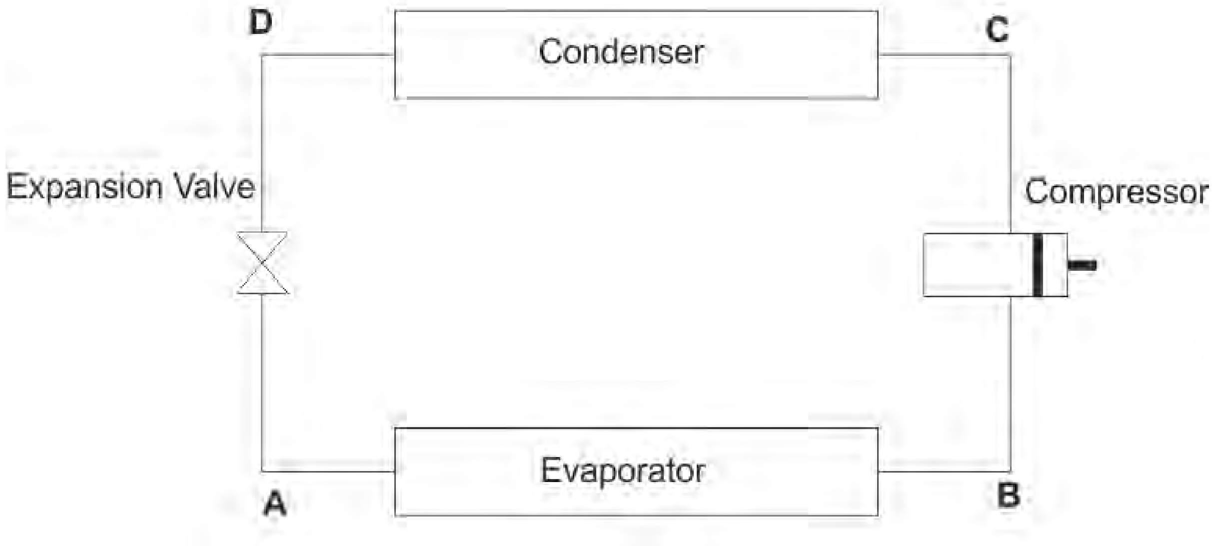

Figure 10 Main components of a heat pump

Figure 11 (Livingstone, 2010) enables to show the thermodynamic cycle of the heat pump in the diagram Temperature-Entropy.

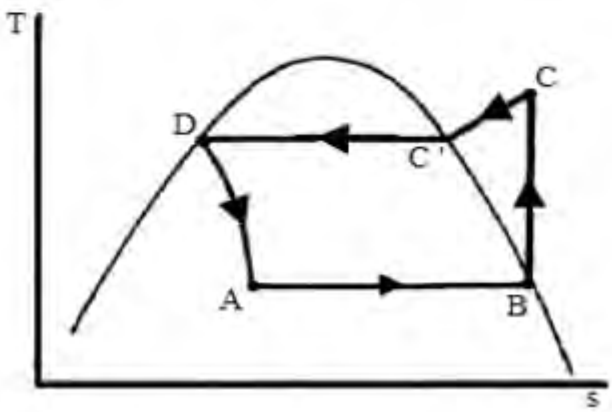

Figure 11 Heat pump thermodynamic cycle (Livingstone, 2010)

The working fluid used in heat pump technology is a volatile evaporating and condensing fluid known as a refrigerant. It evaporates during the process AB keeping constant its values of temperature and pressure. Then it enters the compressor and the process BC is not isentropic. After that the working fluid is overheated vapor and it has to enter the condenser where before it cools down (CC') and then condensate (C'D) with a process that occurs with constant values for pressure and temperature. Finally in the process DA the refrigerant passes through the expansion valve and then the cycle starts again.

The condensation temperature T_D is higher than the hot source's temperature, as well the evaporation temperature T_A is lower than the cold source's temperature (Rossi, 2013).
In fact only in this way it's possible to realize the heat transfers.
The main parameter that better describes the heat pump's operation is the COP, in other words Coefficient Of Performance. It states the ratio between the heat delivered to the hotter source and the work concerning the compressor; in this work the heat pump behavior is shown only during its heating mode.
From the energetic point of view the use of the heat pump is surely advantageous if compared to the electrical heating, because by definition COP is higher than 1. COP is defined in the Equation 10 as:

$$COP = \frac{Q_{th}}{E_{hp}} = \frac{|Q_c| + E_{hp}}{E_{hp}}$$

Equation 10

Where:
- Q_{th} Thermal energy desired as thermal output (condenser), measured in kWh;
- E_{hp} Electrical consumption regarding the compressor, measured in kWh;
- Q_c Thermal energy removed from the cold reservoir (evaporator), measured in kWh.

The heat pump's COP is very sensitive to the minimal and maximal temperature of the cycle (Rossi, 2013). So it's better to keep in mind that COP values taken from heat pump manufacturers have to be checked in all the possible range of real operation.
In fact in the manufacturer's catalogue it's possible to find both the nominal value of the heat pumps performances (in specific conditions of temperatures) and charts with the same parameters extended to wider temperature ranges.
Actually, as will be clearer at a later stage, COP during the heating period reaches lower values than the nominal ones especially because of lower cold source temperatures.

5.2 Different Heat Sources

Throughout the previous paragraph it has been highlighted that the main purpose for a heat pump is to realize a heating effect absorbing heat from a cold source; but it's also important to check more in detail how the choice of this source can influence the seasonal performance of this device. An ideal heat source should have the following properties (Le Feuvre, 2007):
- "High" temperature and stable during the heating season;
- Abundant;
- Not polluted;

- Good thermal properties such as high value of specific heat per unit volume;
- Involve low costs and investment to exploit it.

There is a wide variety of different sources from which a heat pump can extract heat. The option selected depends on many factors such as local circumstances (economic incentives), the location of the building and its heat demand.
The most common heat sources used for heat pumps are air, water and the ground.
In this work will be analyzed the operating behavior of Ground Source and Air Source Heat Pumps because they are the two most used in Germany. In Table 13 the temperature range of the more important cold sources is shown.

Heat source	Temperature range (°C)
Ambient air	-10/-15
Exhaust air	15/25
Ground water	4/10
Lake water	0/10
River water	0/10
Sea water	3/8
Rock	0/5
Ground	0/10
Waste water and effluent	>10

Table 13 Heat sources and temperature range (Livingstone, 2010)

As mentioned beforehand the level of the cold source affects the heat pump performance. In fact a lower evaporation temperature (therefore also a lower pressure) causes the work increase of the compressor in order to reach the same value of condensation temperature; in this case the compressor has to work with a higher compression ratio (Livingstone, 2010).

5.3 Operating modes for heat pumps

The operating mode for heat pumps basically depends on the distribution system chosen for supplying heat to the building. For example if a temperature above the maximum supply temperature of the heat pump is required there will be the need to use the heat pump only as supplement to a conventional heat generator such as a boiler.
So when heat pumps are used for producing space heating and domestic hot water the following operating modes can be identified (Le Feuvre, 2007):
- Monovalent operation;
- Mono energy operation;
- Bivalent operation.

In the following list the difference between these operating modalities will be made clearer (Dimplex, 2015):

- In the Monovalent operation the heat pump is the only heat generator that has to cover the heat demand of the building in terms of space heating and energy need for domestic hot water. The heat distribution system is sized so that its maximum flow temperature is below the maximum supply temperature of the heat pump. At the same time there is the need to ensure that in design condition the heat capacity of the heat pump is larger than the heat demand. Brine to water and water to water heat pumps are normally operated in Monovalent mode;
- In the Mono energy mode it must be installed a second heat source, which is usually an electric booster heater, in order to cover the heat demand in extreme conditions. Usually this electric heater is activated depending on the external temperature or the heating load required. This is the case for instance of Air Water Heat Pumps that work far from their nominal conditions when the outside temperature is very low (-5°C to -20°C). Usually Air Water Heat Pumps, as recommended by heat pumps manufacturers, are designed to supply between 70% and 85% of the design power calculated in accordance with the European Standard EN 12831. In this way the heat pump can cover up to 95% of the annual heat demand (Viessmann, 2015);
- In the Bivalent operation the heat pump is combined with at least one other heat generator that for example can be a boiler. In this operation mode there is the possibility to make these two system work simultaneously or alternatively. In the first case the two devices can operate in parallel, in other words the heat pump is activated alongside the second generator and therefore it will cover a larger portion of heating. In the second case instead above the designated temperature the heat pump will provide the entire heating demand; but once the source temperature falls below this value the heat pump will be switched off and the second heater will take over. This solution is used when old buildings are renovated and the existing boiler is to be kept in use.

6 LOW TEMPERATURE RADIATOR

The most common heat terminal units used for heating residential housings, offices and factories are radiators. Radiators provide for heating indoor environments, in which they are placed, because of a combined effect of convection and radiation. Actually the greater part of the heat is exchanged by convection (70-80%): the ambient air, in contact with the hot surface of the radiator, gets warm and, because of its low density, goes up; that activates a convective movement that facilitates the heat exchange. Furthermore the radiant effect is minimal because the radiator's surface is small and its surface temperature is not so high (Rossi, 2013). The reference Standard that regulates radiator's behavior is the European Standard EN 442 (UNIEN-442, 2004). This Standard establishes the method for calculating the heat output of a radiator; this is evaluated considering 50°C of temperature's difference between the mean water temperature and the ambient temperature.

Since normally the ambient temperature is 20°C that means the heat output is calculated considering a mean water temperature of 70°C; for example this is the case of flow temperature 75°C and return temperature 65°C. All radiators manufacturers have to show the power in technical catalogs respecting the guidelines of EN 442.

The reduction of the operation temperature is useful because in this way it's possible to achieve simultaneously both economical savings and thermal comfort. In fact too high temperatures could provoke fast convective air movements that put in motion fine dust causing phenomena such as allergy or breathing apparatus' irritation. The decrease of the operation temperature causes benefits also as regards the air temperature stratification and the increase of overall wellbeing. Moreover it's very important to highlight that the reduction of flow temperature increases the possibility to operate with higher values of heat pump's efficiency (Rossi, 2013). The investment due to the bigger size of radiators (but operating with a lower temperature) is justified by the pay-back period that is much lower than radiator's lifetime.

Sizing a radiator is basically simple: the radiator has hot water flowing through it; the hot water is warmed up from the heat generator and moved from a pump having the appropriate values of mass flow rate and head. The sizing is made imposing the equality between the energy need of each room per unit time and the radiator's power output. This concept is explained by Equation 11 (Rossi, 2013):

$$E = \dot{Q}$$

Equation 11

Where:

- E Energy need per unit time, measured in W;
- Q Heat output of the radiator, measured in W.

Then imposing supply and return temperature depending on the device used for space heating it's possible to define the mass flow rate that passes through each radiator with the following equation (Equation 12):

$$\dot{m}_w = \frac{\dot{Q}}{c_w \cdot (t_{su} - t_{ex})}$$

Equation 12

Where:

- m_w Water mass flow rate, measured in kg/s;
- t_{su} Design flow temperature, measured in °C;
- t_{ex} Design return flow temperature, measured in °C;
- c_w Specific heat capacity of water, measured in J/kgK.

The heat output of a radiator can be expressed also as a function of the mean water temperature and ambient air temperature as shown in Equation 13 (ASHRAE, 2012):

$$\dot{Q} = K \cdot (t_{mw} - t_a)^n$$

Equation 13

$$t_{mw} = \frac{(t_{su} + t_{ex})}{2}$$

Equation 14

Where:

- K Constant of the radiator determined by test;
- t_{mw} Mean water temperature, measured in °C;
- t_a Ambient air temperature, measured in °C;
- n Exponent of the radiator determined by test (normally it is 1,25-1,35);

Actually the designer has to choose the radiator whose heat output has a higher value, even if only little, than the heat demand of the room per unit time calculated according on EN 12831 (Rossi, 2013). In this thesis, operating in combination with heat pump, radiators have

to work with lower supply temperatures than traditional radiators. Hence a reasonable choice is to use low temperature radiators with 45°C of design flow temperature and 40°C of design return flow temperature; for this reason have been taken into consideration VOGEL&NOOT panel radiators because their features make them compatible also with renewable sources such as heat pumps (Vogel&Noot, 2015).
The radiators chosen for the Simulation are shown in the following table (Figure 12).
This chart has been used for sizing each room's radiator according to the heat demand beforehand calculated and for deciding the characteristic exponent. Other information such as metal weight and water contents, that are very useful in order to consider the thermal inertia, have been found in the same catalogue. This parameter is important for radiators but will be introduced in at a later stage.

45/40/20° C

Side panels and top cover of COMPACT-, T6- and MULTI-FUNCTIONAL VALVE RADIATORS are taken into consideration in the performance data

Radiator power data in watts, in accordance with **DIN EN 442** supply temperature **45** - return temperature **40** - room temperature **20° C**

Height [mm]		300				
Length [mm]	Type / Power	10	11 K 11 KV 11 VM	21 K-S 21 KV-S 21 VM-S	22 K 22 KV 22 VM	33 K 33 KV 33 VM
400	Watt	50	78	116	152	216
520	Watt	66	102	151	197	280
600	Watt	76	117	174	227	324
720	Watt	91	141	209	273	388
800	Watt	101	156	232	303	432
920	Watt	116	180	267	349	496
1000	Watt	126	195	290	379	539
1120	Watt	141	219	325	424	604
1200	Watt	151	234	349	455	647
1320	Watt		258	383	500	712
1400	Watt		274	407	531	755
1600	Watt		313	465	606	863
1800	Watt		352	523	682	971
2000	Watt		391	581	758	1079
2200	Watt		430	639	834	1187
2400	Watt		469	697	910	1295
2600	Watt				985	1402
2800	Watt				1061	1510
3000	Watt				1137	1618
Radiatorexponent n		1,274	1,330	1,327	1,329	1,331

Height [mm]		400				
Length [mm]	Type / Power	10	11 K 11 KV 11 VM	21 K-S 21 KV-S 21 VM-S	22 K 22 KV 22 VM	33 K 33 KV 33 VM
400	Watt	64	97	144	184	262
520	Watt	83	126	188	240	341
600	Watt	96	145	216	276	393
720	Watt	115	175	260	332	472
800	Watt	128	194	289	369	524
920	Watt	147	223	332	424	603
1000	Watt	159	242	361	461	655
1120	Watt	179	272	404	516	734
1200	Watt	191	291	433	553	786
1320	Watt		320	476	608	865
1400	Watt		339	505	645	917
1600	Watt		388	577	737	1048
1800	Watt		436	649	829	1179
2000	Watt		485	722	922	1310
2200	Watt		533	794	1014	1441
2400	Watt		582	866	1106	
2600	Watt				1198	
2800	Watt				1290	
3000	Watt				1382	
Radiatorexponent n		1,283	1,342	1,334	1,353	1,357

Height [mm]		500				
Length [mm]	Type / Power	10	11 K 11 KV 11 VM	21 K-S 21 KV-S 21 VM-S	22 K 22 KV 22 VM	33 K 33 KV 33 VM
400	Watt	76	117	171	213	303
520	Watt	99	152	222	277	394
600	Watt	115	175	256	319	455
720	Watt	138	210	307	383	545
800	Watt	153	233	341	425	606
920	Watt	176	268	393	489	697
1000	Watt	191	291	427	532	758
1120	Watt	214	326	478	596	849
1200	Watt	229	350	512	638	909
1320	Watt	252	385	563	702	1000
1400	Watt	267	408	598	745	1061
1600	Watt	306	466	683	851	1212
1800	Watt	344	525	768	957	1364
2000	Watt	382	583	854	1064	1515
2200	Watt	420	641	939	1170	1667
2400	Watt	459	700	1024	1276	
2600	Watt		758	1110	1383	
2800	Watt			1195	1489	
3000	Watt			1280	1595	
Radiatorexponent n		1,292	1,330	1,323	1,334	1,351

Height [mm]		600				
Length [mm]	Type / Power	10	11 K 11 KV 11 VM	21 K-S 21 KV-S 21 VM-S	22 K 22 KV 22 VM	33 K 33 KV 33 VM
400	Watt	88	131	191	234	339
520	Watt	115	170	248	305	440
600	Watt	133	196	286	352	508
720	Watt	159	236	343	422	609
800	Watt	177	262	381	469	677
920	Watt	204	301	439	539	779
1000	Watt	221	327	477	586	846
1120	Watt	248	367	534	656	948
1200	Watt	265	393	572	703	1016
1320	Watt	292	432	629	774	1117
1400	Watt	310	458	667	821	1185
1600	Watt	354	524	763	938	1354
1800	Watt	398	589	858	1055	1523
2000	Watt	442	655	953	1172	1693
2200	Watt	487	720	1049	1289	1862
2400	Watt	531	786	1144	1407	
2600	Watt		851	1239	1524	
2800	Watt			1335	1641	
3000	Watt			1430	1758	
Radiatorexponent n		1,301	1,319	1,310	1,343	1,333

Height [mm]		900				
Length [mm]	Type / Power	10	11 K 11 KV 11 VM	21 K-S 21 KV-S 21 VM-S	22 K 22 KV 22 VM	33 K 33 KV 33 VM
400	Watt	124	178	260	315	437
520	Watt	161	232	338	410	568
600	Watt	186	268	390	473	655
720	Watt	223	321	467	567	786
800	Watt	248	357	519	630	874
920	Watt	285	410	597	725	1005
1000	Watt	310	446	649	788	1092
1120	Watt	347	500	727	882	1223
1200	Watt	372	535	779	945	1311
1320	Watt	409	589	857	1040	1442
1400	Watt	434	625	909	1103	1529
1600	Watt		714	1039	1260	1748
1800	Watt		803	1169	1418	1966
2000	Watt		892	1299	1575	2184
2200	Watt					
2400	Watt					
2600	Watt					
2800	Watt					
3000	Watt					
Radiatorexponent n		1,305	1,332	1,321	1,340	1,354

Type programme: COMPACT RADIATOR; T6-CENTRALLY CONNECTED RADIATOR and MULTI-FUNCTIONAL VALVE RADIATOR

The availability of any type of radiator, as well as range of sizes, is in accordance with the production programme, as stated in the price list.

Figure 12 Manufacturer's catalogue for radiators (Vogel&Noot, 2015)

7 HEATING SYSTEM WITH A GROUND SOURCE HEAT PUMP

This chapter shows how has been built up the simulation model that simulates the technical performance of a Ground Source Heat Pump. This heat pump covers the energy need of the building including space heating and domestic hot water demand. The simulation of this kind of air conditioning plant has been carried out using Simulation Studio that is a complete simulation package of TRNSYS.

It allows user to design a heating system and to simulate its behavior.

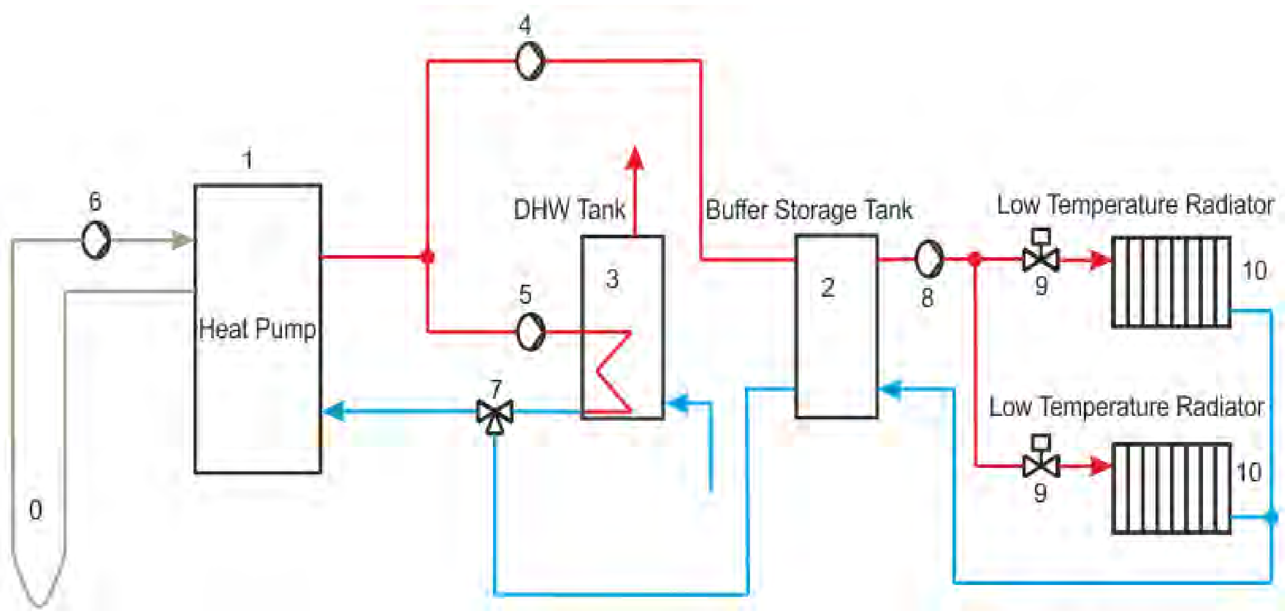

Figure 13 Ground Source Heat Pump

In Figure 13 the scheme of the system with Ground Source Heat Pump is shown considering all its main components. Figure 13 enables to identify the following components:

- 0 represents the borehole containing the ground heat exchanger that can capture heat from the ground source and transfer it into the building;
- 1 is the water to water heat pump with 6,2 kW of rated heating output and 1,38 kW of power consumption;
- 2 is the buffer storage tank with a capacity of 1000 liters. It stores hot water supplied by the heat pump and makes it available for the distribution heating system;
- 3 is the domestic hot water tank with a capacity of 300 liters. It contains a heat exchanger that allows to warm up the cold water originating from the aqueduct in order to satisfy the domestic hot water consumption of the building;
- 4 is the secondary pump that is used to move the water from the condenser of the heat pump to the buffer storage tank. This pump consumes 70 W of power;

- 5 is the DHW pump that is utilized to move the water from the condenser of the heat pump to the domestic hot water tank. This pump consumes 70 W of power too;
- 6 is the primary pump that is used to move the brine-water from the ground heat exchanger to the evaporator of the heat pump. This pump instead consumes 100 W of power;
- 7 is the 3-way valve that conducts water coming from the buffer storage tank or DHW tank to the heat pump;
- 8 is the heating circulation pump that is utilized to move the water from the buffer storage tank to the radiators installed in each room;
- 9 is the thermostatic valve that is used to control room temperature in water heating distribution systems as radiators;
- 10 is the low temperature radiator chosen as distribution heating system.

All these components will be defined in the following paragraphs showing both information taken from technical brochures of air conditioning manufacturers and also main equations that are used by Simulation Studio for simulating their behavior.

7.1 Ground Source Heat Pump

The heat pump chosen for covering the heat demand of the building and also the domestic hot water consumption is a Viessmann product that works in the Monovalent operation mode. For this reason it has been sized after evaluating the heat demand of the building that is 5,3 kW and the power that is necessary to produce domestic hot water. For conventional single family house it has been supposed to have a maximum value of daily domestic hot water need equal to 50 liters/person considering 45 °C as design temperature (Viessmann, 2015). In Germany it's possible to consider this low set point temperature because exists a German Standard that allows this design choice for domestic hot water production in single family house. The same would be not allowed in countries such as Italy where national Standards impose higher values because of the problem with Legionella bacterium. In this specific case the number of the people living in the single family house is 4 and this means that the heat pump has to produce on average 200 liters/day as regards the domestic hot water. Heat pumps manufacturers advise to consider an additional thermal load of 0,25 kW for each person only if the new overall heat power exceeds 20% of the heating load calculated on the basis of EN 12831 (Viessmann, 2015). The total thermal load including space heating and domestic hot water would be 6,3 kW. Hence in this case it's not been necessary to consider this supplementary load because this percentage is 18% and therefore a heat pump that provides 5,3 kW in the operating conditions is enough.

When sizing a Ground Source Heat Pump provision should be made for the temperature of the cold and hot source because both of them influence the heat output. For this reason the result of a lot of measurements carried out in the “Heat Pump Efficiency Project” by the Fraunhofer Institute have been checked (Marek Miara, 2011). Initially great attention has been paid when defining the cold source temperature because it influences the rated heating output of the heat pump. For instance in Figure 14 are shown the real values of the brine-water temperature entering and exiting the heat pump in the months of December and June for the years from 2010 to 2013.

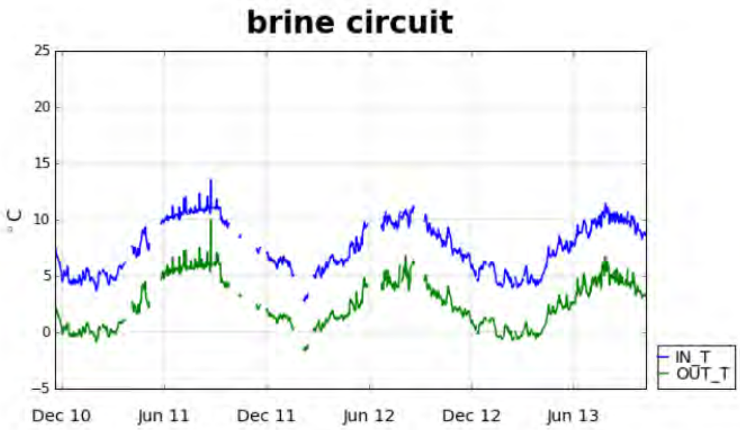

Figure 14 Measurements of brine-water temperature (Marek Miara, 2011)

Figure 14 enables to illustrate that the minimal value for the temperature of brine-water entering the heat pump after the heat exchange with the ground is approximately 5° C.
Hence the heat pump must cover the heat demand required by the building when the brine-water temperature is approximately 5°C, or rather during the winter period during which the building demands the maximum value of heat from the heat pump. At the same time it’s very important also the hot source’s temperature because the heat pump works in better conditions when supplies water at “low” temperatures. For this building has been selected the type Vitocal 300 G produced by Viessmann. Heat pumps’ manufacturers have to show in the technical catalogues the technical performances of their devices basing on the European Standard EN 14511 (DINEN-14511, 2012). That means that they have to illustrate the heating output, the cooling output (in this case it’s not considered the heat pump’s operation in the cooling mode) and the electrical consumption depending on both the brine-water temperature entering the heat pump and the water temperature exiting to the load side (for domestic hot water production or space heating). In the following pictures (Figure 15 and Figure 16) are shown the performances of Vitocal 300 G (Viessmann, 2015):

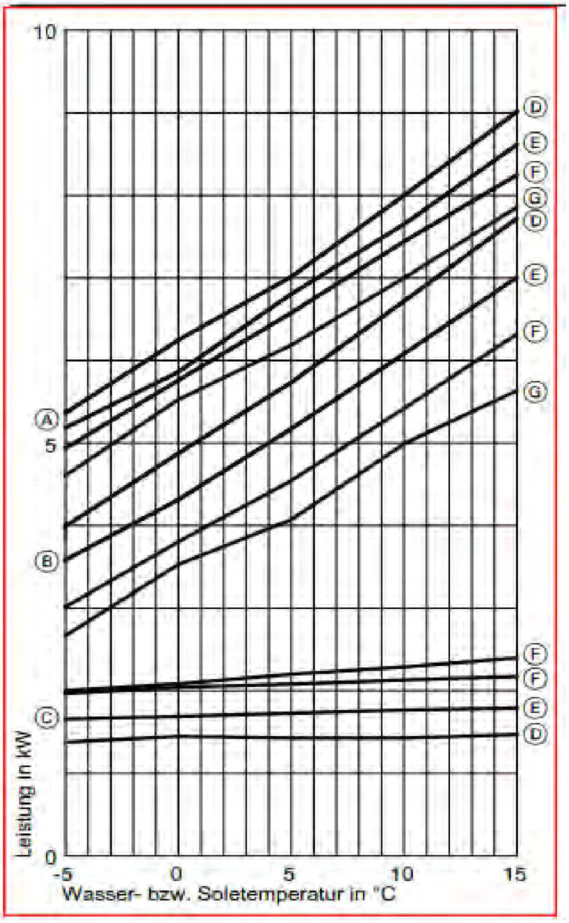

Figure 15 Vitocal 300 G powers (Viessmann, 2015)

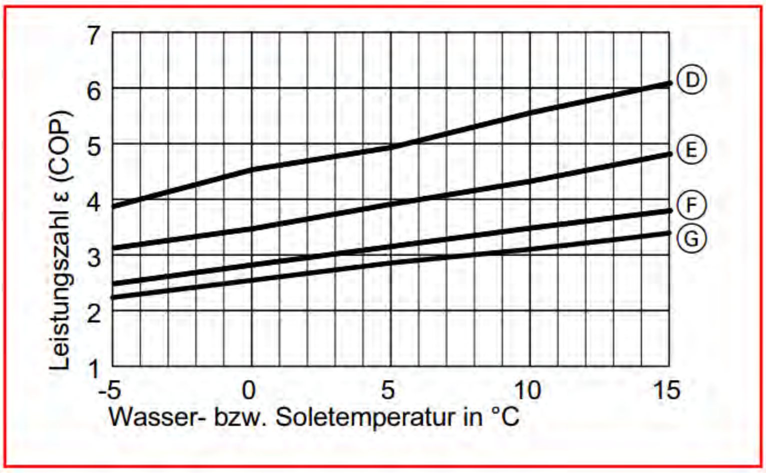

Figure 16 Vitocal 300 G COP (Viessmann, 2015)

More explanations of the previous pictures regarding the heat pump's performances may be found in the following list:

- The X-axis in both pictures represents the inlet temperature to the heat pump of brine-water fluid, expressed as °C;
- The Y-axis in Figure 15 is a power (expressed in kW), on the other hand in Figure 16 is depicted the COP (dimensionless);
- A represents the heating output that heat pump delivered to the load side, expressed in kW;
- B represents the cooling output, expressed in kW;
- C represents the electrical consumption of the heat pump in the specified operating mode, expressed in kW;
- D is the line that describes heat pump's previous parameters (heating output, electrical consumption or COP) when the temperature of the produced water is 35°C;
- E is referred to a hot water temperature of 45°C;
- F represents the line related to a hot water temperature of 55°C;
- G finally is referred to a produced water temperature of 65°C.

Since the heat distribution systems that have been designed for supplying heat to the building are low temperature radiators (maximum supply temperature of 45°C) the heat pump is supposed to work with heating flow temperature lower than 55°C during the space heating mode. On the other hand the heat pump will work with higher values for the flow water temperature when it will be used for providing domestic hot water. Furthermore Figure 15 enables to show how the heat pump chosen in the design condition (5°C for the brine-water temperature and 55°C for the exiting water temperature) can guarantee more than the heat power required by the system. In these conditions it can furnish 6,4 kW of nominal heat power.

7.1.1 Water to water heat pump model in TRNSYS

The type used in Simulation Studio for simulating a water to water heat pump is Type 927 (Thornton, 2014). The behavior of this type is basically influenced by external files that have been put as input by the user. In fact at the heart of this component there are two external files, one containing heat pump's performance related to the heating mode and another one related to the cooling mode; this latter will not be considered. These files contain performance such as the heat capacity and the electrical consumption of the heat pump depending on the entering load and source temperature. In these external files data connected to several operating modes have been introduced, but since the heat pump will work also in other operating points this type is able to linearly interpolate and find the right operating point. The main difference with the performance data furnished by heat pump manufacturers is that there is the need to give as input the inlet load temperature, in other words the water temperature returning from the building. In fact, as explained before, usually manufacturers give device's output based on the flow water temperature.

For this reason it has been necessary to assume a temperature difference in the load side of the heat pump and then check singularly the heat pump performances comparing these results with the brochure data. Moreover heat pump manufacturers recommend to use a higher mass flow rate for the secondary pump than the heating circulation pump (Viessmann, 2015); the mass flow rate of the circulation pump is basically determined as the sum of the design mass flow rate calculated for each radiator.

Since this ground source heat pump will work with different load water flow rate (1000 kg/h in the space heating mode and 530 kg/h in the domestic hot water production) it has been necessary to build up the external file with a compromise between these two different modalities. In fact, on equal terms in the source side, the heat pump will provoke different temperature increases in the load side.

Hence considering that the mass flow rate used in DHW application is around half of the mass flow rate for providing the buffer storage tank, the heat pump will be affected by a double increase of temperature in the load side. Because of that it has been necessary to consider circa 5°C of load difference temperature in the temperature range of the heating mode and admit circa 10°C in the temperature range of domestic hot water production. In Figure 17 is illustrated partially the external file used in the simulation containing the fraction of heat power and electricity consumption depending on the load and source entering temperatures. In particular these relative values are referred to the nominal value of the heat pump producing hot water at 35°C with brine-water entering at 0°C.

```
HPexternalfile_new3.dat - Blocco note
File  Modifica  Formato  Visualizza  ?
1.000   1.000   1.000   !Normalized     Load    Flow    Rates
1.000   1.000   1.000   !Normalized     Source  Flow    Rates
30.000  40.000  45.000  55.000  !Entering       Load    Temperatures    (C)
-5.000  0.000   5.000   10.000  15.000  !Entering       Source  Temperatures    (C)
0.871   1.000   !Total  Heating(kW)and  Power(kW)       at      35/-5
1.008   1.051   !Total  Heating(kW)and  Power(kW)       at      35/0
1.134   1.025   !Total  Heating(kW)and  Power(kW)       at      35/5
1.290   1.029   !Total  Heating(kW)and  Power(kW)       at      35/10
1.452   1.065   !Total  Heating(kW)and  Power(kW)       at      35/15
0.839   1.203   !Total  Heating(kW)and  Power(kW)       at      45/-5
0.948   1.217   !Total  Heating(kW)and  Power(kW)       at      45/0
1.100   1.246   !Total  Heating(kW)and  Power(kW)       at      45/5
1.234   1.275   !Total  Heating(kW)and  Power(kW)       at      45/10
1.387   1.290   !Total  Heating(kW)and  Power(kW)       at      45/15
0.798   1.435   !Total  Heating(kW)and  Power(kW)       at      55/-5
0.932   1.471   !Total  Heating(kW)and  Power(kW)       at      55/0
1.065   1.507   !Total  Heating(kW)and  Power(kW)       at      55/5
1.202   1.543   !Total  Heating(kW)and  Power(kW)       at      55/10
1.331   1.572   !Total  Heating(kW)and  Power(kW)       at      55/15
0.745   1.449   !Total  Heating(kW)and  Power(kW)       at      65/-5
0.892   1.500   !Total  Heating(kW)and  Power(kW)       at      65/0
1.000   1.594   !Total  Heating(kW)and  Power(kW)       at      65/5
1.129   1.652   !Total  Heating(kW)and  Power(kW)       at      65/10
1.266   1.732   !Total  Heating(kW)and  Power(kW)       at      65/15
```

Figure 17 External file for Vitocal 300 G

The type 927 is equipped with a control signal that can activate the heating mode and its definition will be introduced in Chapter 10.
The main equations describing the technical behavior of the heat pump during the heating mode are basically four (Thornton, 2014):

$$COP = \frac{Cap_{heating}}{\dot{P}_{heating}}$$

Equation 15

$$\dot{Q}_{absorbed} = Cap_{heating} - \dot{P}_{heating}$$

Equation 16

$$T_{source,out} = T_{source,in} - \frac{\dot{Q}_{absorbed}}{\dot{m}_{source} \cdot cp_{source}}$$

Equation 17

$$T_{load,out} = T_{load,in} - \frac{Cap_{heating}}{\dot{m}_{load} \cdot cp_{load}}$$

Equation 18

Where:

- COP Coefficient Of Performance (dimensionless);
- $Cap_{heating}$ Heat pump heating capacity, measured in kJ/h;
- $P_{heating}$ Electrical consumption of the heat pump in the heating mode, measured in kJ/h;
- $Q_{absorbed}$ Energy absorbed by the heat pump in the source side , measured in kJ/h;
- $T_{source,out}$ Temperature of brine-water exiting the heat pump in the source side, measured in °C;
- $T_{source,in}$ Temperature of brine-water entering the heat pump in the source side, measured in °C;
- m_{source} Mass flow rate of brine water circulating in the ground heat exchanger, measured in kg/h;
- cp_{source} Specific heat of brine-water, measured in kJ/kgK;
- $T_{load,out}$ Temperature of water exiting the heat pump in the load side, measured in °C;
- $T_{load,in}$ Temperature of water entering the heat pump in the load side, measured in °C;
- m_{load} Mass flow rate of water circulating in the load side of the heat pump, measured in kg/h;
- cp_{load} Specific heat of water, measured in kJ/kgK;

More specifically Equation 16 represents the energy balance considering heat pump's boundaries. Equation 17 represents the energy balance concerning the heat transfer in the evaporator. On the other hand Equation 18 shows the heat transfer occurring in the condenser.
The main parameters (Table 14) and inputs (Table 15) required by the type 927 in Simulation Studio are provided in the following tables.

Parameter	Value	Unit
Source fluid specific heat	3,672	kJ/kgK
Load fluid specific heat	4,190	kJ/kgK
Source fluid density	1046,38	Kg/m^3
Load fluid density	1000	Kg/m^3
Nominal heating capacity	6200	W
Nominal electrical consumption	1380	W
Nominal source flow rate	900	l/h
Nominal load flow rate	1000	l/h

Table 14 Parameters for Vitocal 300 G

The inputs for each type in TRNSYS have been defined through connection with other types' outputs or via values calculated with equations. An example is to connect the brine-water mass flow rate exiting the ground heat exchanger with the input called "Source mass flow rate".

Input	Description
Inlet source temperature	Temperature of brine-water entering from the source side
Source flow rate	Mass flow rate of brine-water
Inlet load temperature	Temperature of water entering from the load side (DHW tank or buffer storage tank)
Load flow rate	Mass flow rate of water
Heating control signal	This control will be introduced at a later stage

Table 15 Inputs for Vitocal 300 G

7.2 Buffer storage tank

The buffer storage tank is the most important component in terms of Demand Side Management applied to heat pumps but it is also used in order to prevent frequent start-ups and stops of heat pumps. In the heating system considered with radiators it's necessary to install a tank that stores the heating water. The buffer storage tank represents a solution for decoupling the distribution heating system and the heat pump.
Moreover this solution allows users to reach several advantages such as (Viessmann, 2015):

- Overcome cut-off times imposed by electricity operator. Effectively heat pumps can be deactivated during the load peak time depending also on the electricity tariff. In this case, via the buffer storage tank, it’s possible to supply hot water to the radiators also if the heat pump is switched off;

- The heat pump works with constant mass flow rate during the heating mode. The tank is useful also to reach a hydraulic decoupling between the secondary mass flow rate (that streams between the heat pump and the buffer storage tank) and the heating circuit mass flow rate (that supplies radiators);

The Buffer Storage Tank chosen is the type Vitocell 100 E with 950 liters of capacity.
It is equipped with a thermal insulation and one temperature sensor (Viessmann, 2015).
In the technical brochure a lot of information have been obtained such as geometrical position of supply and return pipes both for heat pump side and radiators side and its thermal loss through the insulation. Both of them were useful for putting reasonable values as input in the Simulation. For instance in the manufacturer’s catalog is given the thermal loss calculated on the basis of German Standard DIN 4753-8 in the case of temperature difference of 45 K.
In this case the tank losses 4,2 kWh in 24 hours. So dividing this value first of all by the time and then by the total external area and temperature difference has been possible to define the tank loss coefficient per unit area.

7.2.1 Buffer storage tank model in TRNSYS

The Type 60 in Simulation Studio describes the technical behavior of a stratified storage tank with optional internal heaters and heat exchangers. In this case it’s important to consider the thermal stratification existing between the bottom and the top of the tank; that has been done assuming that the tank is composed of N fully mixed equal volume segments. This parameter N is called number of nodes. Type 60 enables to define the inlet and outlet positions of mass flows entering or exiting the tank as heights measured from the tank bottom. Furthermore the model includes the possibility to consider two electric resistance heating elements that can be controlled depending for example on the top tank temperature. In fact these electrical resistances employ a temperature dead band that conditions their behavior. The electrical resistance, if its control signal in that moment is 1, is activated if the top temperature is less than (T_{set} –ΔT_{db}) and also if it was on during the previous step but the temperature to control has not reached its set point value. Type 60 allows the user to consider also an internal heat exchanger (Klein, 2014). It’s important to highlight that inside the buffer storage tank used for space heating purpose there is no heat exchanger but since the same type will be used also for the domestic hot water tank it’s better to introduce now this concept.

The main parameters that influence the heat exchanger behavior are the mass flow rate, the temperature of fluid flowing inside and the thermal conductivity of the material which comprises the internal heat exchanger.

In the Figure 18 (Klein, 2014) is illustrated the energy balance calculated in Simulation Studio in each node. Since its parameters are too many in this context will be only made a list of them.

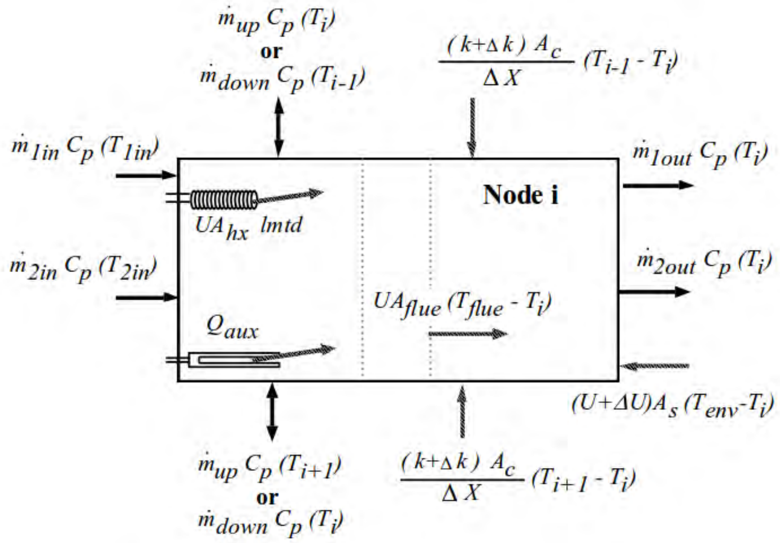

Figure 18 Energy balance inside the tank (Klein, 2014)

As can be seen in Figure 18 the energy variation of the water mass inside the node depends on the energy input or output due to mass flow rates entering or exiting the node i, the energy transfer due to mass flow rates crossing the node and coming from an upper or lower node, the energy emitted by the electrical resistance, the energy released by the heat exchanger located in the node and the thermal losses through the insulation that wraps the tank. The parameters connected to the heat exchanger are calculated iteratively. For instance in the manual may be found the following equations (Equation 19 and Equation 20):

$$h_o = \frac{Nu_D \cdot k}{d_o}$$

Equation 19

$$Nu_D = C \cdot Ra^n$$

Equation 20

Where:

- h_0 Outside convection factor for the heat exchanger, measured in W/m^2K;
- Nu_D Nusselt number for the outside flow around the tube with diameter D. This parameter is dimensionless and is defined as the ratio between heat power transferred via convection and the heat power transferred via conduction;
- k Water thermal conductivity, measured in W/mK;
- d_o Outside diameter of the heat exchanger tube, measured in m;
- C Constant used in Nusselt equation;
- Ra Rayleigh number is another dimensionless parameter associated to the convection theory and describes the natural convection;
- n Exponent used in Nusselt correlation.

Table 16 and Table 17 provide the main parameters and inputs required by the Type 60 in Simulation Studio.

Parameter	Value	Unit
Tank volume	1	m^3
Tank height	2,2	m
Height of flow inlet radiator side	0,4	m
Height of flow outlet radiator side	2,2	m
Height of flow inlet HP side	1,85	m
Height of flow outlet HP side	0	m
Fluid specific heat	4,19	kJ/kgK
Fluid density	1000	Kg/m^3
Tank loss coefficient	0,66	W/m^2K
Fluid thermal conductivity	1,40	kJ/hmK

Table 16 Parameters for Vitocell 100 E

Input	Description
Flow rate at inlet 1	Water mass flow rate returning from radiators
Flow rate at outlet 1	Water mass flow rate that supplies radiators
Flow rate at inlet 2	Water mass flow rate from the secondary pump
Temperature at inlet 1	Return water temperature from radiators
Temperature at inlet 2	Supply water temperature from heat pump
Environmental temperature	Air temperature of the basement; it allows to calculate thermal losses of the tank

Table 17 Inputs for Vitocell100 E

7.3 Domestic hot water tank

The production of Domestic Hot Water through heat pump has features completely different from those of the heating mode. In fact during the year the heat and the temperature, which are required in order to guarantee hot water, are constant. It’s important to highlight that the regulation of heat pump must allow to give the priority to the domestic hot water production than the space heating (Viessmann, 2015). Furthermore the maximum flow temperature from the heat pump that can be reached in order to satisfy the domestic consumptions depends on the heat pump chosen and the system configuration. For example in the Ground Source Heat Pump selected the maximum flow temperature that can be produced is 65°C. This means that set point temperatures above 45°C can be achieved only using an additional heater such as other external heat generators, electrical resistances or instantaneous heat exchanger. In the manufacturer's catalogue is suggested to choose a tank with 300 liters of capacity in order to fulfill the consumption for a four member family (Viessmann, 2015). As the buffer storage tank, the DHW tank is equipped with a thermal insulation that decreases thermal losses due to the difference between internal tank temperature and environment temperature. The set point temperature that will be considered for providing domestic hot water is 45°C with a dead band of 5°K in order to prevent intermittent start-ups and stops of the heat pump. As mentioned before the average daily hot water consumption has been assumed as 200 liters/day. Obviously this consumption is not shared equally during the day so it has been necessary to decide the daily profile. To do this a program developed by University of Kassel, called DHWcalc, has been used; it allows user to generate a text-file containing a list of hot water consumptions for each time step (Ulrike Jordan, 2005). This file will be used as input by the Type 60. The program distributes DHW-draw offs throughout the year with statistical means, according to a probability function and its distribution depends also on the typology of the building (single or multifamily house). Figure 19 enables to show an example of domestic hot water daily profile considering 3 days in a single family house.

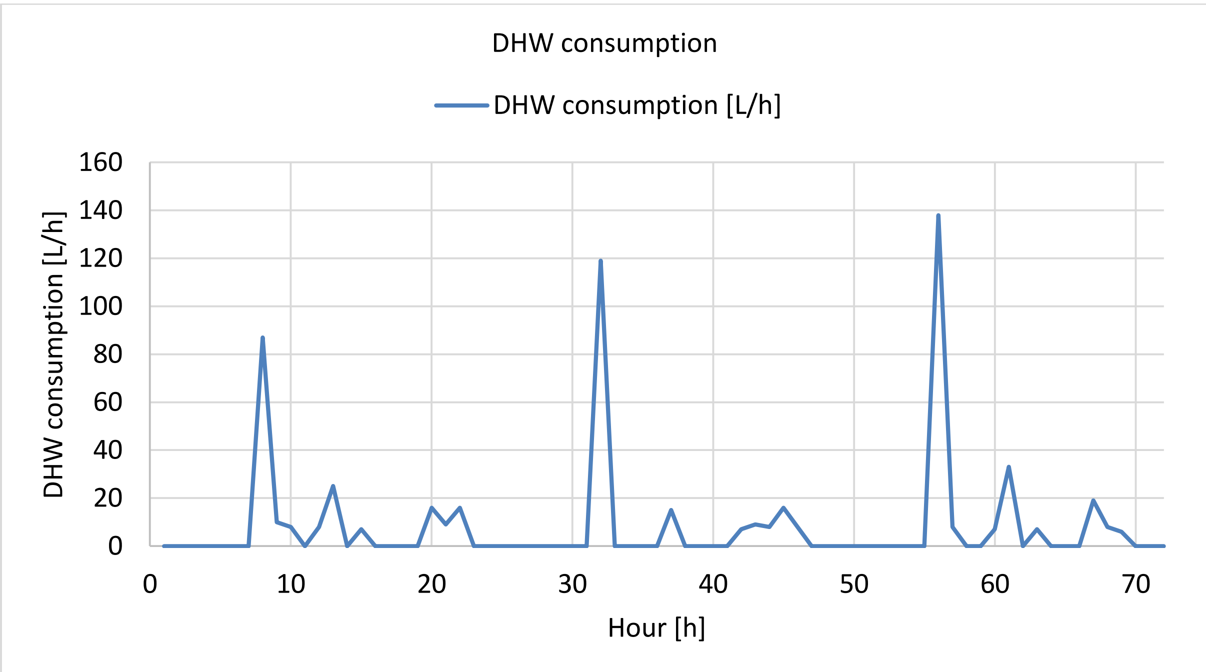

Figure 19 Domestic hot water user profile

The Hot Water Tank that has been used in the simulation is the type PSR 300.

It contains an internal heat exchanger that permits the heat transfer between the water supplied by the heat pump and the cold water coming from the aqueduct. All the information regarding the heat exchanger such as position e thermal properties have been taken from the technical brochure.

Since the mathematical theory has been already introduced in the previous paragraph (Paragraph 7.2.1), now will be shown in Table 18 and Table 19 only parameters and inputs used in TRNSYS simulation and taken from manufacturer's brochure.

7.3.1 Domestic hot water tank model in TRNSYS

Parameter	Value	Unit
Tank volume	0,3	m^3
Tank height	1,46	m
Height of flow inlet 1 (aqueduct)	0	m
Height of flow outlet 1 (consume)	1,46	m
Fluid specific heat	4,19	kJ/kgK
Fluid density	1000	Kg/m^3
Tank loss coefficient	0,66	W/m^2K
Fluid thermal conductivity	1,40	KJ/hmK
Number of internal heat exchanger	1	-
Heat exchanger inside diameter	0,0251	m
Heat exchanger outside diameter	0,0254	m
Surface area heat exchanger	1	m^2
Heat exchanger length	12,538	m
Heat exchanger material conductivity	15	W/mK
Height heat exchanger inlet	0,73	m
Height heat exchanger outlet	0,1	m

Table 18 Parameters for PSR 300

Input	Description
Flow rate at inlet 1	Water mass flow rate coming from aqueduct (DHWcalc)
Temperature at inlet 1	Temperature of aqueduct water supposed 10°C
Environmental temperature	Temperature of the basement; it allows to calculate tank's thermal losses
Flow rate for heat exchanger	Water mass flow rate supplied by the heat pump through the DHW pump
Inlet temperature for heat exchanger	Temperature of the water supplied by DHW pump
Nusselt constant	This value is assumed 0,5
Nusselt exponent	This value is assumed 0,25

Table 19 Inputs for PSR 300

7.4 Radiator's model in TRNSYS

The heating distribution systems chosen to cover the heat demand of the building are radiators working with low flow temperatures. In Simulation Studio the type that describes radiator operating is Type 362. The most important equations that describe how radiators work in the simulation are shown below; in particular the first correlation (Equation 21) represents the heat balance for the radiator (Holst, 2014):

$$\dot{m}_w \cdot c_w \cdot (t_{su} - t_{ex}) = C_{rad} \cdot \frac{\partial t_{ex}}{\partial \tau} + \dot{Q}_N \cdot \left(\frac{\Delta t_{lg}}{\Delta_{lg,N}}\right)^n$$

Equation 21

$$C_{rad} = M_w \cdot c_w + M_{met} \cdot c_{met}$$

Equation 22

$$\Delta t_{lg} = \frac{(t_{su} - t_{ex})}{\ln \frac{(t_{su} - t_a)}{(t_{ex} - t_a)}}$$

Equation 23

Where:

- C_{rad} is the lumped radiator capacitance, measured in kJ/K;
- n is the exponent of the radiator; normally it takes values in the range of 1,25-1,35;
- Q_N is the nominal radiator power calculated with Δt_N=60°C, measured in kJ/h;
- Δt_{lg} is the logarithmic temperature difference, measured in °C;
- $\Delta t_{lg,N}$ is the nominal logarithmic temperature difference, measured in °C;
- M_w is the water contents in the radiator, measured in L;
- M_{met} is the metal weight of the radiator, measured in kg;
- c_{met} is the metal heat capacity, measured in J/kgK.

Equation 22 introduces the concept of thermal inertia applied to radiator. This parameter is influenced by the metal mass of the radiator and also by its water content. A low value of the radiator capacitance would provoke sudden changes of operative temperature during the control via the thermostatic valve. On the other hand a high value of this capacitance makes it difficult to regulate heat output from radiator. In this case the radiator is not enough flexible with regard to its control (Rossi, 2013). The main parameters and inputs that have been chosen for each radiator for running the simulation are summarized in Table 20 and Table

21. In particular are shown the design value for the mass flow rate (maximum), the nominal power, the exponent n taken from radiators' catalogue and the capacitance calculated as shown beforehand (Vogel&Noot, 2015).

Room	Maximum mass flow rate (kg/h)	Nominal power (kJ/h)	Exponent n	Capacitance (kJ/K)
Living room	216,72	16883,84	1,340	107,29
Guest room	82,22	6299,22	1,323	42,47
WC	57,10	4505,85	1,353	24,40
Utility room 1	36,64	2837,43	1,334	16,39
Air trap	24,77	1918,26	1,334	12,90
Kitchen	75,51	5711,97	1,310	40,47
Parent's room	116,45	9009,64	1,333	55,97
Child 1 room	72,93	5754,45	1,353	31,09
Work	52,12	4016,59	1,329	20,76
Utility room 2	33,88	2611,45	1,329	13,67
Child 2 room	102,86	7880,62	1,323	52,97
Bathroom	100,79	7875,46	1,343	46,22

Table 20 Parameters for low temperature radiators

The nominal power with Δt_N=60°C was calculated using Equation 24 (Rossi, 2013) because the characteristic constant of the radiator doesn't change its value. In fact it has been possible to take the heat output of each radiator working with 22,5°C of temperature difference between the mean water temperature (42,5°C) and ambient temperature (20°C) in the manufacturer's catalog. Effectively in design condition low temperatures radiators will work with 45°C as flow temperature and 40°C as return temperature.

$$\dot{Q}_N = \dot{Q}_{22,5} \cdot \left(\frac{60}{22,5}\right)^n$$

Equation 24

Input	Description
Supply temperature	Temperature of the water mass flow rate supplied from the buffer storage tank
Inside room temperature	Star node temperature of the room
Mass flow rate control	This parameter will be introduced in the paragraph concerning the control system

Table 21 Inputs for low temperature radiators

Furthermore Figure 20 and Figure 21 enable to show the positioning of each distribution system installed in the single family house.

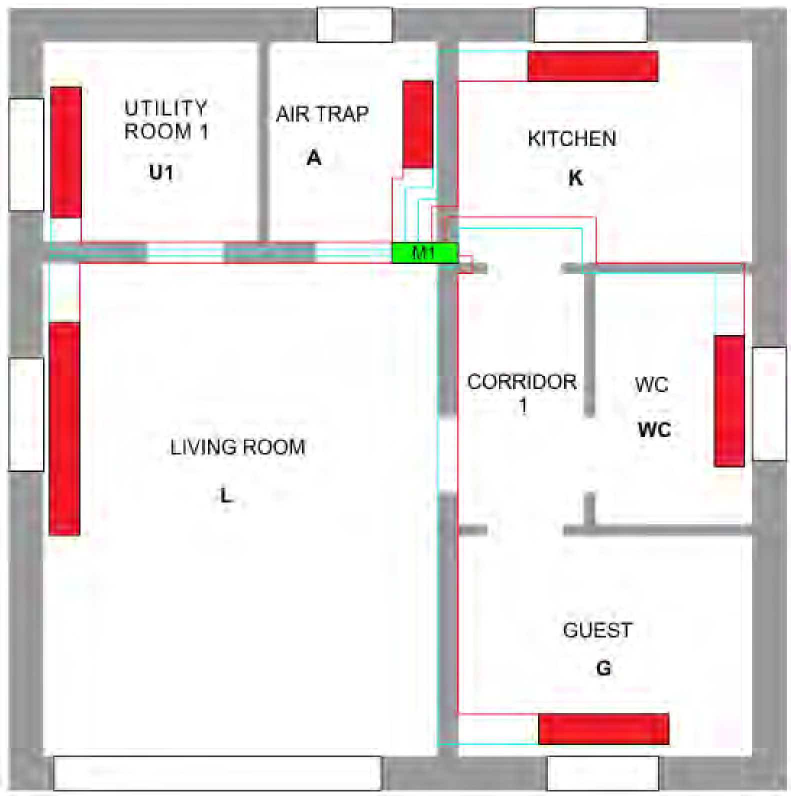

Figure 20 Radiators placed in the first floor

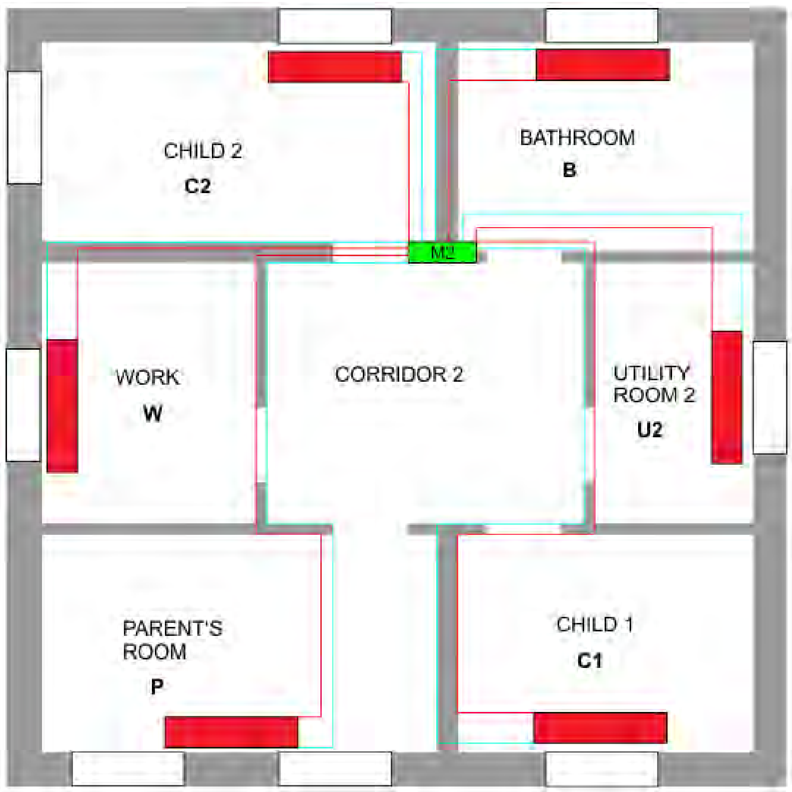

Figure 21 Radiators placed in the second floor

7.5 Vertical ground heat exchanger

A vertical heat exchanger has been considered in order to use the underground as a thermal reservoir for heating. Sizing the heat exchanger has been carried out on the basis of the German Standard VDI 4640 which can be applied to the thermal use of the underground up to 400 meters depth (VDI-4640, 2001). This standard states that in the case of systems with a heat pump heating capacity of up to 30 kW, which are only utilized for space heating and domestic hot water production, the design can be carried out using specific heat extraction values in (W/m) from the Table 22 taken from the mentioned Standard.

Underground	Specific heat extraction [W/m]	
General guideline values	1800 h	2400 h
Poor underground (λ<1,5W/mK)	25W/m	20W/m
Normal rocky underground (λ=1,5-3,0W/mK)	60W/m	50W/m
Consolidated rock (λ>3,0W/mK)	84W/m	70W/m

Table 22 Specific heat extraction depending on the underground (VDI-4640, 2001),

For this study it has been assumed a mean value of 50W/m for the specific heat extraction. Since in its nominal operating the heat pump provides 6,2 kW consuming 1,38 kW of electrical power, the thermal power captured from the ground is 4,82 kW. In order to evaluate the length of the borehole this last one value must be compared to the mean heat extraction per unit length; the specific heat extraction is the heat captured by 1 meter of borehole heat exchanger. In this way has been calculated a borehole whose length is 98 meters. Inside the borehole is installed a simple U-pipe plastic heat exchanger. The remaining space in the hole is grouted with a pumpable material that has to fill from the foot piece to the surface without any gaps. That is very important for securing the heat transport from the rock to the heat carrier fluid during heat extraction (VDI-4640, 2001). The main parameters used during the simulation have been taken from the database of Earth Energy Designer (EED, 2015). EED is a program that allows to design vertical borehole heat exchanger but in this context has been only used because it is rich in information, especially with regards to the thermal properties of the ground layers, the fill material and the plastic pipe. In particular as regards the ground stratification have been taken into account two layers; the first one is constituted by 65 meters of gravel saturated. On the other hand the second one is constituted by 40 meters of sand dry compacted. Their thermal properties will be shown in Table 23. The filling material chosen is a Raugeo product with thermal conductivity of approximately 2 W/mK.

Parameters and inputs used in Simulation Studio for defining the behavior of the ground heat exchanger are illustrated in Table 23 and Table 24. Its mathematical description would be too ample in this context so it has not been taken into consideration.

Parameter	Value	Unit
Storage volume	2000	m^3
Borehole depth	98	m
Borehole radius	0,065	m
Storage thermal conductivity	3,5	W/mK
Storage heat capacity	2400	kJ/m^3K
Outer radius of U-tube pipe	0,016	m
Inner radius of U-tube pipe	0,0131	m
Fill thermal conductivity	2	W/mK
Pipe thermal conductivity	1,512	W/mK
Reference borehole flow rate	900	Kg/h
Fluid specific heat	3,672	kJ/kgK
Fluid density	1046,38	Kg/m^3
Thermal conductivity layer 1	3,5	W/mK
Heat capacity of layer 1	2400	kJ/m^3K
Thickness of layer 1	65	m
Thermal conductivity layer 2	4,32	W/mK
Heat capacity of layer 2	1700	kJ/m^3K
Thickness of layer 2	40	m

Table 23 Parameters for vertical ground heat exchanger

Input	Description
Inlet fluid temperature	The temperature of the fluid exiting the evaporator of the heat pump and entering the ground heat exchanger
Inlet flow rate	Brine-water mass flow rate entering the ground heat exchanger
Temperature on top of storage	External air temperature
Air temperature	External air temperature

Table 24 Inputs for vertical ground heat exchanger

8 HEATING SYSTEM WITH AN AIR SOURCE HEAT PUMP

Chapter 7 showed the heating system that exploits the ground as heat source. On the other hand in this chapter an Air Source Heat Pump will be introduced for providing the energy need concerning the space heating and the domestic hot water consumption. Since the scheme regarding this heating system is almost the same if compared with Figure 13 in this chapter only the heat pump will be analyzed. In Figure 22 it's illustrated the system with Air Source Heat Pump:

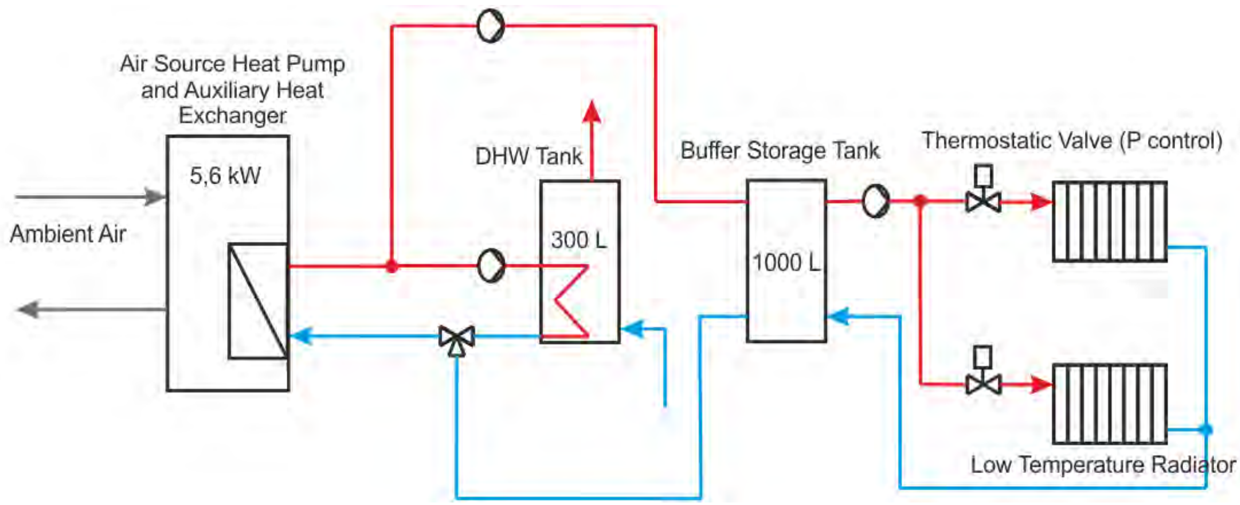

Figure 22 Air Source Heat Pump

Compared to Ground Source Heat Pumps air to water heat pumps operate mainly coupling with a second heat generator. This is due to the fact that during the coldest winter period they work very far from their design operating state and so in a less efficient way. In fact in this period the ambient temperature that is equal to the cold source temperature is very low and also it's required the highest heat demand from the building. For this reason it has been necessary to consider an auxiliary heat exchanger in order to support heat pump's operation. In this specific case the auxiliary heater works simultaneously with the heat pump when the external temperature falls below -7°C. The Air Source Heat Pump chosen for satisfying the energy demand of the building is a Viessmann product, to be more precise type Vitocal 200 S (Viessmann, 2015). The heat pump selected provides 5,6 kW of rated heating output in nominal conditions defined on the basis of the Standard EN 14511, that is 2°C for the air temperature and 35°C for the flow temperature. Furthermore a supplementary heat exchanger, that is essentially an electrical resistance whose power is 2 kW, has been considered. For this supplementary heat exchanger the electrical power corresponds exactly to its thermal power.

The most important information about the performances of the heat pump have been found in the manufacturer's catalogue. In particular will be shown the operating and performance diagrams in Figure 23 and Figure 24 (Viessmann, 2015):

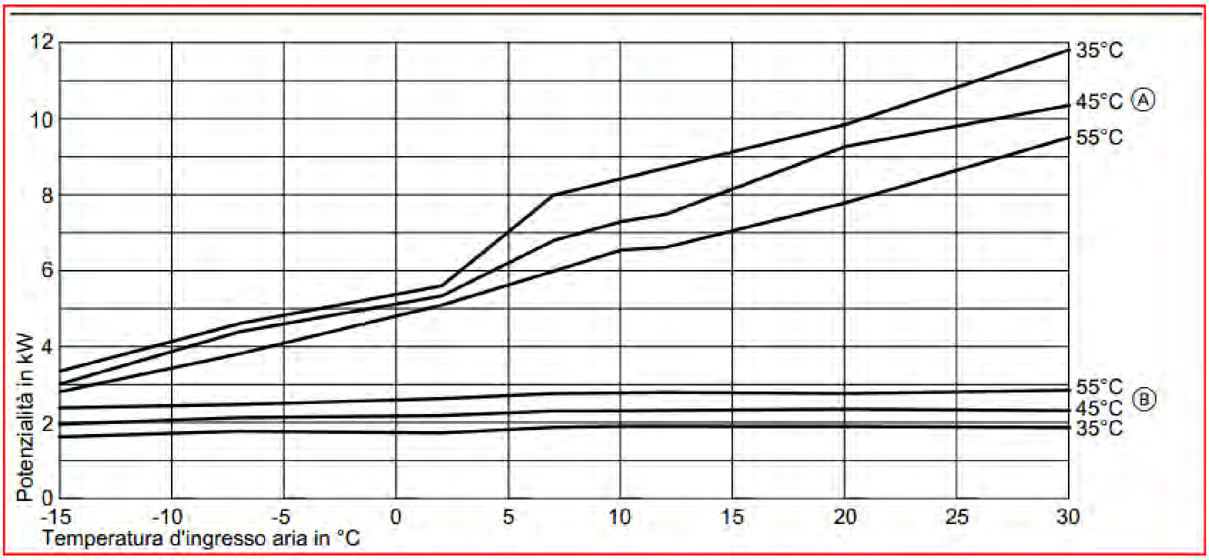

Figure 23 Vitocal 200 S powers (Viessmann, 2015)

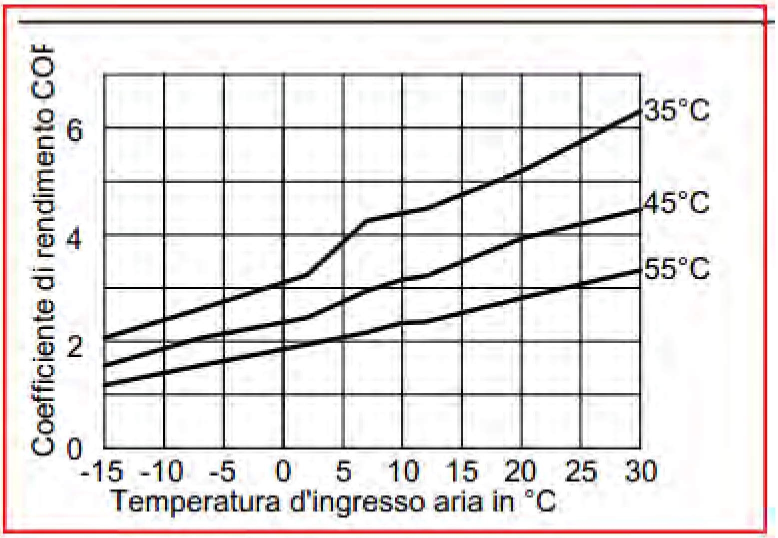

Figure 24 Vitocal 200 S COP (Viessmann, 2015)

Figure 23 enables to show the heat pump heating capacity (A) and its electrical consumption (B) depending on the external air temperature and the flow temperature to the buffer storage tank or the domestic hot water tank. Moreover the same figure explains that the heat pump Vitocal 200 S is sufficient to cover the building demand in design conditions. In fact considering -12°C as design ambient temperature the heat pump can provide 3,4 kW with

50°C of supply temperature. Then, considering the contribute of the supplementary heat exchanger (2 kW), it has been demonstrated the correct sizing of the heat pump.
Figure 24 allows to show the Coefficient Of Performance depending on the same variables. The latter, if it's compared with Figure 16, permits to compare the different temperature ranges of the two technologies. In fact on the one hand Ground Source Heat Pump works with higher temperature as regards the cold source because the cold source temperature depends on the ground temperature that is characterized by less fluctuation during the year. On the other hand since the cold source temperature for Air Source Heat Pump is the ambient temperature and it takes low values during the winter period, performances of this system will be lower than the first one considered. It will be clearer in the chapter of results.
A further important difference between the previous heat pump and Vitocal 200 S is the minimum mass flow rate that can be supplied by the heat pump. In fact the manufacturer in this case recommends to supply more than 820 kg/h of hot water during the operation time. So the nominal mass flow rate that has been chosen for the domestic hot water production is 820 kg/h (Viessmann, 2015).

8.1 Air to water heat pump model in TRNSYS

The Type that describes air to water heat pumps in TRNSYS is the Type 941. In consideration of the fact that all the equations regarding the heat pump operation have been already introduced in Chapter 7 now only the external file that describes heat output and electricity consumption will be presented. The external file has been built up starting from Figure 23 shown beforehand considering 8 temperatures for the cold source.

AirWaterHP2.dat - Blocco note

File Modifica Formato Visualizza ?

```
30      40      50                                                             !Twater_in
-15.0     -7.0     2.0     7.0     10.0     12.0     20.0     30.0     !Tair_in
0.607   0.942    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=-15
0.821   1.023    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=-7
1.000   1.000    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=2
1.429   1.087    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=7
1.464   1.104    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=10
1.518   1.104    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=12
1.768   1.098    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=20
2.107   1.087    !Fraction Heat Capacity and Power Consumption with Twater=30 and Tair=30
0.537   1.133    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=-15
0.786   1.237    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=-7
0.946   1.266    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=2
1.214   1.335    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=7
1.304   1.343    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=10
1.340   1.347    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=12
1.661   1.364    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=20
1.857   1.341    !Fraction Heat Capacity and Power Consumption with Twater=40 and Tair=30
0.500   1.387    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=-15
0.679   1.433    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=-7
0.911   1.526    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=2
1.071   1.601    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=7
1.161   1.613    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=10
1.179   1.618    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=12
1.393   1.601    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=20
1.696   1.653    !Fraction Heat Capacity and Power Consumption with Twater=50 and Tair=30
```

Figure 25 External file for Vitocal 200 S

Figure 25 shows how the external file regarding the Air Source Heat Pump used during the simulations in TRNSYS has been made. In fact in the first line it’s explained that the temperature difference considered in the load side in this case is 5°C in all the operating range of the heat pump.

The following charts enable to explicate the main parameters (Table 25) and inputs (Table 26) utilized during the simulations in Simulation Studio:

Parameter	Value	Unit
Specific heat of liquid stream	4,19	kJ/kgK
Blower power	70	W
Total air flow rate	3600	m^3/h
Nominal heating capacity	5600	W
Nominal electrical consumption	1730	W
Capacity of auxiliary	2000	W

Table 25 Parameters for Vitocal 200 S

Input	Description
Inlet liquid temperature	Temperature of water entering the condenser of heat pump
Inlet liquid flow rate	Water coming from buffer storage tank or domestic hot water tank
Inlet air temperature	External ambient temperature
Heating control signal	Control signal for the heating mode
Auxiliary control signal	Control signal used to activate the auxiliary heater

Table 26 Inputs for Vitocal 200 S

9 HEAT RECOVERY VENTILATION

Over recent years a trend that is shared by all the construction sector is to build up buildings well insulated and weather stripping in order to achieve energy savings. Especially new buildings are more airtight and consequently less well ventilated because of the envelope insulation and the enhancement of windows' properties (Rossi, 2013). In fact an adequate air change rate is essential for health and well-being of the occupants. Without an appropriate value of air change in the modern buildings problems may arise regarding the mould growth, the air humidity and the production of carbon dioxide (Viessmann, 2015).
A mechanical ventilation system with heat recovery is one of the most common solutions in order to reach better conditions concerning the indoor environmental quality (J. Laverge, 2013). Basically this system has a heat recovery exchanger which employs a cross flow heat exchange between fresh and exhaust air. The air is continuously extracted from rooms characterized by high values of humidity production such as bathrooms, kitchens and toilets. At the same time fresh air is supplied to rooms that require higher thermal comfort such as living rooms and bedrooms. During the winter period recovering energy from the exhaust air is a way to save energy and heat incoming fresh air. In Figure 26 it's shown a schematic diagram of the system considered.

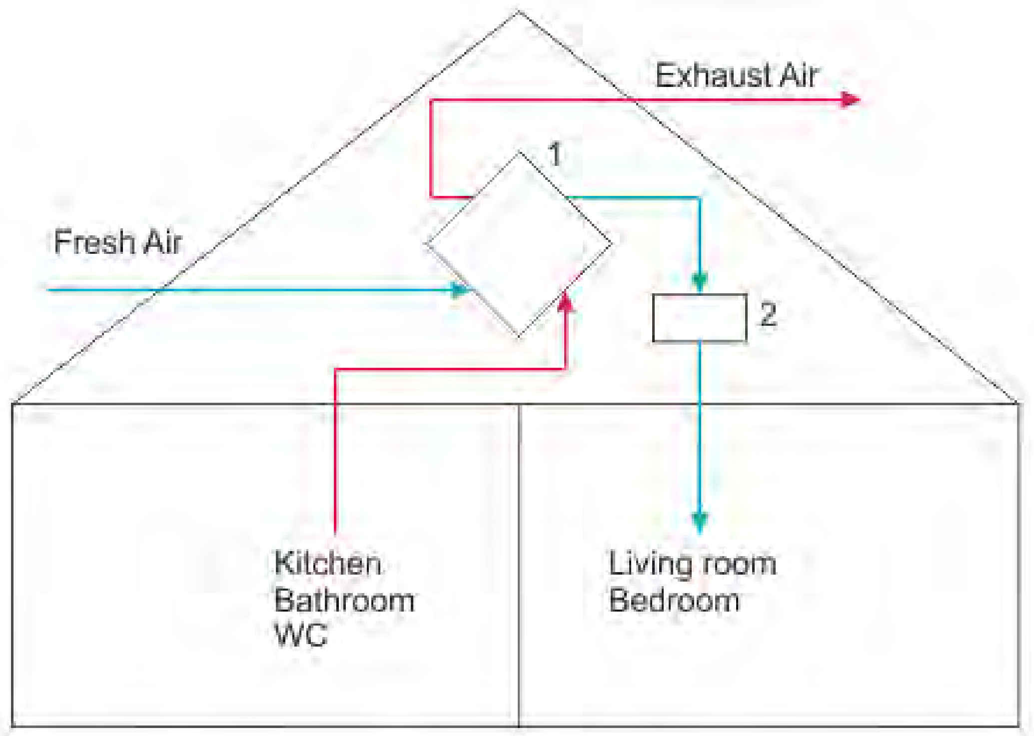

Figure 26 Mechanical ventilation system

Figure 26 shows the main components of the mechanical ventilation system:

- 1 is the recuperator air to air heat exchanger that permits to heat or preheat the fresh air to the detriment of exhaust air. Its effectiveness is 0,8;
- 2 is a device that can heat the fresh air stream if its temperature is too low; in this case it heats the fresh air stream in order to reach 18°C. If the fresh air stream during the winter period is warmer than this value the heater is bypassed.

In the single family house considered in this work the mechanical ventilation extracts during the winter period air from Kitchen, WC, Bathroom and Utility room 2 and replaces it with fresh air in Living room, Guest room, Parent's room, Child 1 room and Child 2 room.
The air mass flow rate required by each room has been calculated on the basis of the Standard EN 15251 and then all these values have been adapted for the house examined (UNIEN-15251, 2007). After that it has been necessary to consider that different rooms are used in different periods during the day; so daily profile regarding the use of the rooms have been taken into account in order also to decrease the air mass flow rate when the rooms were supposed to be empty.
A decrease of 70% if compared to the nominal value has been hypothesized.
In the following table are shown the nominal values of air change flow rate chosen in order to maintain indoor comfort conditions inside the most important rooms.

Room	Air mass flow rate (kg/h)
Living room	72
Guest room	30
Parent's room	60
Child 1 room	30
Child 2 room	30

Table 27 Ventilation mass flow rates

10 CONTROL STRATEGIES FOR HEAT PUMP

This chapter aims to explain the control strategies for the heat pump in order to cover the space heating and domestic hot water demand of the building.

The main feature that must be highlighted if the heat pump is used to provide both space heating and domestic hot water is the priority of domestic hot water over the heating circuit; in fact when the domestic hot water tank demands heat, because its inside temperature lower than the set point value, the heat pump has to supply hot water through the DHW pump. Obviously in that time the secondary pump has to be switched off.

Furthermore through the buffer storage tank it is possible to decouple the secondary mass flow rate, which is the mass flowing from the heat pump to the buffer storage tank, and the mass flow rate of the heating distribution system, that is the sum of the flow rates required by each radiator. So the controls relative to the mass flow rates that provide radiators will be considered independent from the heat pump control. This concept will be shown in the next paragraphs of Chapter 10. As can be seen in the following picture (Figure 27) the control strategy for the analyzed heating system is schematized. In particular are highlighted 3 different control loops concerning the distribution system (purple), the secondary circuit (red) and the DHW circuit (black), respectively.

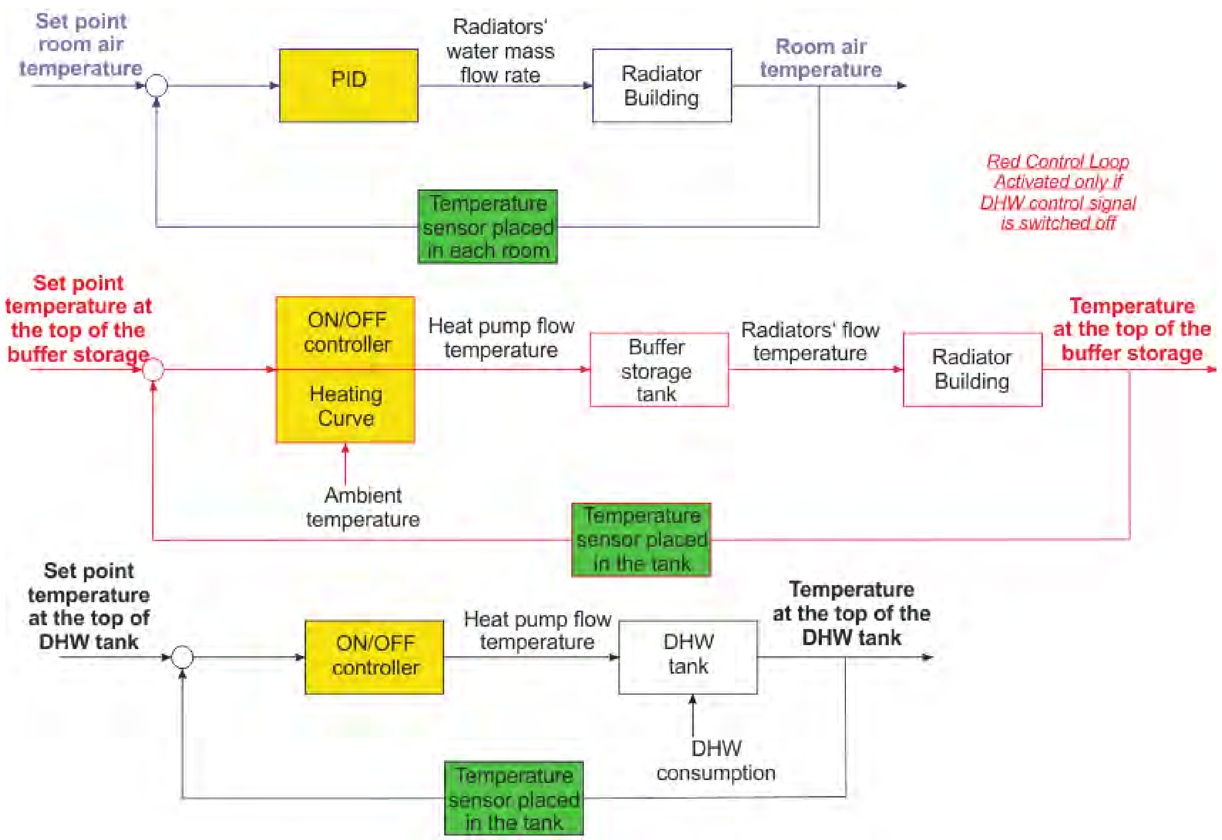

Figure 27 Explanation of heat pump control loops

10.1 Domestic hot water control loop

In the following picture (Figure 28) is depicted the control loop regarding the domestic hot water production.

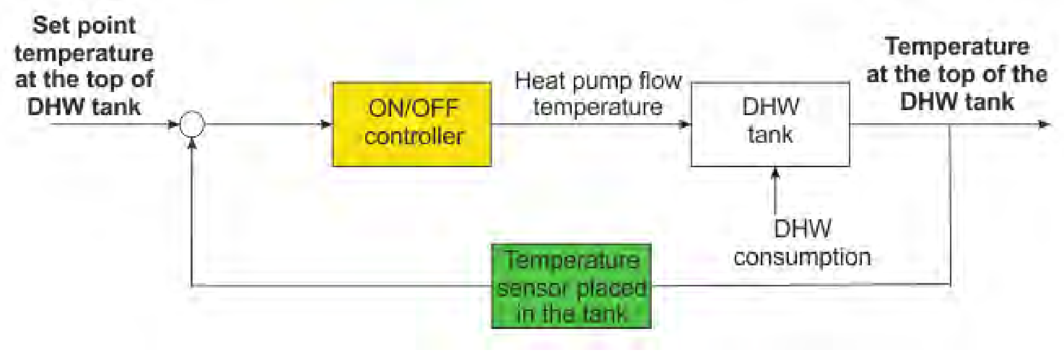

Figure 28 Control loop for DHW

Figure 28 illustrates that the variable that must to be continually controlled is the temperature at the top of the domestic hot water tank. It's useful to remind the heat pump must maintain the top temperature above 45°C in order to satisfy the domestic hot water need of the building. The top temperature in the tank is measured with a temperature sensor immersed and placed at the top of the tank. This temperature sensor gives a feedback to the heat pump in order to maintain the controlled variable in the range between 45°C and 50°C. In order to achieve this result it has been necessary to use in the simulation an on-off controller that switches on the heat pump when the mentioned temperature falls below 45°C and switches off when it exceeds 50°C. In Simulation Studio the Type that simulates the differential controller is the Type 2 (Klein, 2014). The controller generates an output that can have values 0 or 1 depending on the previous state of the controller and on the controlled temperature value. In the following table (Table 28) it's possible to find inputs for Type 2 used for controlling the domestic hot water production.

Input	Description
Set point temperature	Set point temperature for the domestic hot water tank (45°C)
Temperature to watch	Temperature of the flow exiting the tank
Turn on temperature difference	0°C
Turn off temperature difference	-5°C

Table 28 Inputs on off controller for DHW

10.2 Space heating control loop

Figure 29 illustrates the control loop concerning the production of hot water for the heating circuit. Obviously this control loop in the Simulation is set equal to zero outside the winter period.

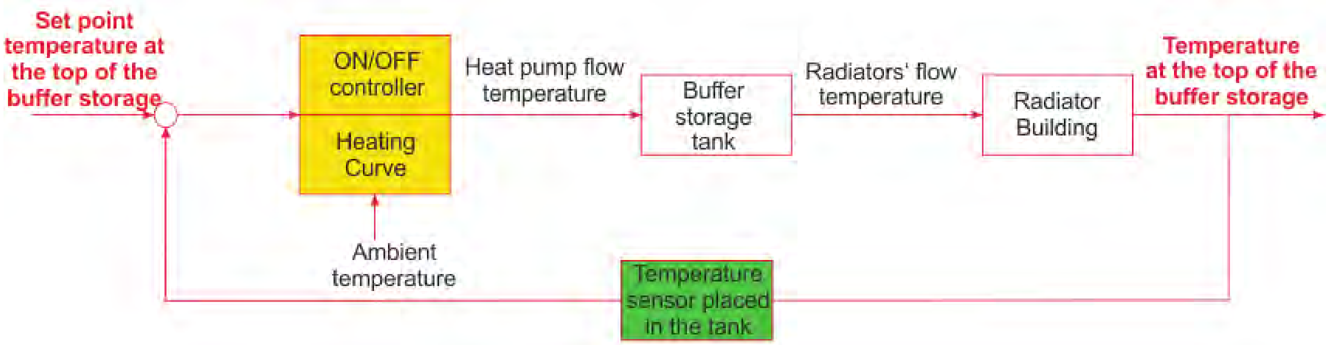

Figure 29 Control loop for space heating

Figure 29 shows that the controlled parameter in this case is the temperature at the top of the buffer storage tank. In fact this temperature will be also the flow temperature for low temperature radiators installed in each room. This temperature is measured with a temperature sensor placed at the top of the buffer storage tank that gives a feedback to the controller of the heat pump. Compared to domestic hot water set point temperature the set point temperature considered in order to provide space heating to the building is not a constant. In this case it has been necessary to consider a flow temperature for radiators depending on the external ambient temperature. In fact most heat pumps have a weather compensation function, and it is generally set-up by the installation engineer (Dimplex, 2015). The heating curve is the function that defines the flow temperature for radiators depending on the external temperature. It enables to decrease the supply temperature depending on the external conditions, in other words radiators can work with lower temperatures when ambient temperature increases. The heating curve has been built up in order to maintain the flow temperature for radiators close to the design value of 45°C when the external temperature is low. At the same time it's not possible to decrease considerably this temperature because radiators manufacturers guarantee low temperature radiators' operating only with temperatures higher than 35°C (Vogel&Noot, 2015). Furthermore this decrease of temperature is also a benefit for the heat pump since it can work with higher efficiency because of its lower supply temperature.
In the following picture (Figure 30) is shown an example of heating curve:

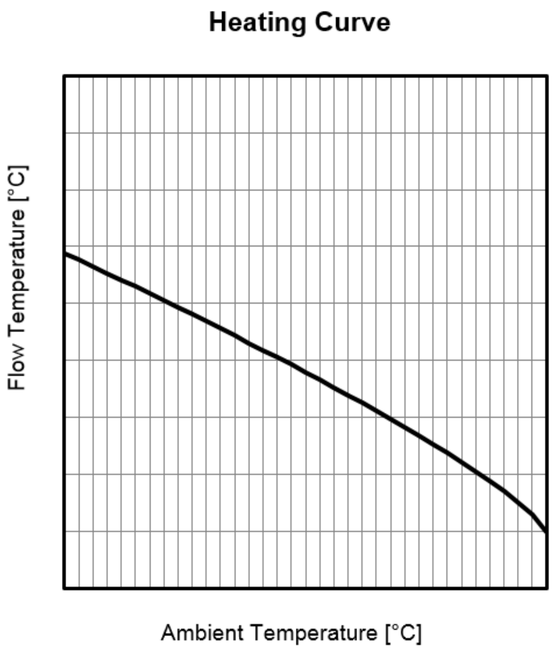

Figure 30 Heating curve

In Simulation Studio it has been considered an on-off controller with ±1°C of dead band. So that means the heat pump must work for the space heating if the temperature at the top of the buffer storage tank drops by 1°C below the set value defined via the heating curve. It's important also to remind that the domestic hot water production has the priority on the space heating, so that means the space heating can be provided only if domestic hot water demand is satisfied. As for the previous case the on-off controller has been simulated with the Type 2 in Simulation Studio (Klein, 2014). Table 29 shows inputs used for this Type in TRNSYS.

Input	Description
Set point temperature	Heating curve depending on the ambient temperature
Temperature to watch	Temperature of the flow exiting the buffer storage tank toward radiators
Turn on temperature difference	1°C
Turn off temperature difference	-1°C

Table 29 Inputs on off controller for space heating

10.3 Heat distribution system control loop

Figure 31 shows the control loop regarding the thermostatic valve used in each radiator.

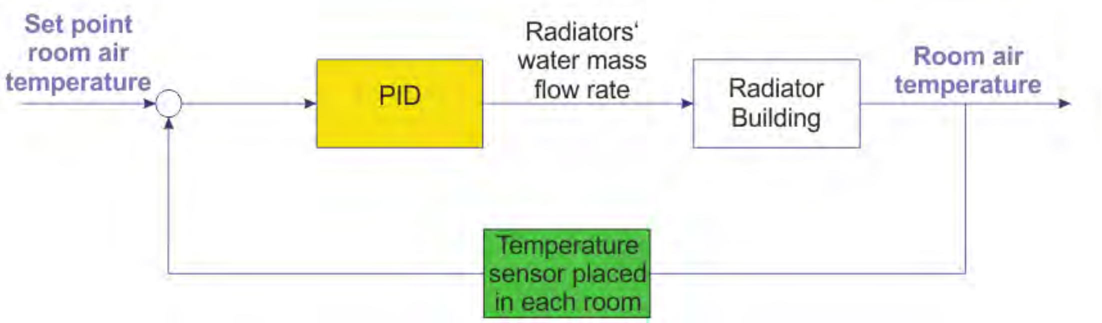

Figure 31 Control loop for radiators

The main feature of this control is to be absolutely independent from the heat pump control. So this means that for instance it's possible to have mass flow rate supplying radiators also when the heat pump is not providing heat to the buffer storage tank. Each radiator is controlled by its own thermostatic valve that regulates the mass flow rate in order to keep the set point room air temperature; hence the room air temperature is the controlled variable. In this single family house the set point temperature considered for maintaining thermal comfort conditions is 21°C during the whole day. The thermostatic valve is modeled in Simulation Studio through the Type 23 (Klein, 2014). This Type implements a proportional, integral and derivative controller but in this case study only the first control's action has been evaluated. Type 23 is applied only for controlling the mass flow rate of each radiator because the hot source is already available in the buffer storage tank. The output control signal (which may take values between 0 and 1) increases when the tracking error, defined as the difference between the set point room air temperature and the controlled temperature, increases. In fact in this situation more heating must be provided because the room air temperature is below its set point value. Furthermore in the following table can be found inputs used in the simulation.

Input	Description
Set point	Room air set point temperature
Controlled variable	Room air temperature
Minimum control signal	0
Maximum control signal	1
Gain constant	0,5

Table 30 Inputs PID controller for radiator

In closing here below is shown Figure 32 that summarizes concepts introduced in the previous paragraphs regarding the control strategies.

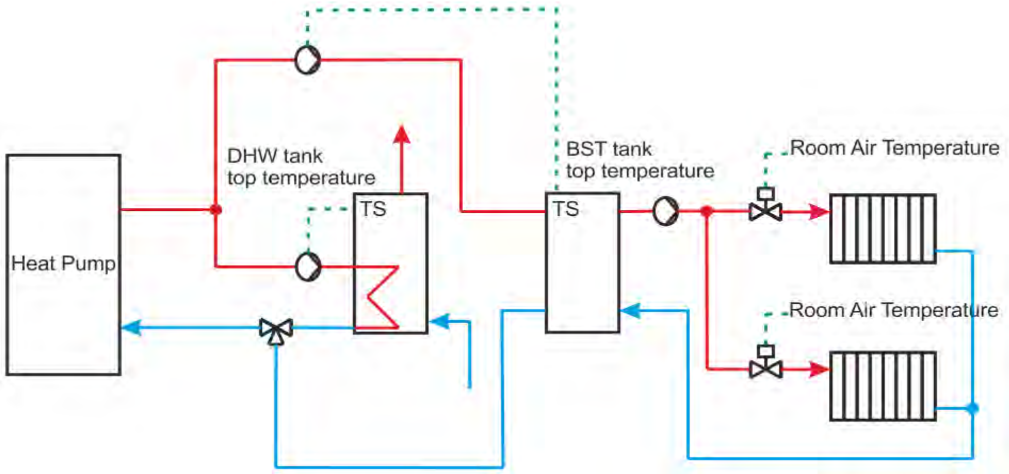

Figure 32 Explanation of the control strategies

Figure 32 highlights the controlled parameters respectively:

1) Top temperatures inside the domestic hot water tank and the buffer storage tank are the variables that influence heat pump operation. As can be seen on the picture above these control parameters are measured by means a Temperature Sensor (TS);
2) Room air temperature instead is the controlled variable that influences radiators' behavior.

10.4 Reference case and optimizations

In this study, since the aim is to apply the Demand Side Management to Ground and Air Source Heat Pump, will be considered and compared basically three cases for each heat pump:

1) <u>Reference case</u>. As regards the heat pump control the control strategy introduced in the previous paragraphs (10.1 10.2 10.3) has been applied considering also the cut-off time for the heat pump during the daily peaks of electricity demand. In particular in these periods the electricity grid operator can suspend the electrical feeding for at most 3x2 hours during a day (Viessmann, 2015). In this case two cut-off times have been considered, one from 11 to 12 in the morning and the second one from 7 to 8 in the evening. This choice is reinforced by the fact that during these periods peaks of electricity demand occur;
2) <u>Application of Demand Side Management to the space heating</u>. In this first optimization it has been studied the effect of DSM moving the heating curve depending on the electricity Tariff introduced in the chapter regarding the electricity

data (Chapter 2). In this context positive and negative shifts of the heating curve have been defined depending on the Tariff; moreover the 2 cut-off time during the day have not been taken into consideration.

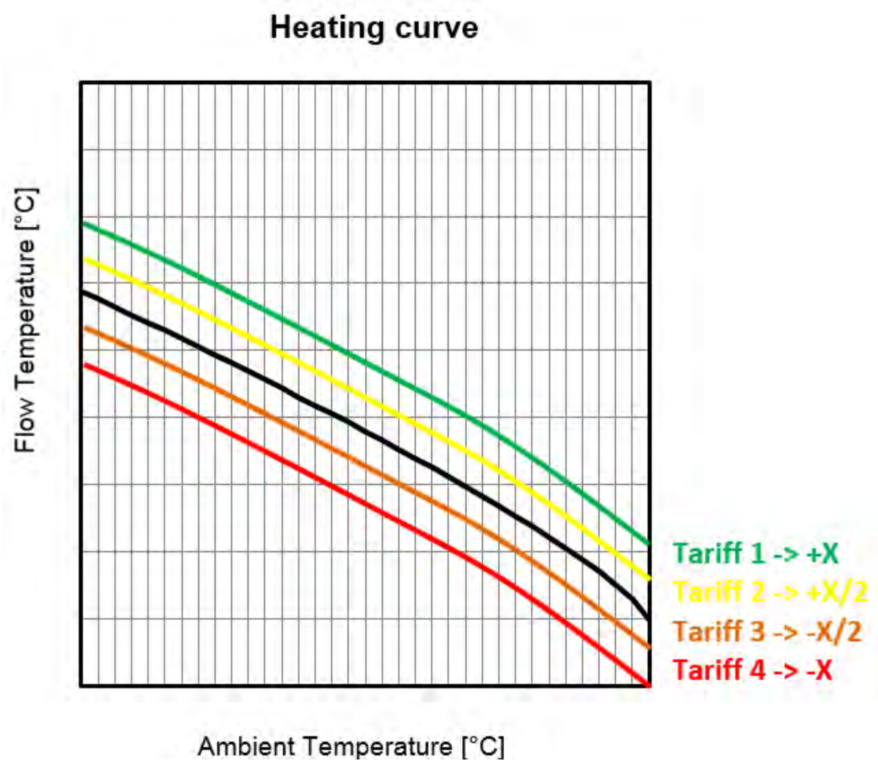

Figure 33 Adaption of the heating curve depending on the Tariff

Figure 33 enables to show the adaption of the heating curve depending on the value of the Tariff. The black line represents the heating curve used in the reference case. In particular if residual load has negative values (Tariff 1 and Tariff 2) the heating curve is increased (by X and X/2 degrees Celsius respectively), instead if it has positive values (Tariff 3 and Tariff 4) the heating curve is decreased (by X/2 and X degrees Celsius respectively). This first optimization has been carried out considering three values for the parameter X:

- ✓ X=3°C. This means moving the heating curve by +3°C, +1,5°C, -1,5°C and -3°C when the Tariff is 1, 2, 3 and 4 respectively;
- ✓ X=5°C. This means moving the heating curve by +5°C, +2,5°C, -2,5°C and -5°C when the Tariff is 1, 2, 3 and 4 respectively;
- ✓ X=10°C. This means moving the heating curve by +10°C, +5°C, -5°C and -10°C when the Tariff is 1, 2, 3 and 4 respectively.

3) Application of Demand Side Management to the domestic hot water production. The third case analyzed has entailed the use of an electrical heating element inside the domestic hot water tank. In particular an immersion heater, placed inside the domestic hot water tank, has been evaluated in order to increase the set point temperature for the DHW consumption from 45°C up to 65°C. This electrical heater works only when the residual load is negative, that is when the Tariff is 1 or 2. The main properties of this heater are summarized in the following list:
   - ✓ It's placed 0,85 m above the bottom of the tank;
   - ✓ Its control is connected to a thermostat positioned on the tank's top;
   - ✓ Its set point temperature is 65°C considering 10°C of dead band temperature difference. That means that the thermostat will enable the electrical resistance when the temperature of the fluid falls below 55°C and continue to heat the fluid until it reaches 65°C;
   - ✓ The electrical resistance is a Viessmann product and its power is 2kW (Viessmann, 2015);
   - ✓ The heat pump control for domestic hot water production is switched off when the electrical heater is operating.

In Figure 34 is shown graphically the third case considered with Demand Side Management applied to the domestic hot water tank.

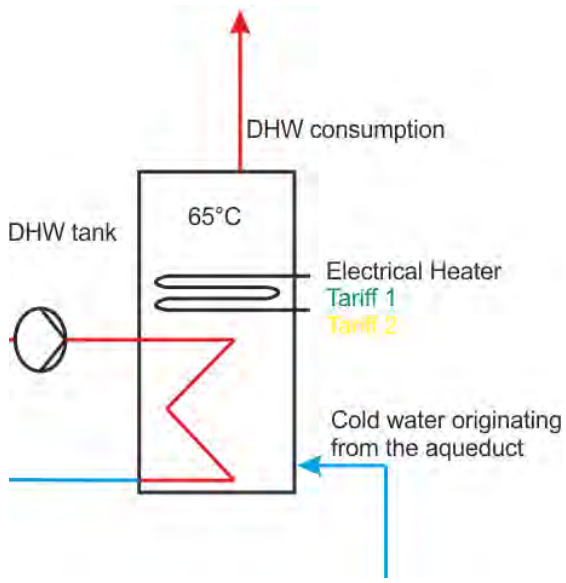

Figure 34 Domestic hot water production depending on the Tariff

Furthermore it's important to highlight two more details. First of all also in this case the two cut-off times during the day haven't been taken into account. Secondly the heat pump operation for providing the space heating is equal to the reference case; so the heating curve considered is the nominal one.

11 RESULTS

The impact of Demand Side Management applied both to Ground Source Heat Pump and Air Source Heat Pump will be shown in this chapter. For each case considered in the previous chapter will be shown graphical results about:

- Monthly and yearly electrical consumption depending on the Tariff;
- Space heating and domestic hot water demand provided by the heat pump;
- Seasonal Performance Factor (SPF) and Seasonal Primary Energy Efficiency Factor (SPEEF); these two parameters will be introduced in the relevant section;
- Over Degree Temperature Factor (ODTF) and Under Degree Temperature Factor (UDTF); the explanation of these two parameters will be given in their paragraph;
- Room operative temperature of several rooms;
- Load shifting potential of Demand Side Management applied to space heating.

As said in Chapter 10 for each heat pump system three different control strategies have been considered :

1) Reference case;
2) Demand Side Management applied to space heating. In this chapter this optimization will take the name of variation 1;
3) Demand Side Management applied to domestic hot water. This optimization instead will take the name of variation 2.

11.1 Electrical consumption depending on the Tariff

The overall electrical consumption for the space heating and the domestic hot water production will be illustrated in this paragraph considering its distribution depending on the Tariff. The following electrical consumptions have been taken into account:

- Heat pump; in the Air Source Heat Pump has been evaluated also the contribute of the auxiliary heat exchanger;
- Primary pump; obviously this contribute has been considered only in the Ground Source Heat Pump;
- Secondary pump and DHW pump;
- Electrical resistance placed inside the domestic hot water tank; this electrical consumption has been evaluated only in the case variation 2.

11.1.1 Ground Source Heat Pump

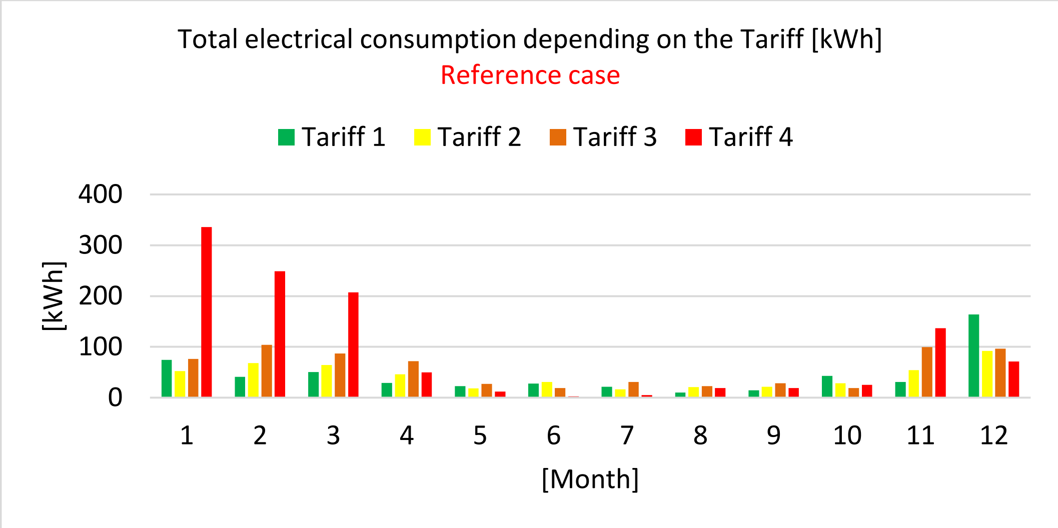

Figure 35 Monthly electrical consumption GSHP reference case

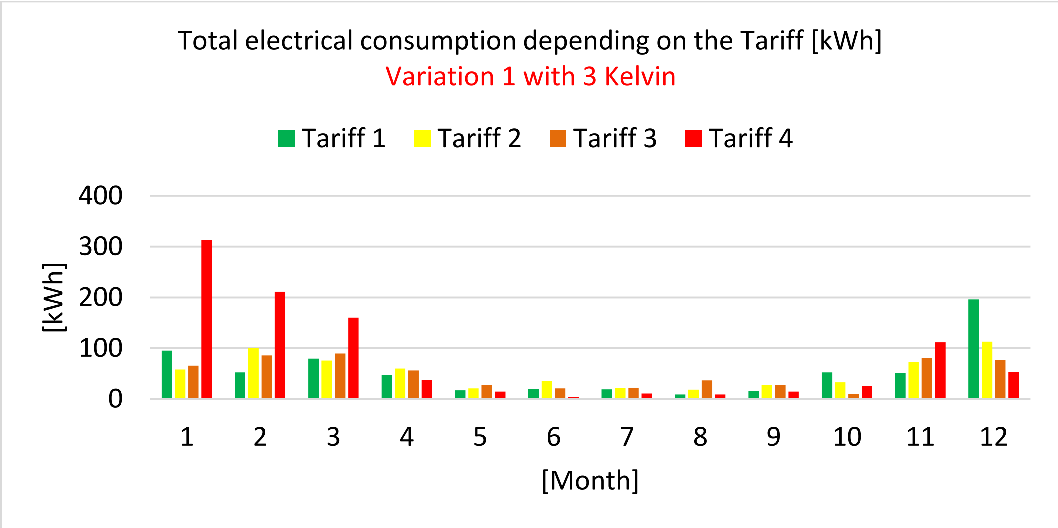

Figure 36 Monthly electrical consumption GSHP variation 1 with 3 Kelvin

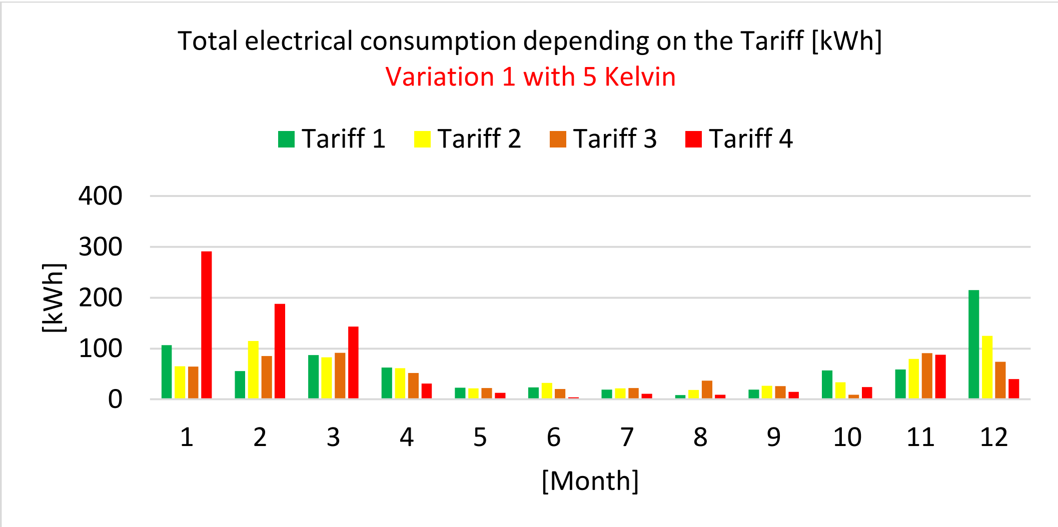

Figure 37 Monthly electrical consumption GSHP variation 1 with 5 Kelvin

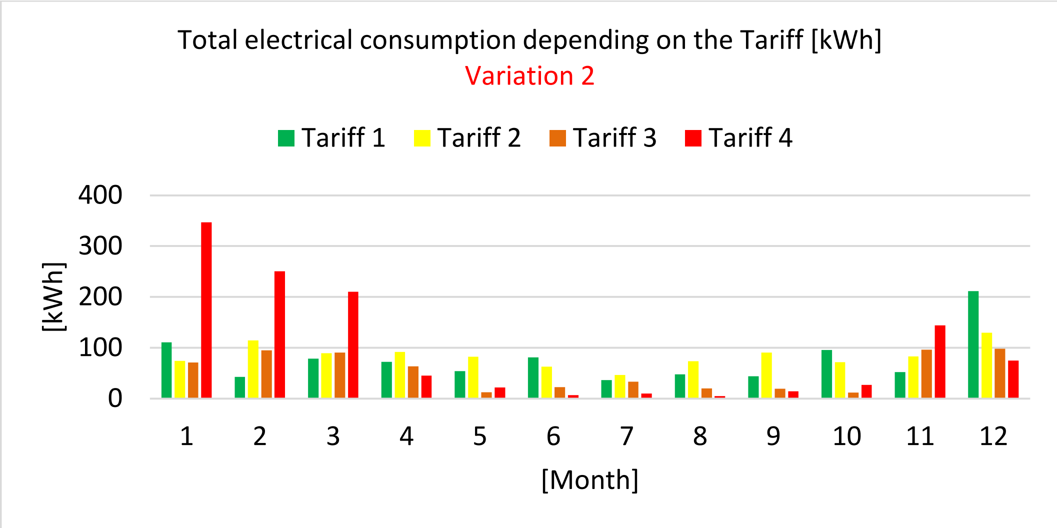

Figure 38 Monthly electrical consumption GSHP variation 2

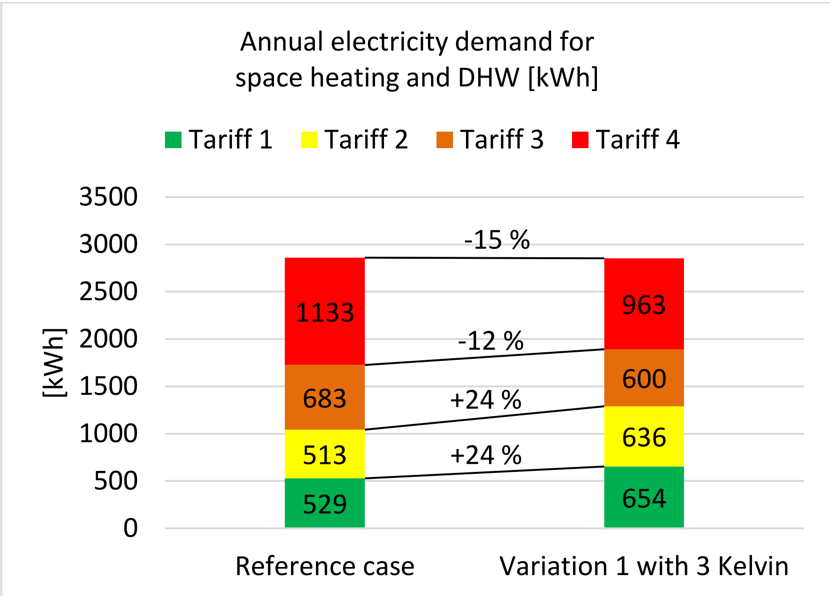

Figure 39 Annual electrical consumption GSHP reference case and variation 1 with 3 Kelvin

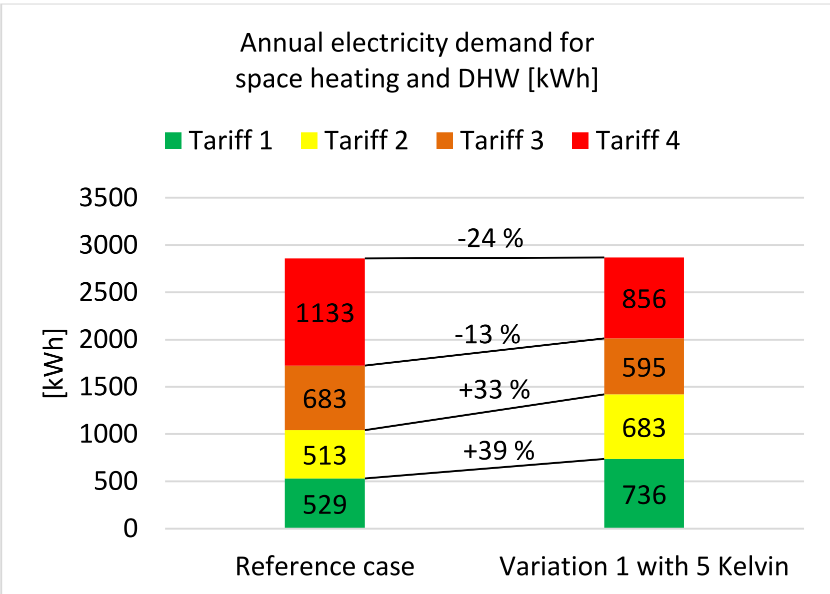

Figure 40 Annual electrical consumption GSHP reference case and variation 1 with 5 Kelvin

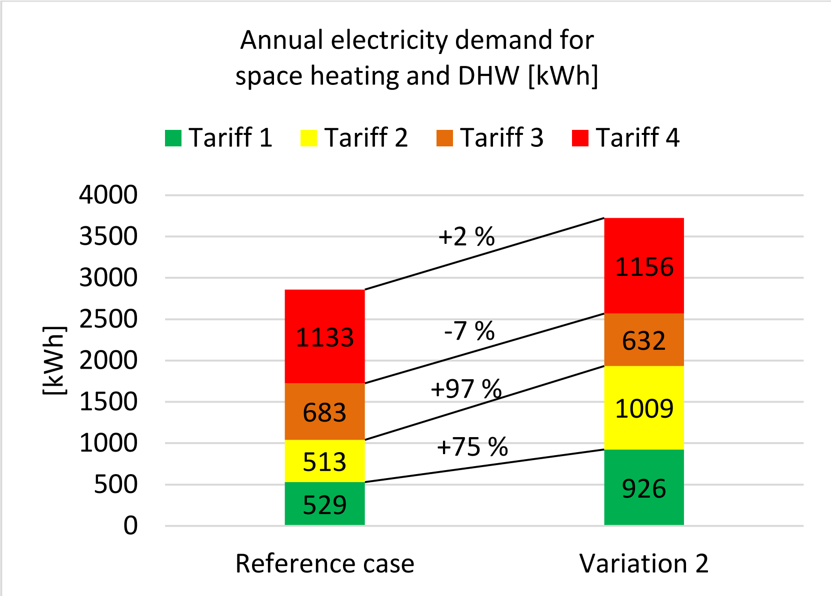

Figure 41 Annual electrical consumption GSHP reference case and variation 2

Figure 39 and Figure 40 show how the application of Demand Side Management in the space heating doesn't increase the total electricity demand. In fact in the reference case the yearly electricity demand is 2858 kWh and in the variation 1 with 3°C and 5°C it is 2853 kWh and 2869 kWh respectively. Furthermore in this type of optimization another important result is to move electrical consumption from times with positive residual load to times characterized by negative residual load during which the renewable energy production is higher than the electricity demand. That is very important in order to maximize the use of renewable energy. This behavior is more evident shifting the heating curve with the parameter X that is increased from 3°C to 5°C. In fact on the one hand the electricity consumption connected to negative values of residual load is 36% in the reference case, on the other hand this percentage increases to 45% and 49% in the other two cases. At the same time the application of Demand Side Management to the production of domestic hot water (Figure 41) in this context isn't an effective solution. The mainly result is to increase the electricity demand to 3723 kWh. Since the only consequence has been to increase the electricity consumption when residual load is negative (Tariff 1 and Tariff 2) the combined optimization of space heating and domestic hot water has not been considered. In fact the electrical resistance that increments the set point temperature at the top of the domestic hot water tank to 65°C is not effective in terms of storing energy inside the tank because the tank capacity is too small and because the water originating from the aqueduct is cold (10°C).

11.1.2 Air Source Heat Pump

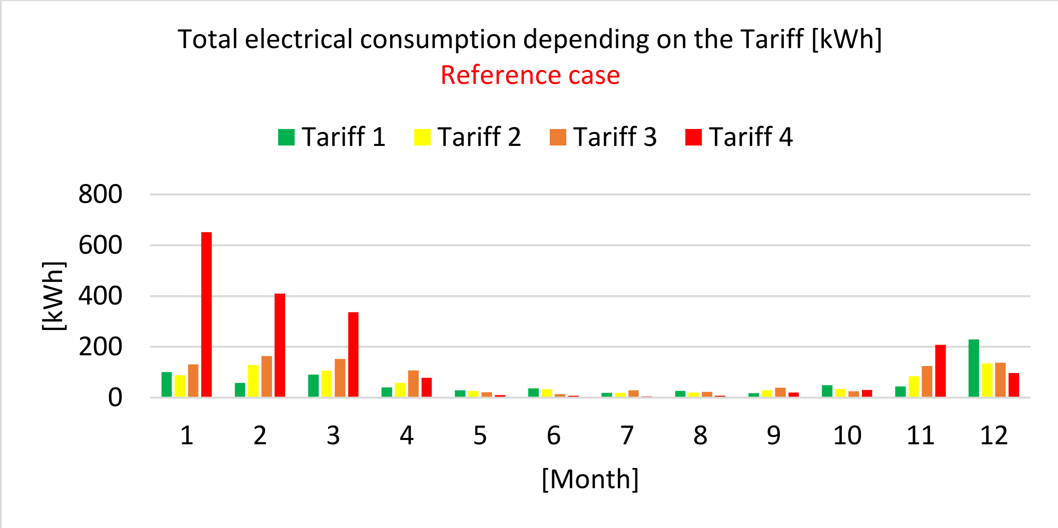

Figure 42 Monthly electrical consumption ASHP reference case

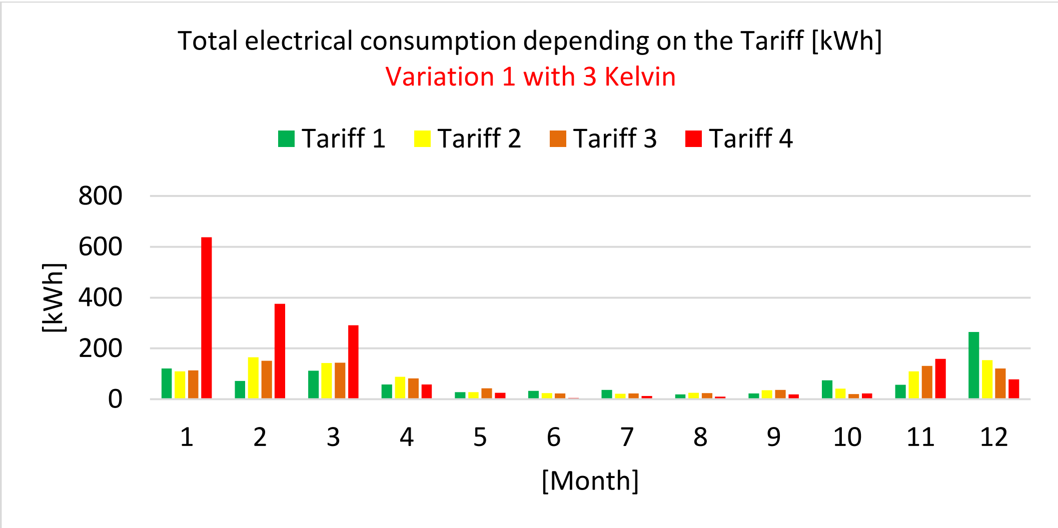

Figure 43 Monthly electrical consumption ASHP variation 1 with 3 Kelvin

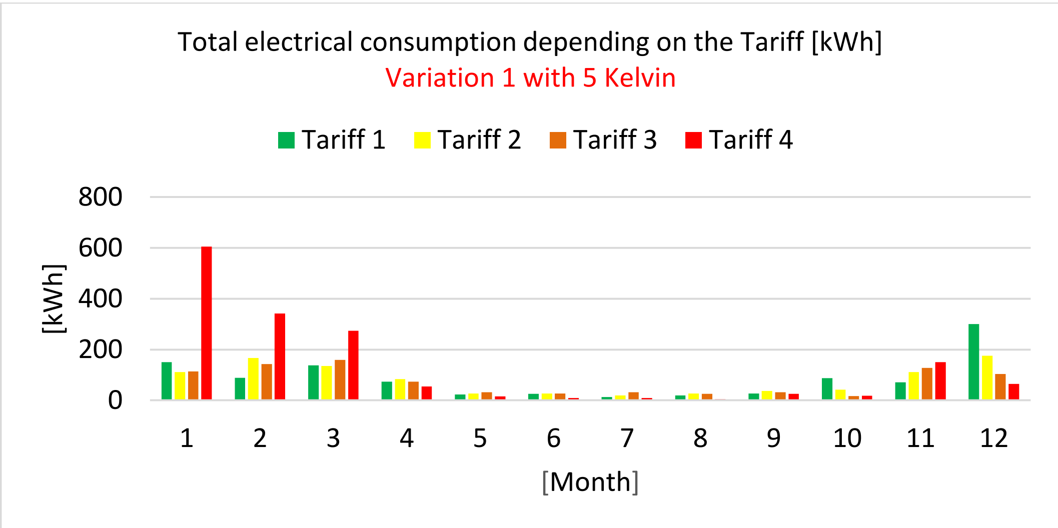

Figure 44 Monthly electrical consumption ASHP variation 1 with 5 Kelvin

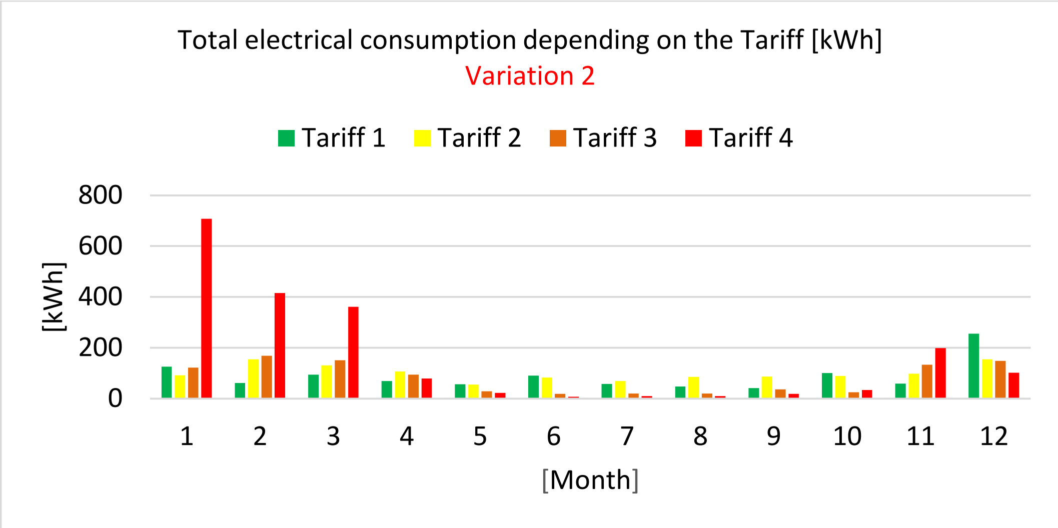

Figure 45 Monthly electrical consumption ASHP variation 2

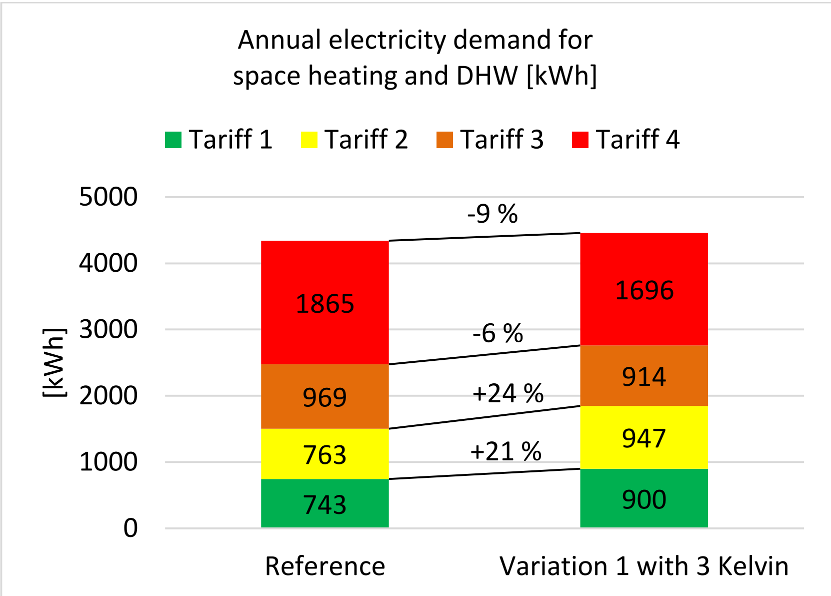

Figure 46 Annual electrical consumption ASHP reference case and variation 1 with 3 Kelvin

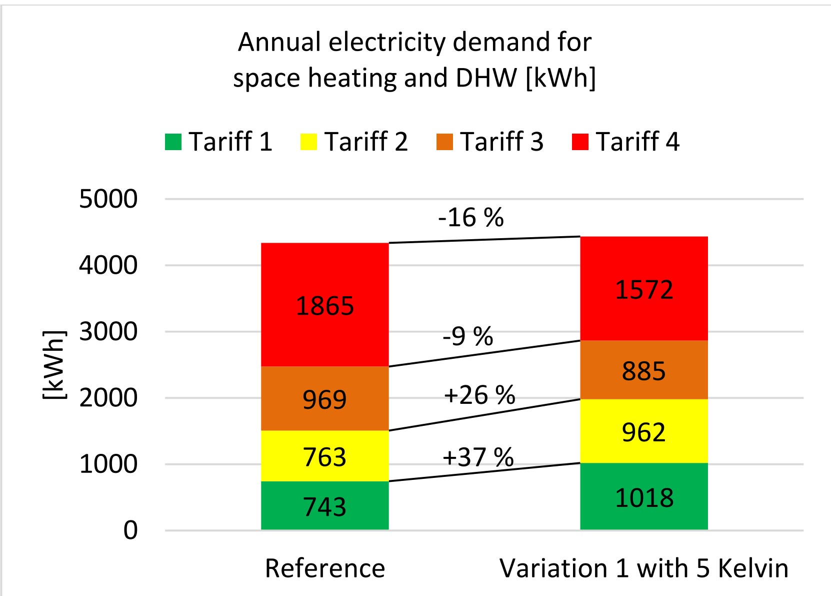

Figure 47 Annual electrical consumption ASHP reference case and variation 1 with 5 Kelvin

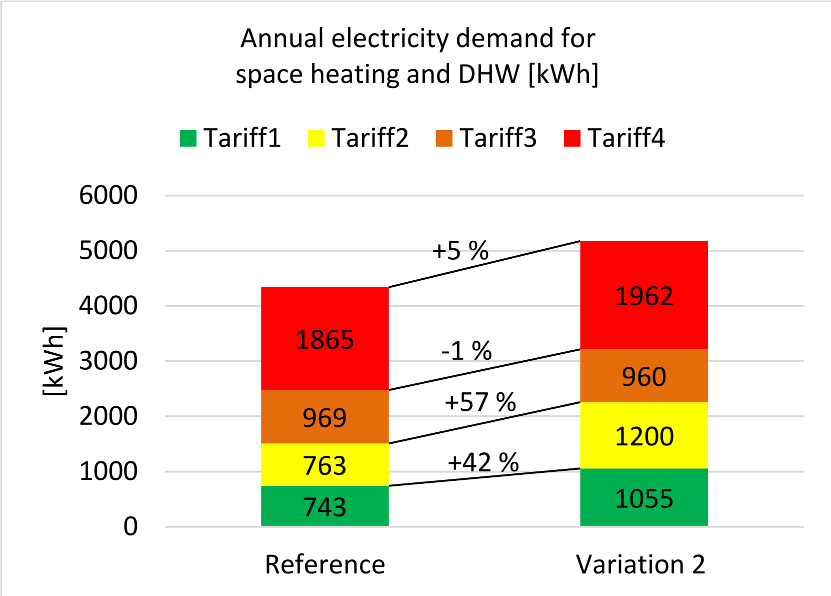

Figure 48 Annual electrical consumption ASHP reference case and variation 2

Figure 46 and Figure 47 show that also in this case the Demand Side Management applied to the space heating is effective in order to increase the portion of electrical consumptions characterized by negative values of residual load. More specifically the electricity demand covered by electrical feeding with Tariff 1 and 2 is 34% in the reference case, 41% in the variation 1 with 3 Kelvin and 45% in the variation 1 with 5 Kelvin.

Furthermore it's important to highlight that the overall electrical demand increases only of 2% in both variation 1 with 3°C (4457 kWh) and variation 1 with 5 Kelvin (4436 kWh) if it is compared to the reference case (4339 kWh). At the same time its value is higher than the one of Ground Source Heat Pump because the efficiency of this system is lower. As for the previous case the Demand Side Management applied to the production of domestic hot water is not effective because the mainly consequence is to increase considerably the overall electrical consumption up to 5177 kWh. In fact on the one hand the electrical consumption connected to Tariff 3 and 4 remains almost unchanged, on the other hand the consumption connected to Tariff 1 and 2 increases of 42% and 57% respectively.

11.2 Space heating and Domestic Hot Water demand

In paragraph 11.2 the space heating and domestic hot water demand will be shown in terms of thermal energy provided by the heat pump. These two parameters are respectively the heat output from heat pump to the buffer storage tank and to the domestic hot water tank.

The unit of measure used in the following figures is thermal kWh.

11.2.1 Ground Source Heat Pump

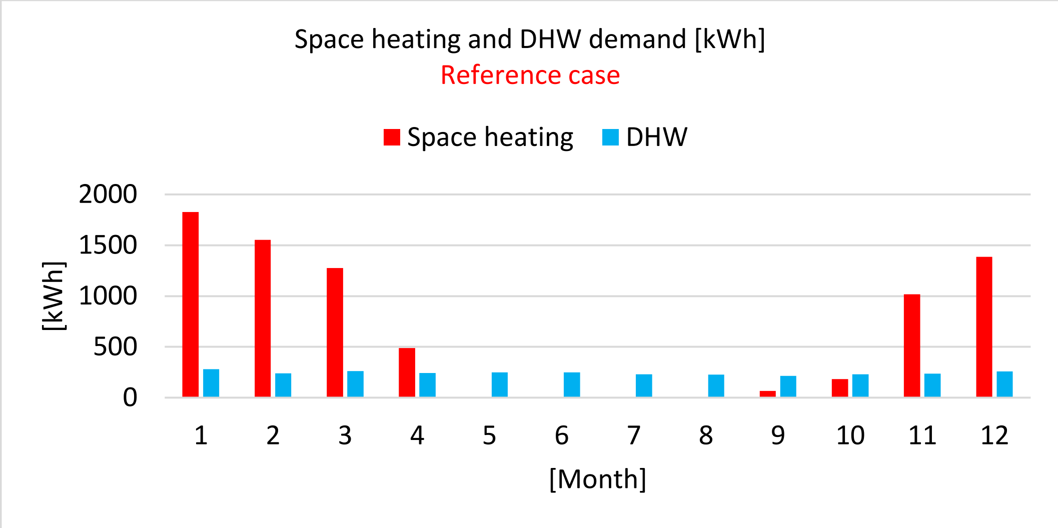

Figure 49 Monthly space heating and DHW demand GSHP reference case

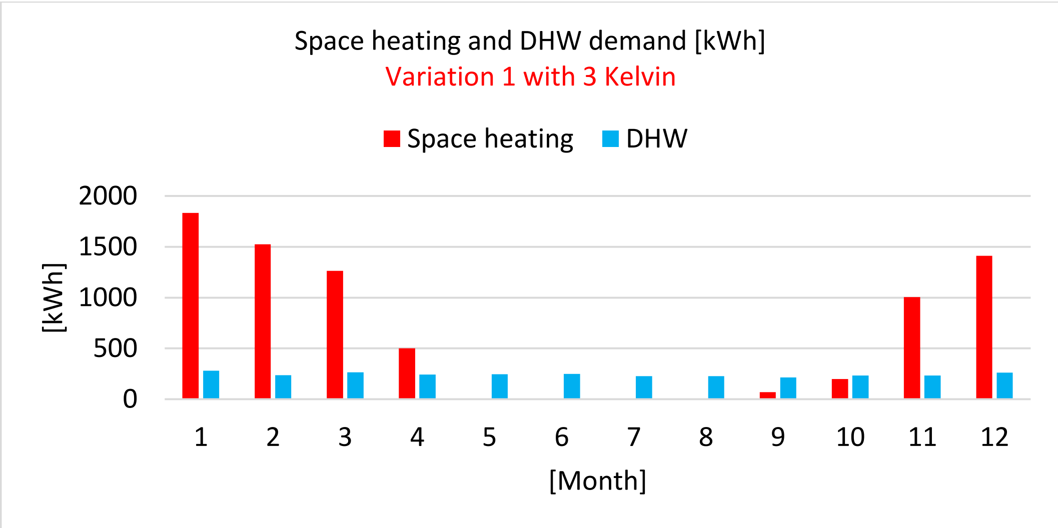

Figure 50 Monthly space heating and DHW demand GSHP variation 1 with 3 Kelvin

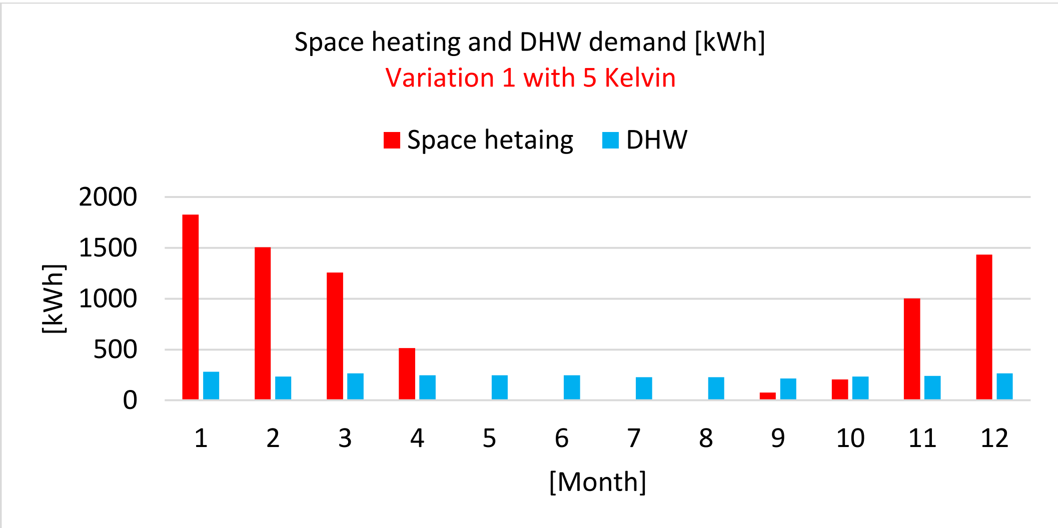

Figure 51 Monthly space heating and DHW demand GSHP variation 1 with 5 Kelvin

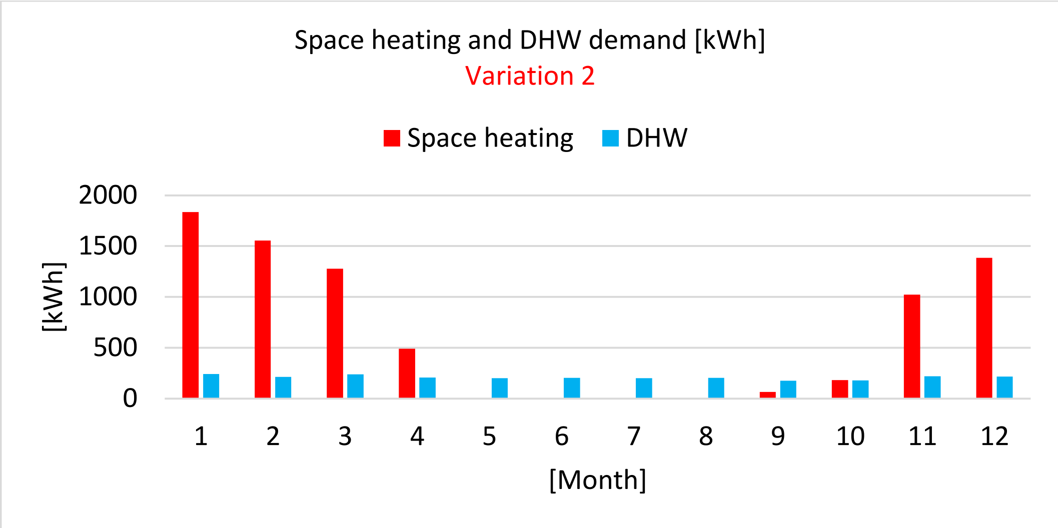

Figure 52 Monthly space heating and DHW demand GSHP variation 2

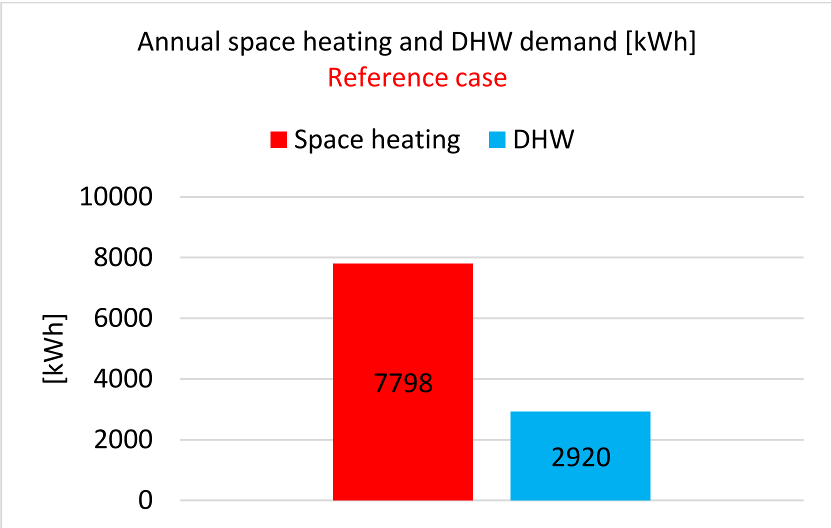

Figure 53 Annual space heating and DHW demand GSHP reference case

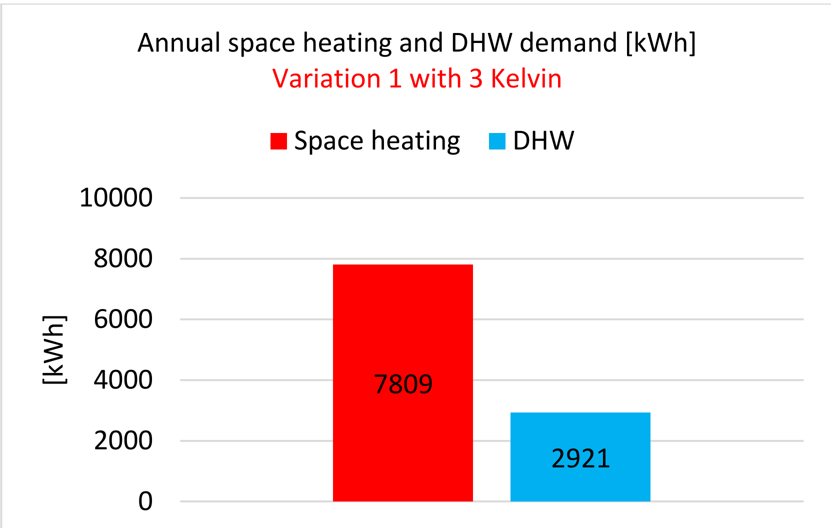

Figure 54 Annual space heating and DHW demand GSHP variation 1 with 3 Kelvin

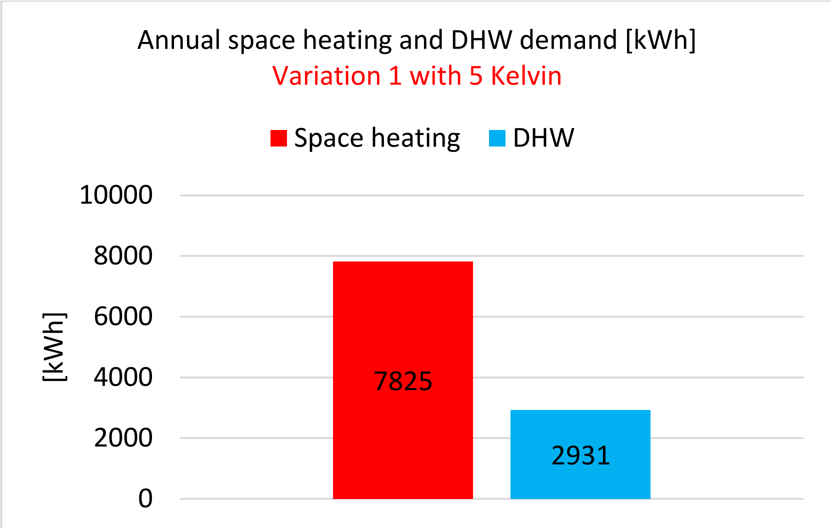

Figure 55 Annual space heating and DHW demand GSHP variation 1 with 5 Kelvin

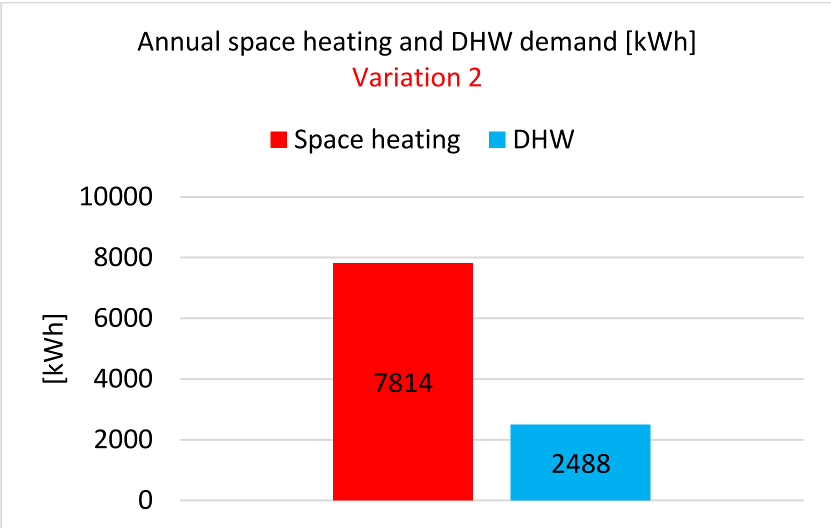

Figure 56 Annual space heating and DHW demand GSHP variation 2

The previous figures show how the space heating demand has significant values especially from November to March. It reaches its peak in January with 1829 kWh and its overall value during the heating season is 7798 kWh. These values connected to the reference case are circa equal also in all the other cases examined. On the other hand the domestic hot water demand is characterized by a constant trend during the whole year taking values about 240 kWh each month. Furthermore its annual value is 2920 kWh in the reference case with insignificant differences in the cases variation 1 with 3°C and 5°C. In the case variation 2 the overall annual domestic hot water demand is lower because of the electrical resistance operation. In fact when the resistance is switched on the heat pump doesn't work for the domestic hot water production. Finally it's important to evaluate the specific annual demand for space heating and domestic hot water production considering the heated floor area of the building (163 m^2). Hence the annual relative space heating need is 47,8 kWh/m^2year and the annual relative domestic hot water need is 17,9 kWh/m^2year.

11.2.2 Air Source Heat Pump

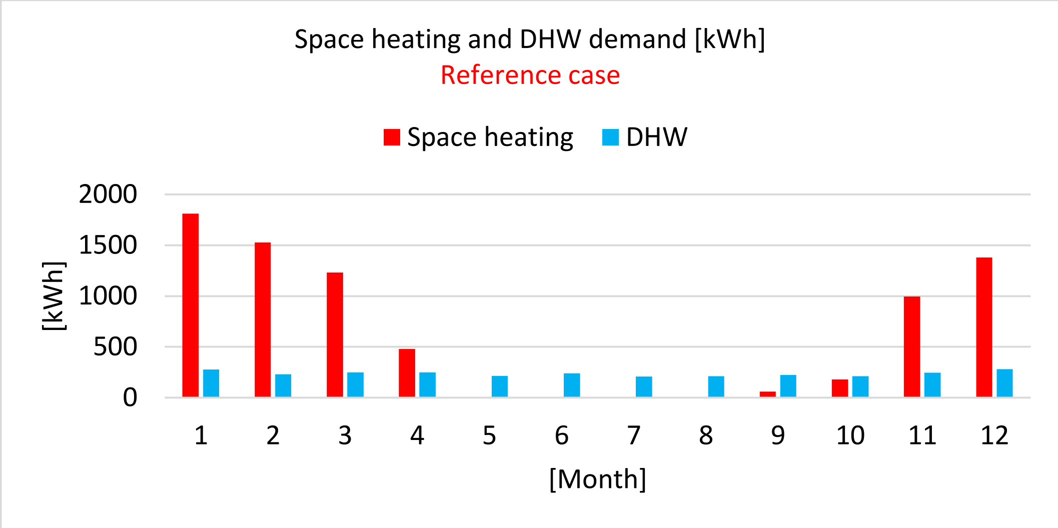

Figure 57 Monthly space heating and DHW demand ASHP reference case

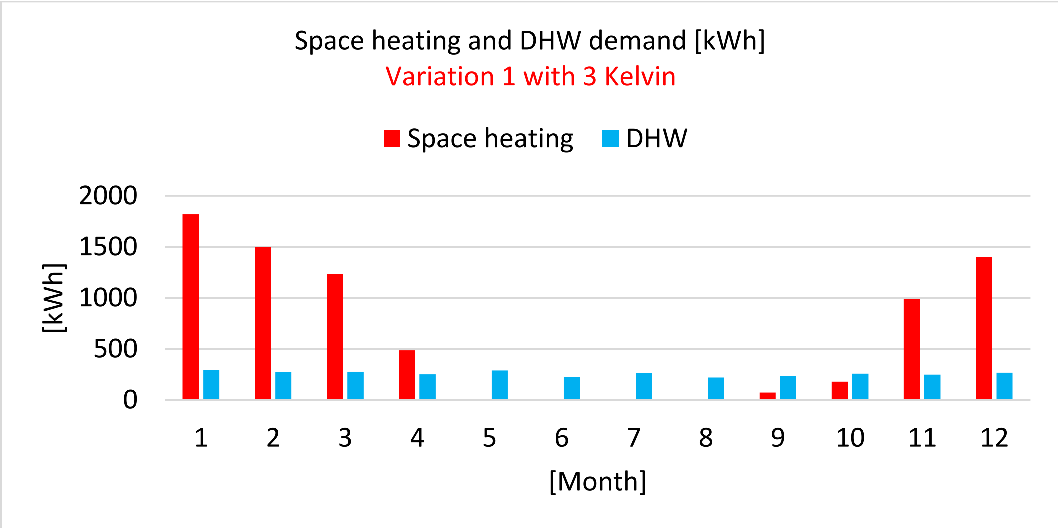

Figure 58 Monthly space heating and DHW demand ASHP variation 1 with 3 Kelvin

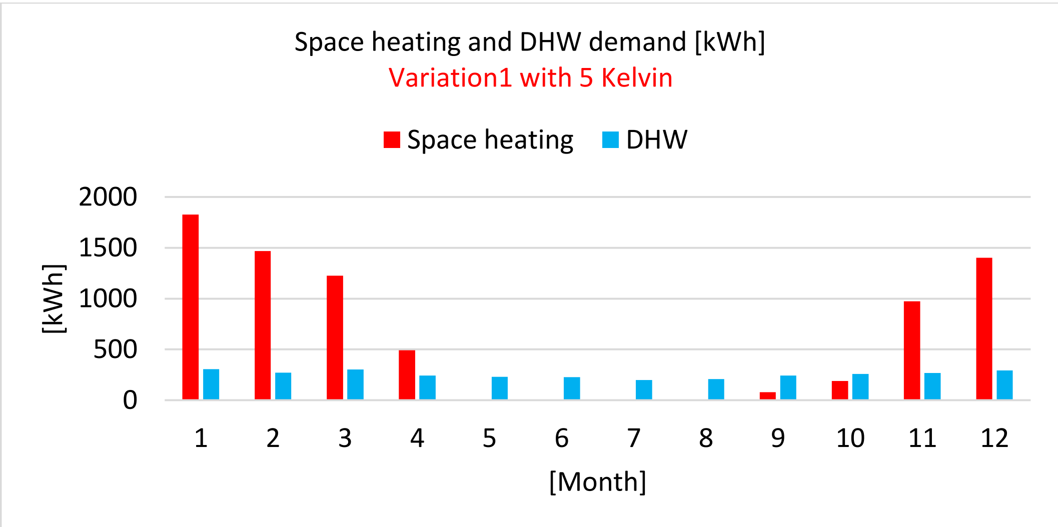

Figure 59 Monthly space heating and DHW demand ASHP variation 1 with 5 Kelvin

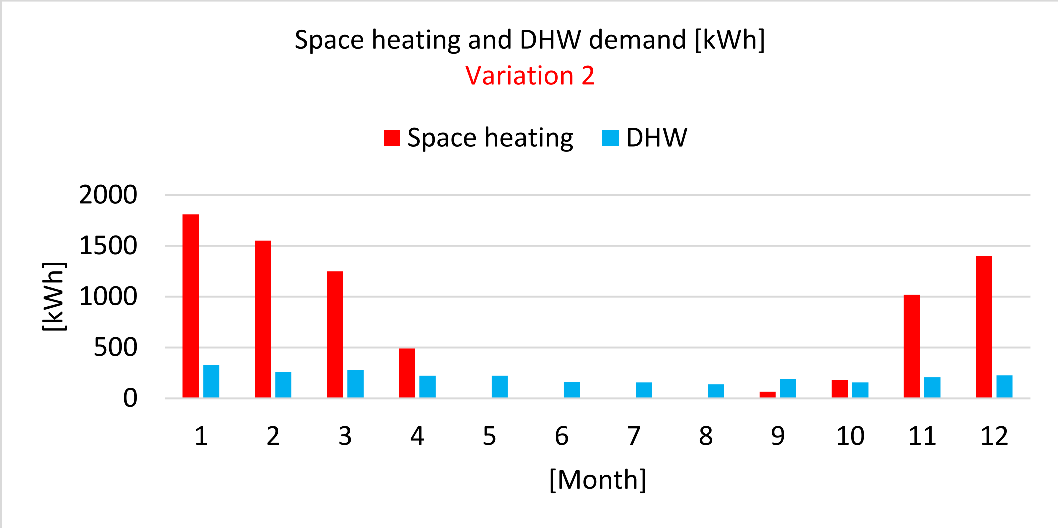

Figure 60 Monthly space heating and DHW demand ASHP variation 2

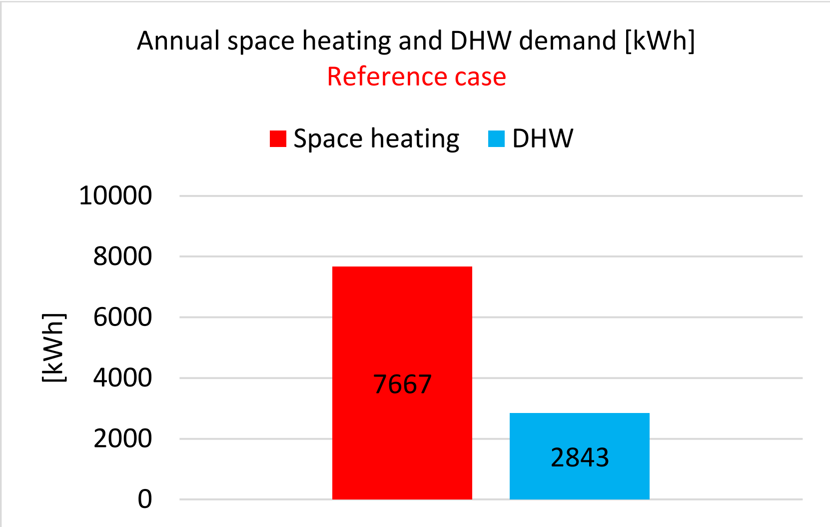

Figure 61 Annual space heating and DHW demand ASHP reference case

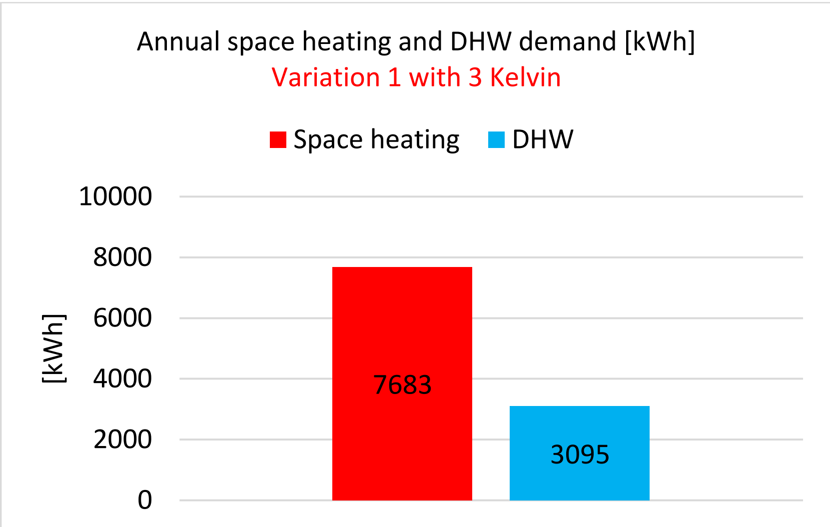

Figure 62 Annual space heating and DHW demand ASHP variation 1 with 3 Kelvin

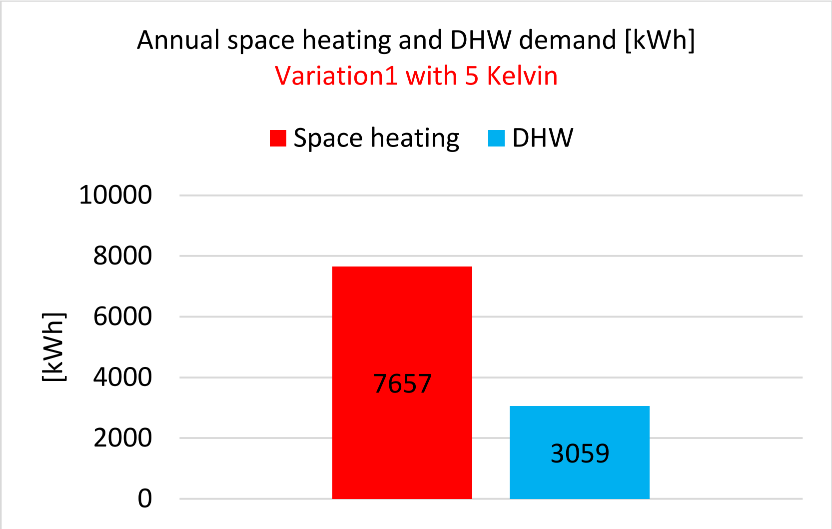

Figure 63 Annual space heating and DHW demand ASHP variation 1 with 5 Kelvin

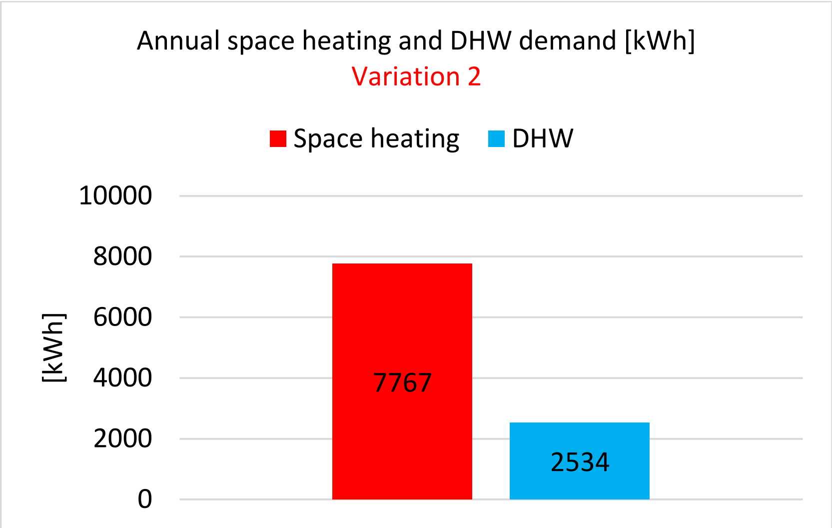

Figure 64 Annual space heating and DHW demand ASHP variation 2

The previous figures enable to show that the consumption for the space heating and the Domestic Hot Water production don't depend on the heating system that provide them to the building. In fact for instance in the reference case the space heating (7667 kWh) and DHW demand (2843 kWh) differ from Ground Source Heat Pump results only by 1,7% and 2,6% respectively. As for the previous case the space heating demand takes very low values during the months of September and October; that means the heat pump in these months works mainly for the DHW production. Moreover the last figure explains the same concept introduced beforehand, in other words that Demand Side Management applied in the variation 2 lets the heat pump work less than the other cases because of the electrical resistance placed at the top of DHW tank. Finally Figure 49 to Figure 64 demonstrate that the relative importance of the Domestic Hot Water demand in new and well insulated buildings is increasing if compared with the space heat demand. Effectively the ratio between the space heating need and DHW demand in this building is 2,68.

11.3 Seasonal Performance Factor (SPF) and Seasonal Primary Energy Efficiency Factor (SPEEF)

The efficiency of a heat pump is defined by the Seasonal Performance Factor (SPF), which is defined as the ratio between the thermal output of the heat pump (for providing the space heating and domestic hot water consumption) and the electrical consumption (Yu, 2012). In the term electrical consumption may be considered, depending on the boundaries defined, only the heat pump or also the other pumps (Miara, 2010). In this thesis the whole system, comprehending the heat pump and the pumps, has been analyzed. So that means that for the Ground Source Heat Pump has been calculated the electrical consumption of heat pump, primary pump, secondary pump and DHW pump. On the other hand for the Air Source Heat Pump has been evaluated the electrical consumption of heat pump (this term includes the auxiliary heat exchanger and the blower too), secondary pump and DHW pump. In Figure 65 and Figure 66 are better shown the system boundaries in the two cases.

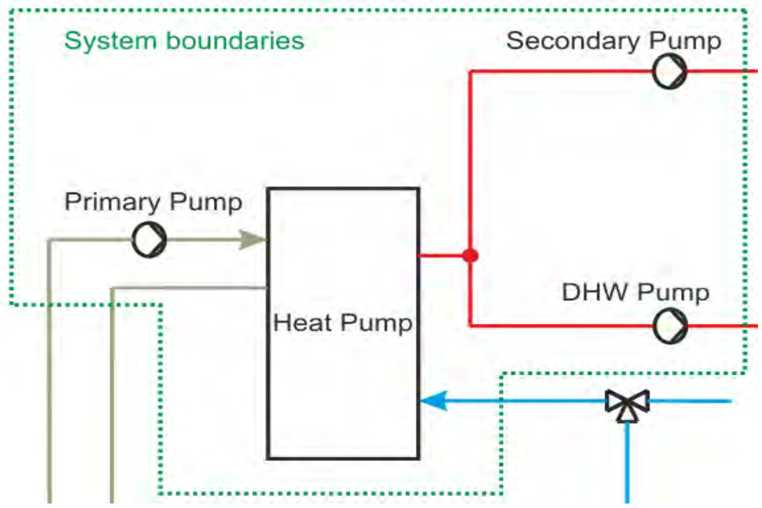

Figure 65 System boundaries for Ground Source Heat Pump

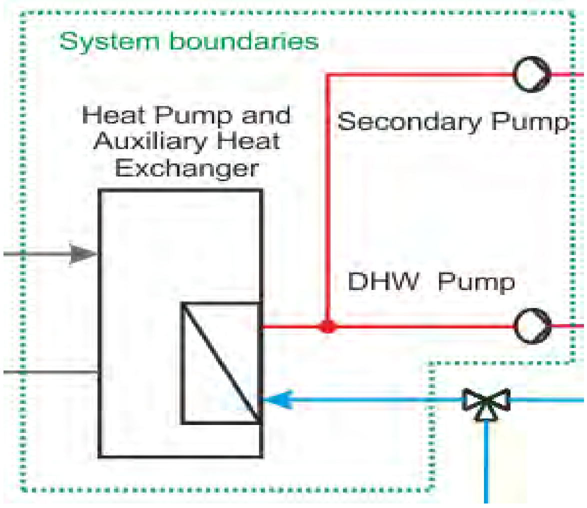

Figure 66 System boundaries for Air Source Heat Pump

In order to evaluate also the behavior of the heat pump depending on the renewable energy production it has been necessary to introduce the Seasonal Primary Energy Efficiency Factor (SPEEF) that doesn't take into account the electrical consumption from renewable energy (when the Tariff is 1 or 2) (Yu, 2012). These two efficiency parameters are described in Equation 25 and Equation 26:

$$SPF = \frac{Q_{space-heating} + Q_{DHW}}{E_{HP} + E_{pumps}}$$

Equation 25

$$SPEEF = \frac{Q_{space-heating} + Q_{DHW}}{E_{HP} + E_{pumps} - E_{RE}}$$

Equation 26

Where:

- $Q_{space\text{-}heating}$ is the heat consumption for space heating, measured in kWh;
- Q_{DHW} is the heat consumption for domestic hot water, measured in kWh;
- E_{HP} is the energy consumption of the heat pump, measured in kWh;
- E_{pumps} is the energy consumption of all the pumps, measured in kWh;
- E_{RE} is the renewable energy consumption of the heat pump and pumps, measured in kWh.

11.3.1 Ground Source Heat Pump

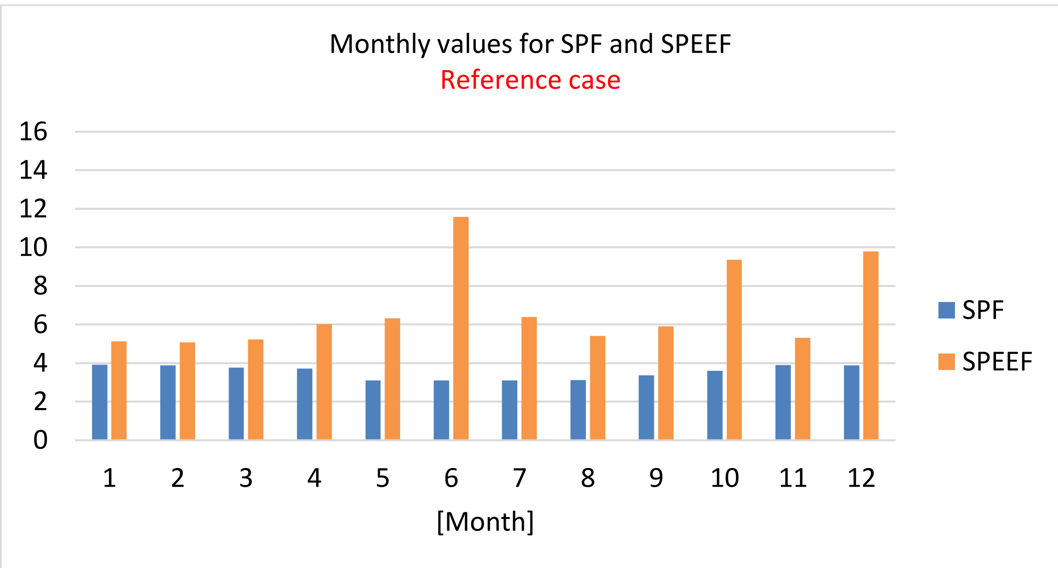

Figure 67 Monthly SPF and SPEEF GSHP reference case

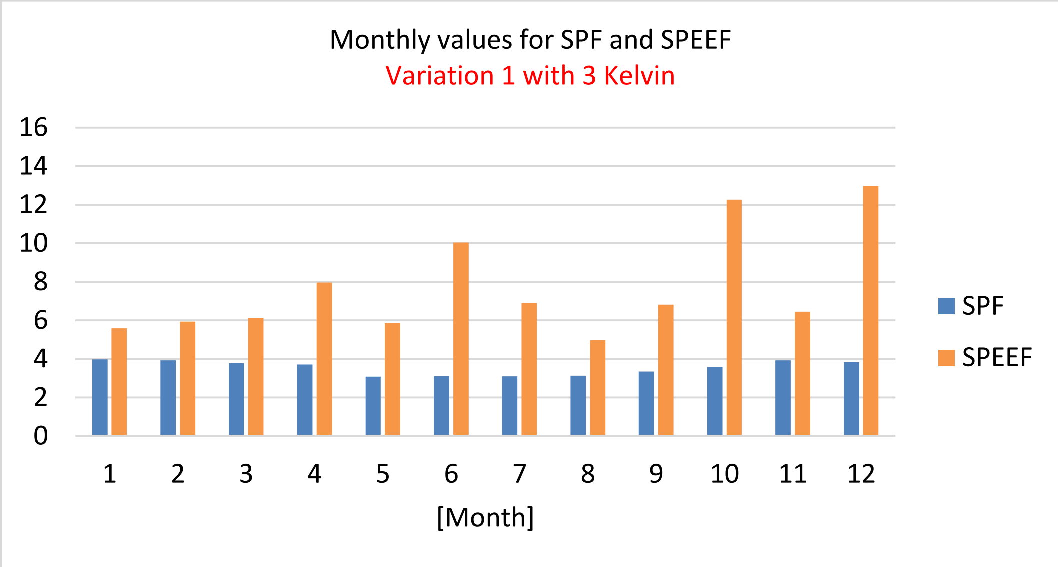

Figure 68 Monthly SPF and SPEEF GSHP variation 1 with 3 Kelvin

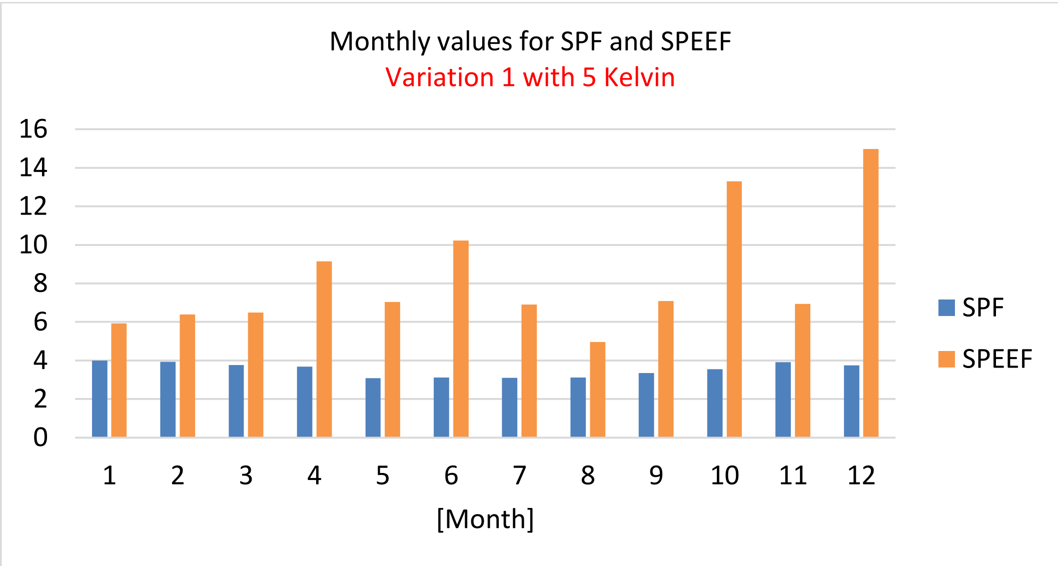

Figure 69 Monthly SPF and SPEEF GSHP variation 1 with 5 Kelvin

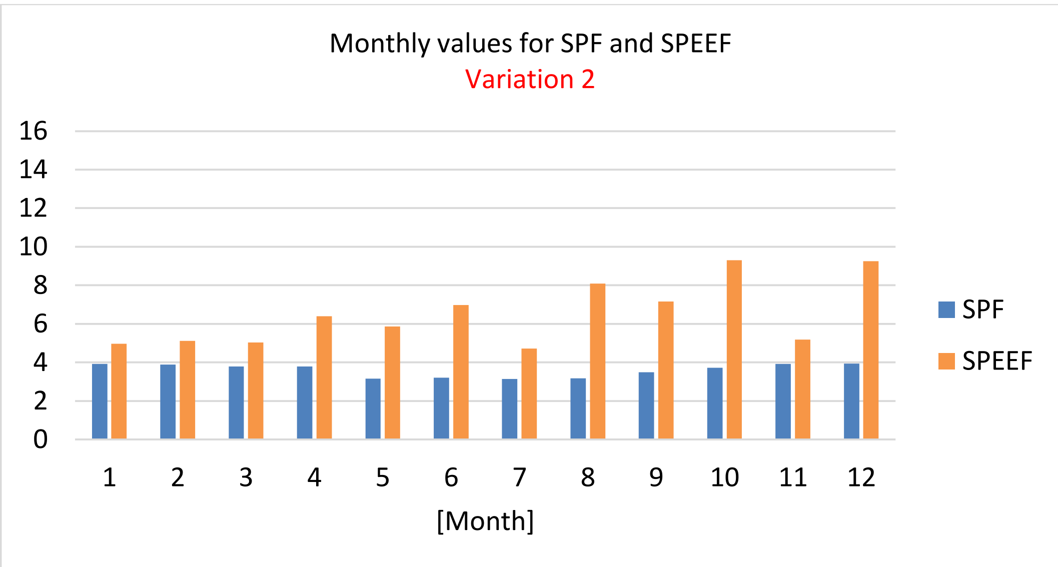

Figure 70 Monthly SPF and SPEEF GSHP variation 2

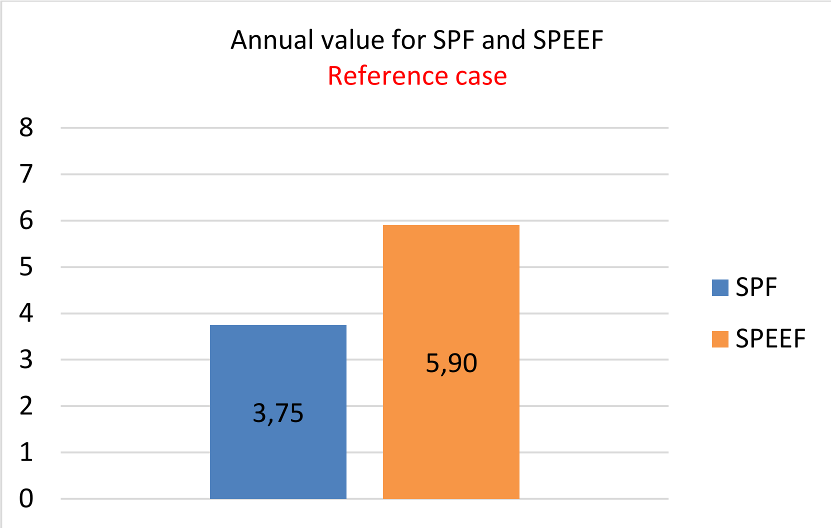

Figure 71 Annual SPF and SPEEF GSHP reference case

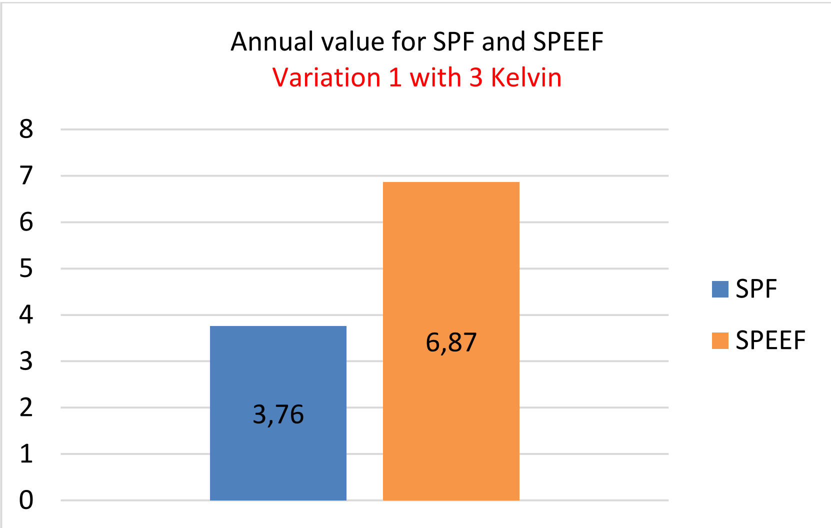

Figure 72 Annual SPF and SPEEF GSHP variation 1 with 3 Kelvin

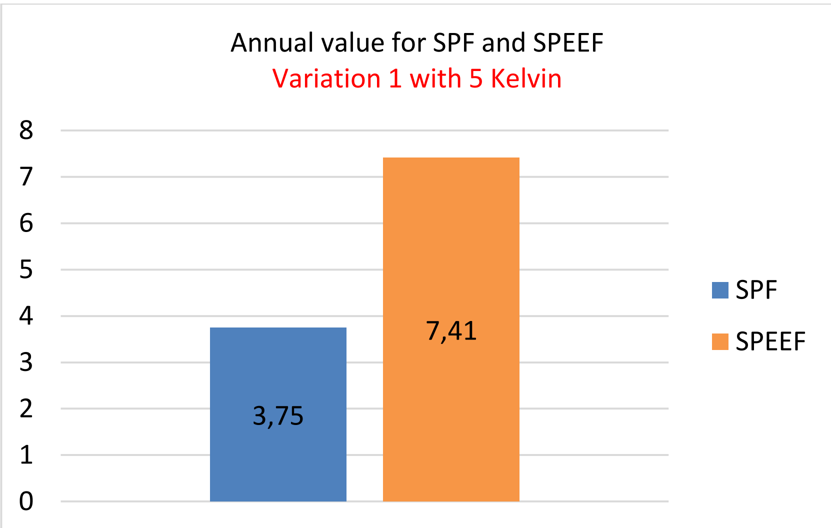

Figure 73 Annual SPF and SPEEF GSHP variation 1 with 5 Kelvin

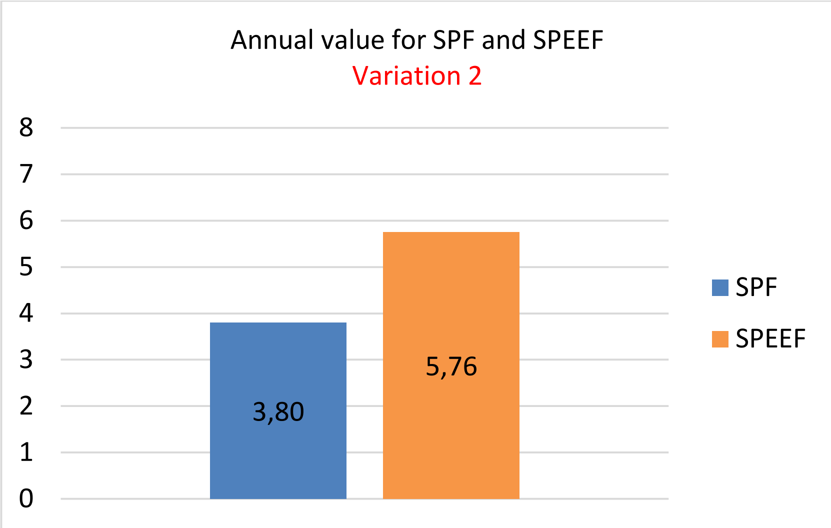

Figure 74 Annual SPF and SPEEF GSHP variation 2

The previous figures enable to show how the Seasonal Performance Factor is higher during the winter period (circa 3,9) and lower during the summer period (circa 3,1) when the heat pump is used only for providing the domestic hot water need. In fact during the production of domestic hot water the heat pump has to work with very high values of flow temperature, whereas when it works for satisfying the space heating demand it can work with lower temperatures because low temperature radiators require a lower level of temperature; it's also important to highlight that the cold source temperature (depending on the ground temperature) doesn't have big fluctuations during the whole year. Moreover it has been demonstrated how the application of Demand Side Management doesn't have any significant consequence in the annual value of the SPF. In fact in all the cases considered (from Figure 71 to Figure 74) it takes a value between 3,75 and 3,80. It means that considering the one year balance the heat pump consumes 1 electrical kWh in order to provide between 3,75 and 3,80 thermal kWh. But the most important parameter for considering the renewable energy consumption is the Seasonal Primary Energy Efficiency Factor. Comparing Figure 72 and Figure 73 with Figure 71 is clear that the application of Demand Side Management to the space heating is effective in order to increase the renewable energy consumption of the heat pump and so also increase the Seasonal Primary Energy Efficiency Factor. In fact its value is increased from 5,90 (reference case) up to 6,87 and 7,41 in the Variation 1 with 3°C and 5°C respectively. Furthermore the monthly value of SPEEF takes higher values during the months characterized by high renewable energy production such as June, October and December (as can be seen in Figure 1).

11.3.2 Air Source Heat Pump

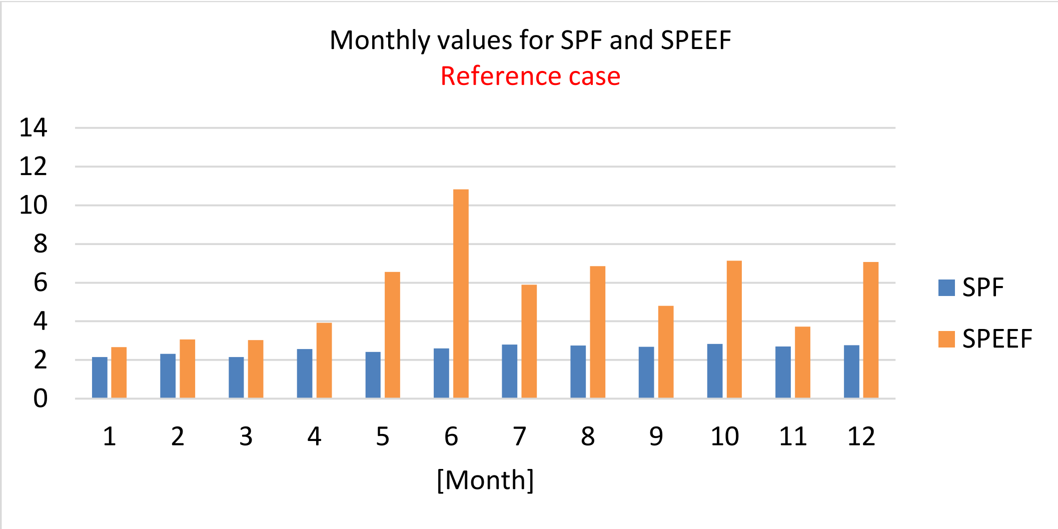

Figure 75 Monthly SPF and SPEEF ASHP reference case

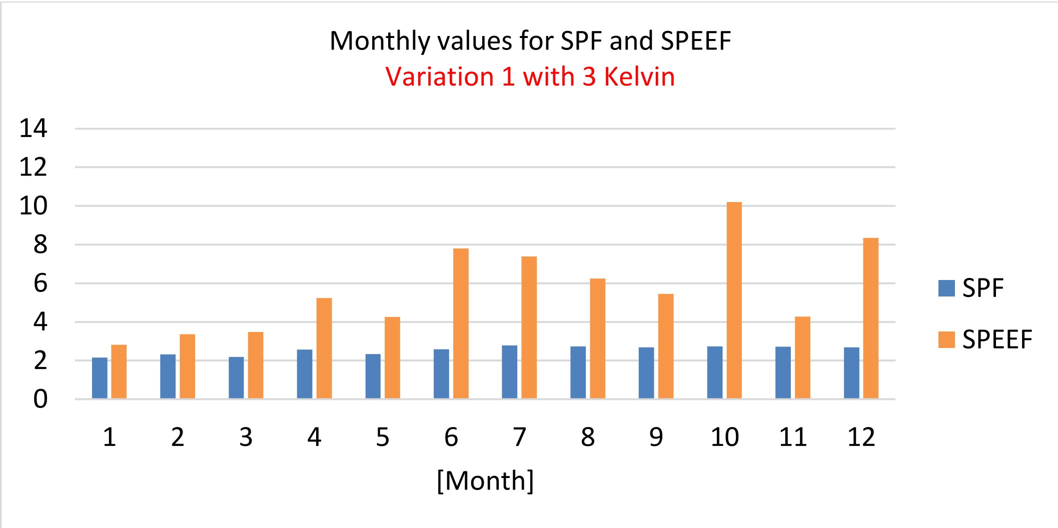

Figure 76 Monthly SPF and SPEEF ASHP variation 1 with 3 kelvin

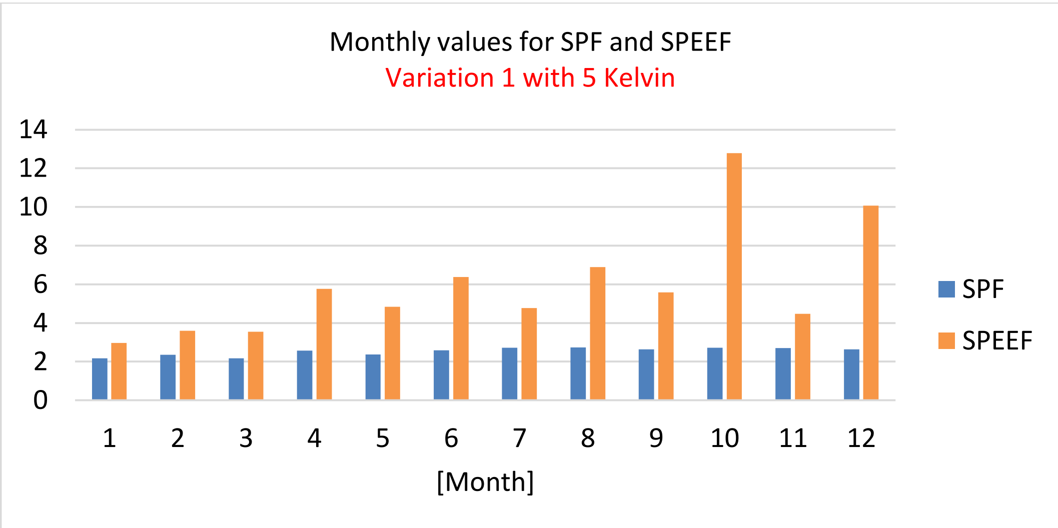

Figure 77 Monthly SPF and SPEEF ASHP variation 1 with 5 Kelvin

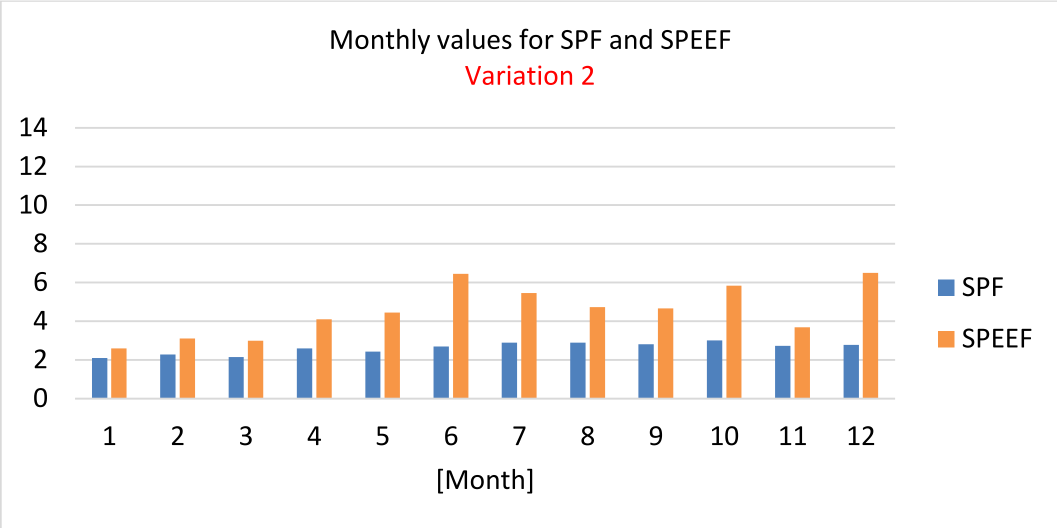

Figure 78 Monthly SPF and SPEEF ASHP variation 2

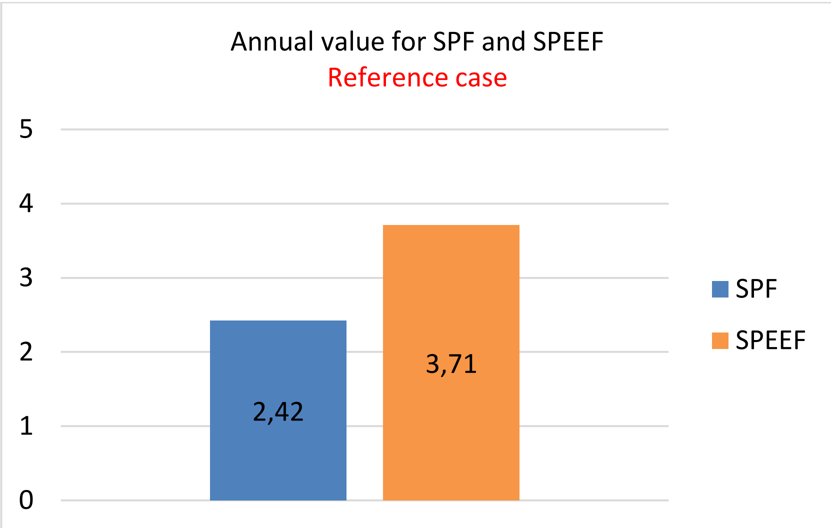

Figure 79 Annual SPF and SPEEF ASHP reference case

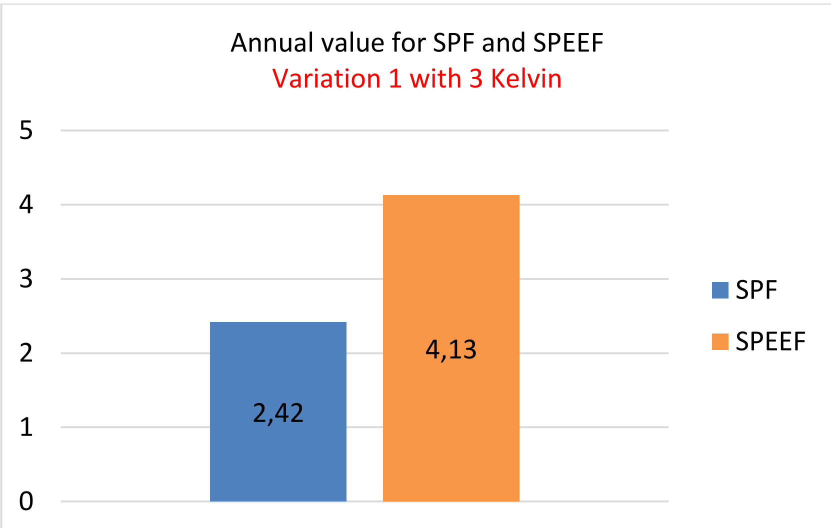

Figure 80 Annual SPF and SPEEF ASHP variation 1 with 3 Kelvin

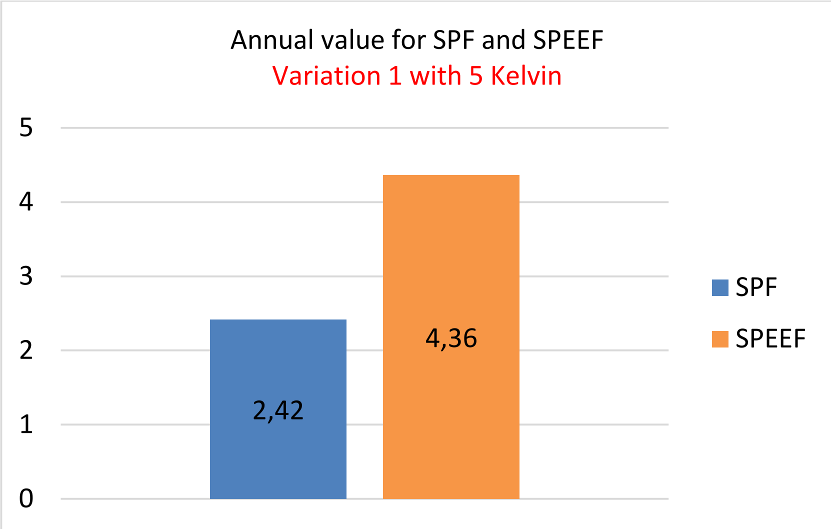

Figure 81 Annual SPF and SPEEF ASHP variation 1 with 5 Kelvin

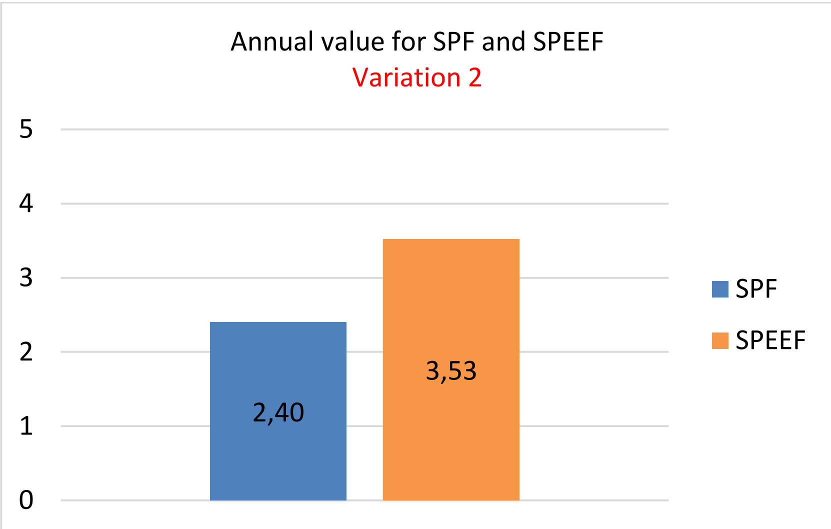

Figure 82 Annual SPF and SPEEF ASHP variation 2

Figure 75 to Figure 78 show how the Air Source Heat Pump efficiency is highly influenced by the cold and hot source temperatures. During the coldest months such as January and February it takes its lowest values (circa 2,3) because of the high difference between the cold source temperature (external air temperature) and the hot source temperature (for space heating and Domestic Hot Water production). In fact comparing the annual value of SPF with the one of Ground Source Heat Pump is evident the difference; for instance in the reference cases it is 3,75 and 2,42 for Ground and Air Source Heat Pump respectively. It's also interesting to note that the annual value of SPF doesn't depend so much by the kind of optimization chosen; effectively it takes value between 2,40 and 2,42 (from Figure 79 to Figure 82). Also in this case the application of Demand Side Management to the space heating causes an increase of consumption from renewable energy. In fact the SPEEF increases from 3,71 to 4,13 and 4,36 in the Variation 1 with 3°C and °C respectively. This increase is lower if compared with the results of the Ground Source Heat Pump.

11.4 Over Degree Temperature Factor (ODTF) and Under Degree Temperature Factor (UDTF)

In order to consider the thermal behavior of the building as consequence of the control strategies applied to the heat pumps, two parameters have been introduced for taking into account the phenomena of over-heating and under-cooling concerning the room operative temperature. In particular the results derived from the reference case were compared with the optimizations regarding the space heating. In fact the aim of this paragraph is to understand the effect of move the heating curve (by 3°C, 5°C and 10°C) in terms of over-heating and under-cooling. First of all the operative temperature is defined as the arithmetic mean of the room air temperature and the average radiant temperature. This parameter is more appropriate for describing the thermal comfort of the building and has been used as standard in the design of the buildings (Rossi, 2013). In this thesis a band of thermal comfort has been considered with room operative temperature between 20° and 22°C. For this reason have to be evaluated the Over Degree Temperature Factor and the Under Degree Temperature Factor defined during the heating period from September to April as:

$$ODTF = \sum_{\substack{heating \\ period}} \left(\Theta_{op} - 22\right) \; when \; \Theta_{op} > 22°C$$

Equation 27

$$UDTF = \sum_{\substack{heating \\ period}} \left(20 - \Theta_{op}\right) \; when \; \Theta_{op} < 20°C$$

Equation 28

Where:

- Θ_{op} is the room operative temperature, measured in °C.

These two parameters have been evaluated on the basis of room operative temperatures calculated as outputs by TRNSYS simulation. In particular have been considered only the most important rooms such as Living room (L), Guest room (G), WC (WC), Kitchen (K), Parent's room (P), Child 1 room (C1), Workroom (W), Child 2 room (C2) and Bathroom (B). The operative temperatures are related to the 15 minutes of Simulation time step; so that means that the terms indicated in brackets in the previous equations (Equation 27 and Equation 28) represents the over-heating or under-cooling contribute considering constant temperature in this time interval of fifteen minutes.

11.4.1 Ground Source Heat Pump

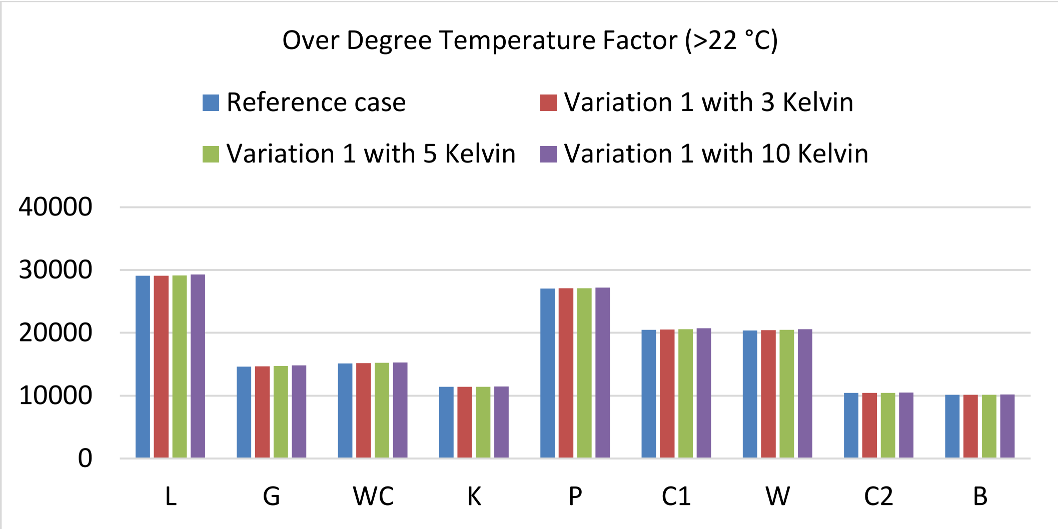

Figure 83 Over Degree Temperature Factor GSHP

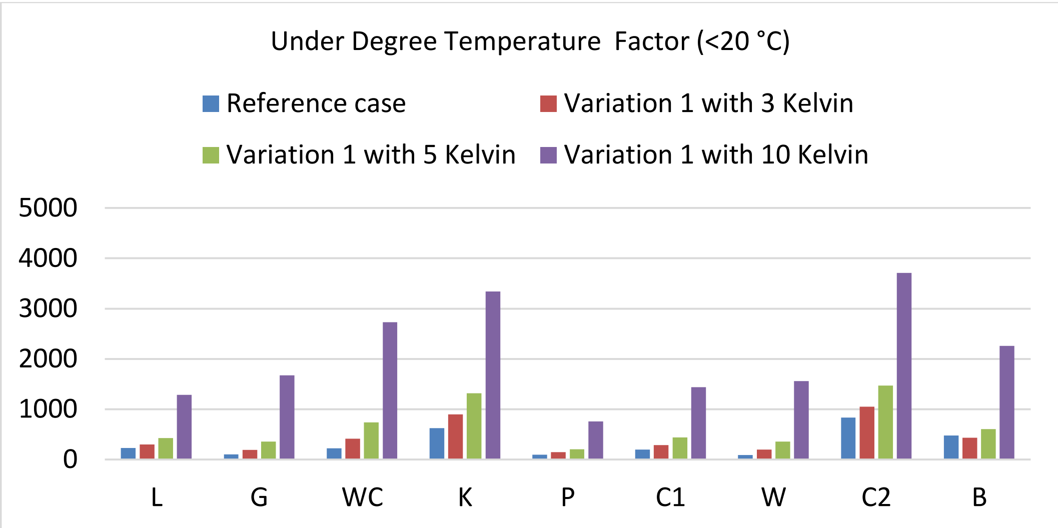

Figure 84 Under Degree Temperature Factor GSHP

11.4.2 Air Source Heat Pump

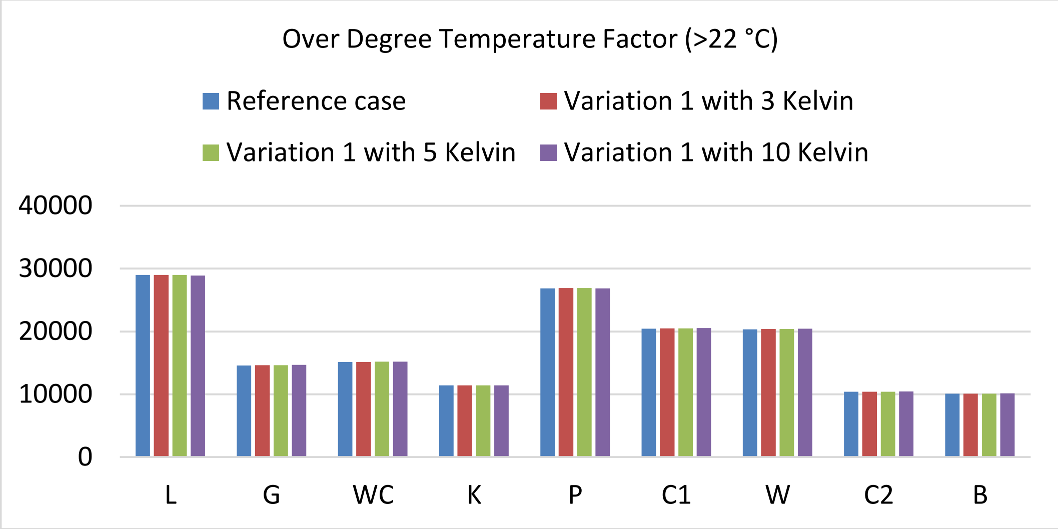

Figure 85 Over Degree Temperature Factor ASHP

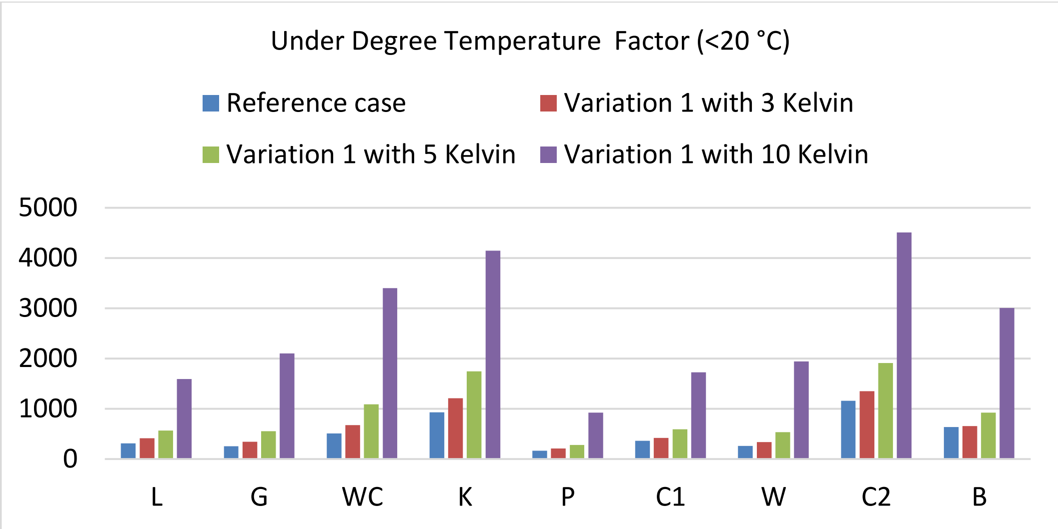

Figure 86 Under Degree Temperature Factor ASHP

Figure 83 and Figure 85 illustrate that the higher values of over-heating parameter are connected to the south facing rooms such as Living room, Parent's room and Child 1 room; this is certainly due to the solar heat gain that takes higher values for south facing rooms. At the same time the lower values of the Over Degree Temperature Factor are related to north facing rooms such as Kitchen, Child 2 room and Bathroom. Furthermore the same figures demonstrate how different moves of the heating curve don't have significant effect in the over-heating parameter. In fact the over-heating phenomena can be controlled by the use of the thermostatic valves. Figure 84 and Figure 86 enable to show the Under Degree Temperature Factor that is used as explanation for the under-cooling phenomenon. These figures demonstrate that the under-cooling, that is related to operative temperatures lower than 20°C, is more evident for north facing rooms such as Child 2 room and Kitchen.

Moreover, unlike the previous parameter, the different moves of the heating curve can provoke higher and more evident phenomena of under-cooling. In fact is shown how consider the shift of heating curve by +/- 10°C is excessive and risky in terms of thermal comfort for the building. It's for this reason that in the previous paragraphs were not considered the optimization named variation 1 with 10 Kelvin. Finally comparing Figure 84 with Figure 86 it's possible to appreciate that the under-cooling phenomenon is slightly more marked for the Air Source Heat Pump than for the Ground Source Heat Pump.

11.5 Room operative temperature

In paragraph 11.4 it has been clarified that the over-heating phenomenon is more marked for south facing rooms (Figure 83 and Figure 85) and the under-cooling is more evident for north facing rooms(Figure 84 and Figure 86). In particular was shown that these problems were more accentuated for Living room and Child 2 room respectively. For this reason it has been decided to depict the room operative temperature of these two rooms in order to quantify better the phenomena of over-heating and under-cooling. Also in this paragraph three cases of space heating optimization (variation 1 with 3°C, 5°C and 10°C) have been compared with the reference case. These results will be shown both for the Ground Source Heat Pump and Air Source Heat Pump. It should be noted that the optimal range for the room operative temperature hypothesized in this work is between 20°C and 22°C.

11.5.1 Ground Source Heat Pump

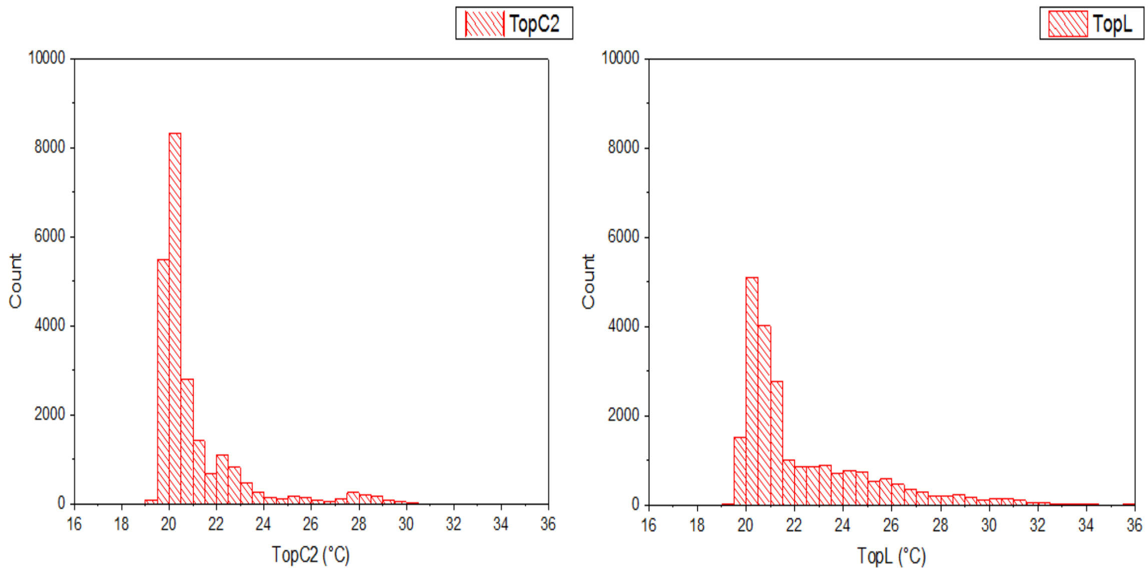

Figure 87 Operative temperature of two rooms GSHP reference case

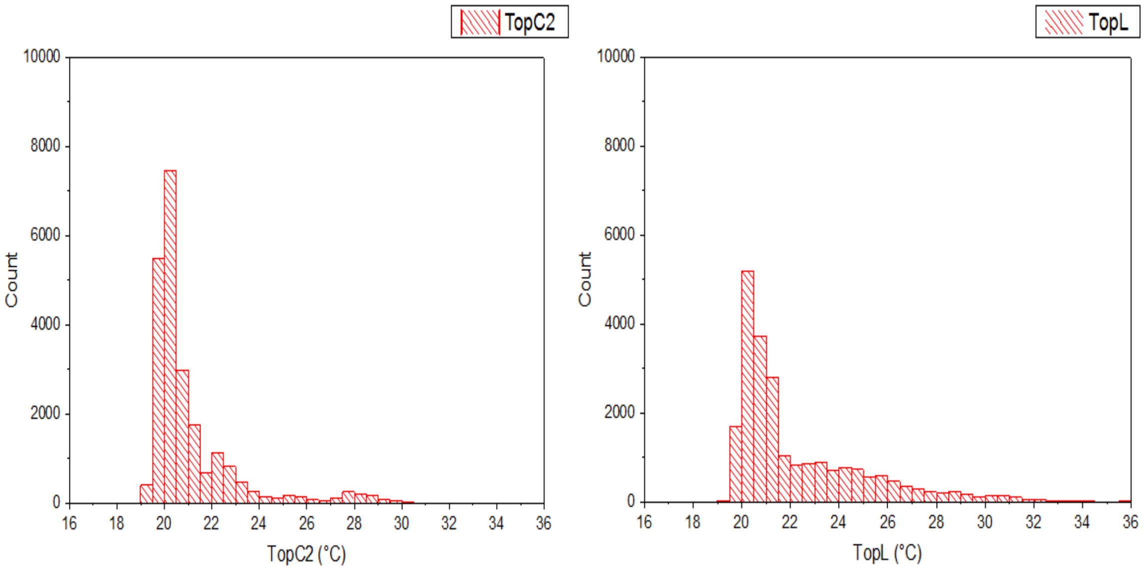

Figure 88 Operative temperature of two rooms GSHP variation 1 with 3 Kelvin

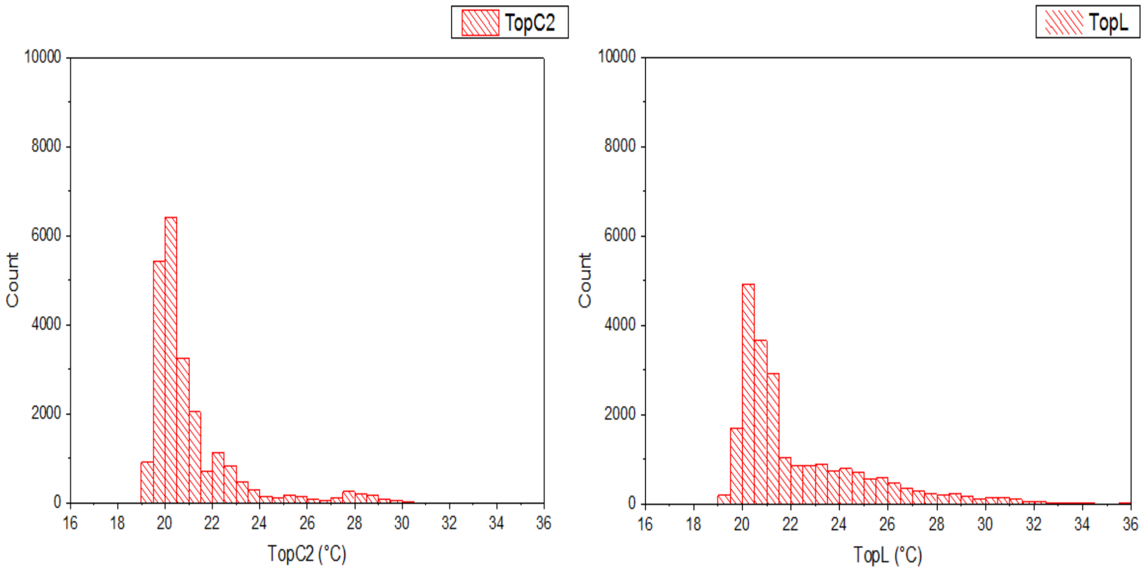

Figure 89 Operative temperature of two rooms GSHP variation 1 with 5 Kelvin

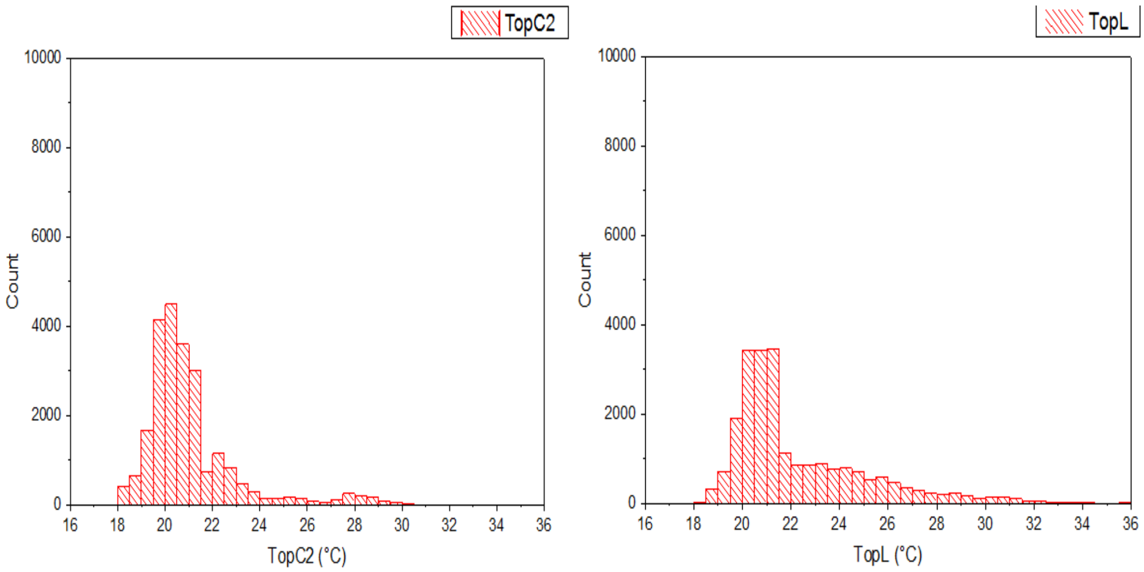

Figure 90 Operative temperature of two rooms GSHP variation 1 with 10 Kelvin

Figure 87 to Figure 89 enable to demonstrate that moving the heating curve up to 5°C has not a big influence in terms of under-cooling. In fact the room operative temperature never falls below 19°C for both Child 2 room and Living room and for most of the time it's higher than 20°C. At the same time the over-heating phenomenon is more marked for the Living room than the Child 2 room because of its orientation to the south. Furthermore in these figures it seems that for a considerable part of the heating season the room operative temperature is too high. But that owes to the fact that in months such as April, September and October the space heating demand takes a low value, as can be seen in Figure 49. On the other hand Figure 90 explains that the space heating optimization with 10°C may be too excessive because the operative temperature can reach values close to 18°C.

11.5.2 Air Source Heat Pump

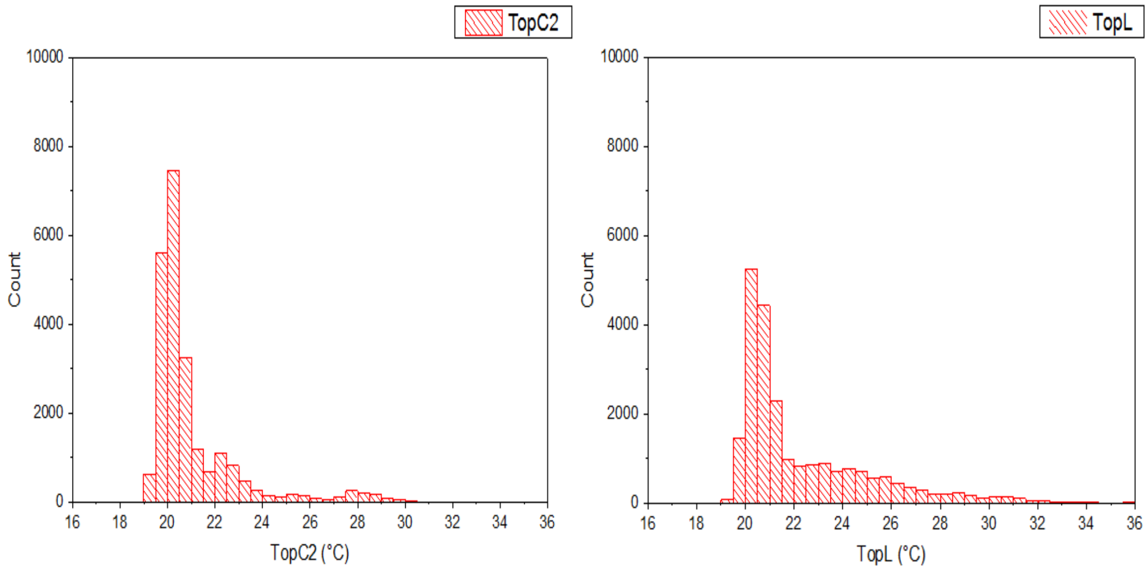

Figure 91 Operative temperature of two rooms ASHP reference case

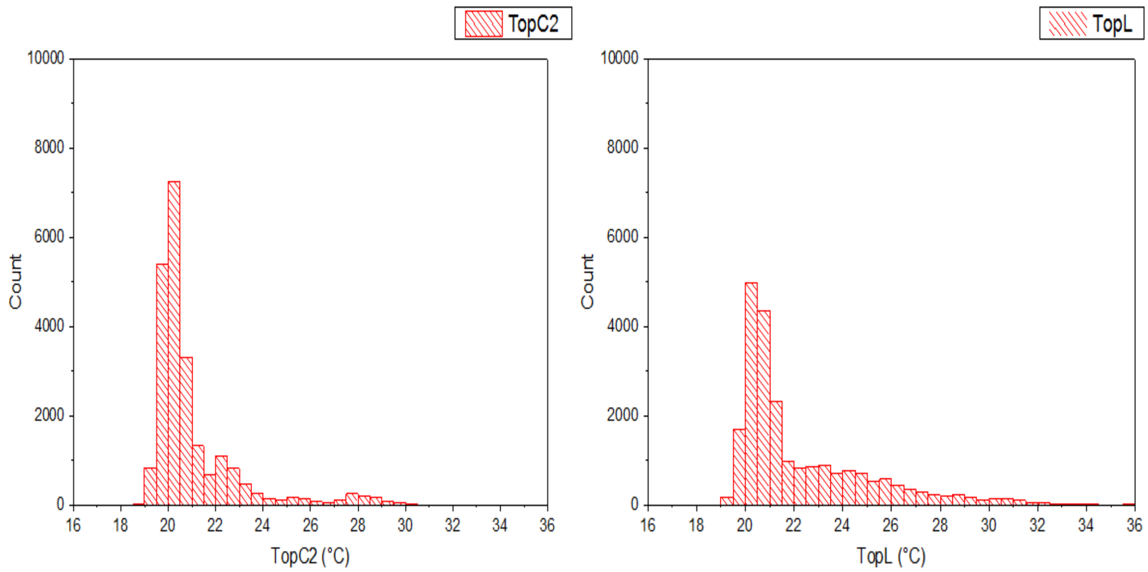

Figure 92 Operative temperature of two rooms ASHP variation 1 with 3 Kelvin

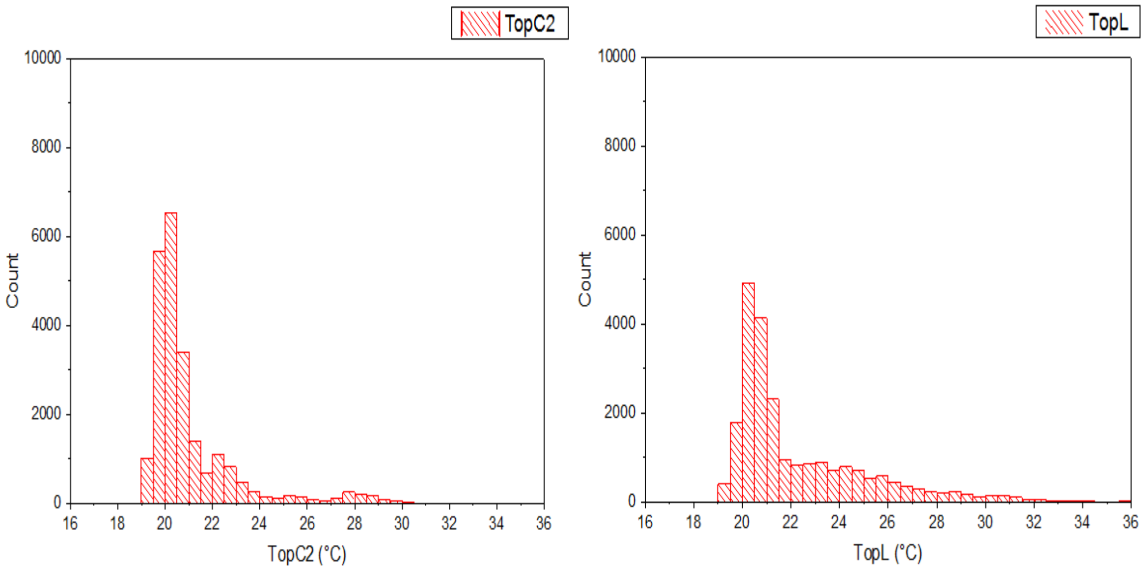

Figure 93 Operative temperature of two rooms ASHP variation 1 with 5 Kelvin

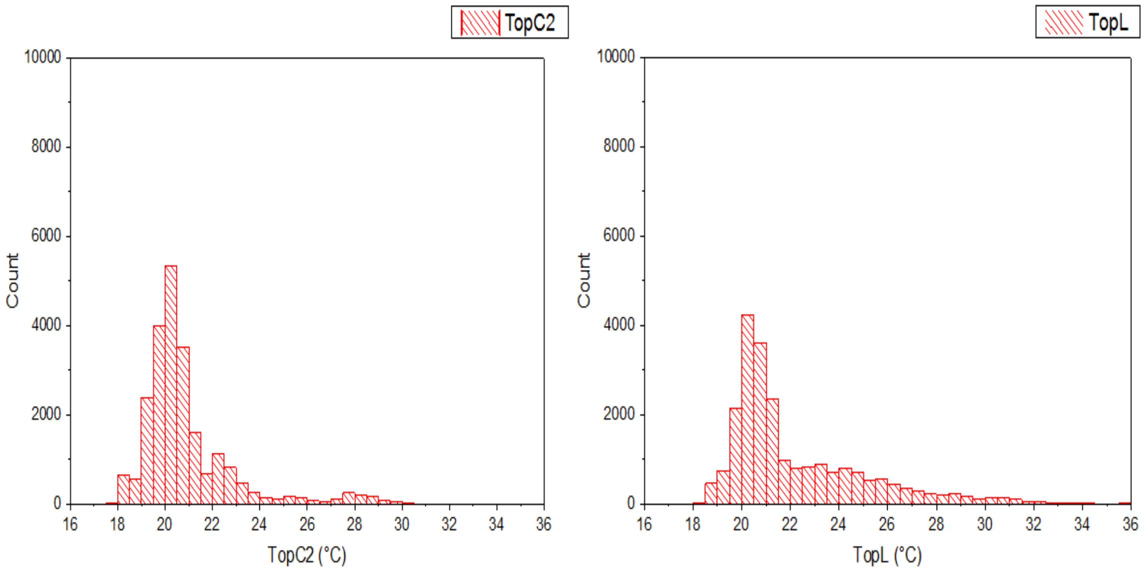

Figure 94 Operative temperature of two rooms ASHP variation 1 with 10 Kelvin

Figure 91 to Figure 93 show that the room operative temperature doesn't fall below 19°C; so, as for the case of Ground Source Heat Pump, the space heating optimization up to 5°C doesn't cause any problem regarding the thermal comfort of the building. Also in this case the over-heating phenomenon is more accentuated for the Living room because of its south orientation. On the other hand considering a heating curve shifted by 10°C provokes room operative temperatures also lower than 18°C. This is due to the fact that radiators work with too low flow temperature because when the Tariff is 4 the heating curve is decreased of 10°C. In fact it's important to remind that radiator manufacturers guarantee radiator operating only if the supply temperature is higher than 35°C (Vogel&Noot, 2015).

11.6 Load shifting potential

In paragraph 11.6 will be graphically shown how the Demand Side Management applied to the space heating allows the heat pump to work for less time characterized by Tariff 3 and Tariff 4. In fact raising the heating curve during time steps characterized by Tariff 1 and Tariff 2 permits to store heat in the buffer storage tank. This concept is better explained in Figure 95 and Figure 96 in which the optimization called “variation 1 with 5 Kelvin” is applied to the Ground Source Heat Pump:

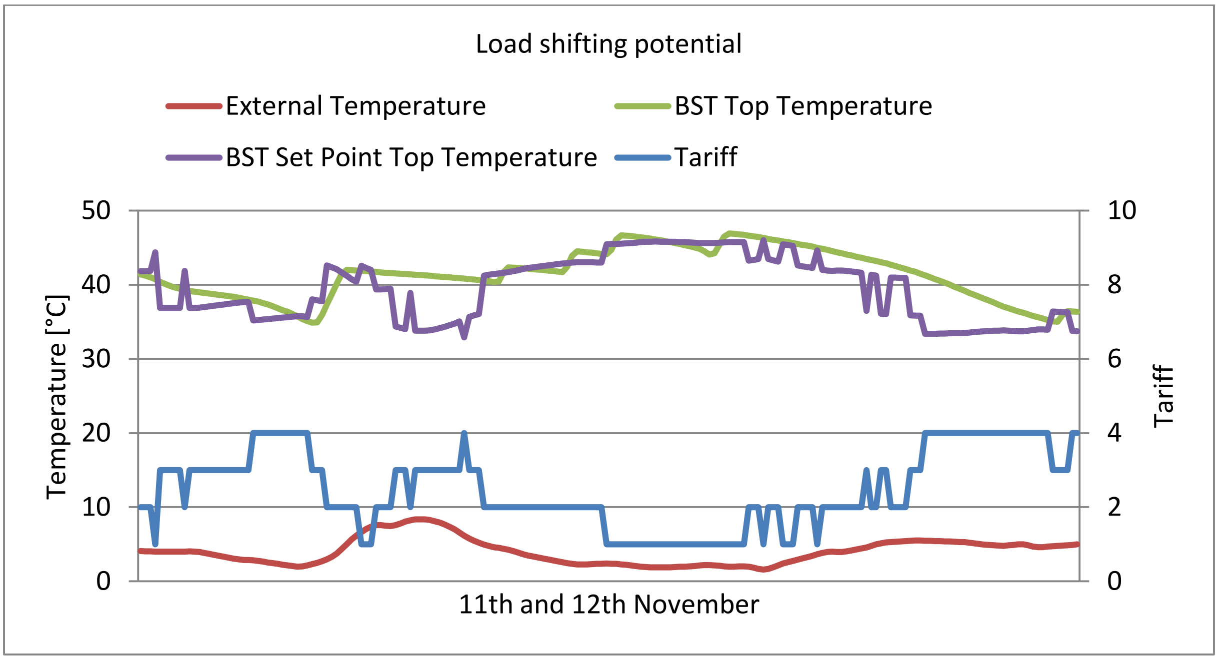

Figure 95 Load shifting potential (1)

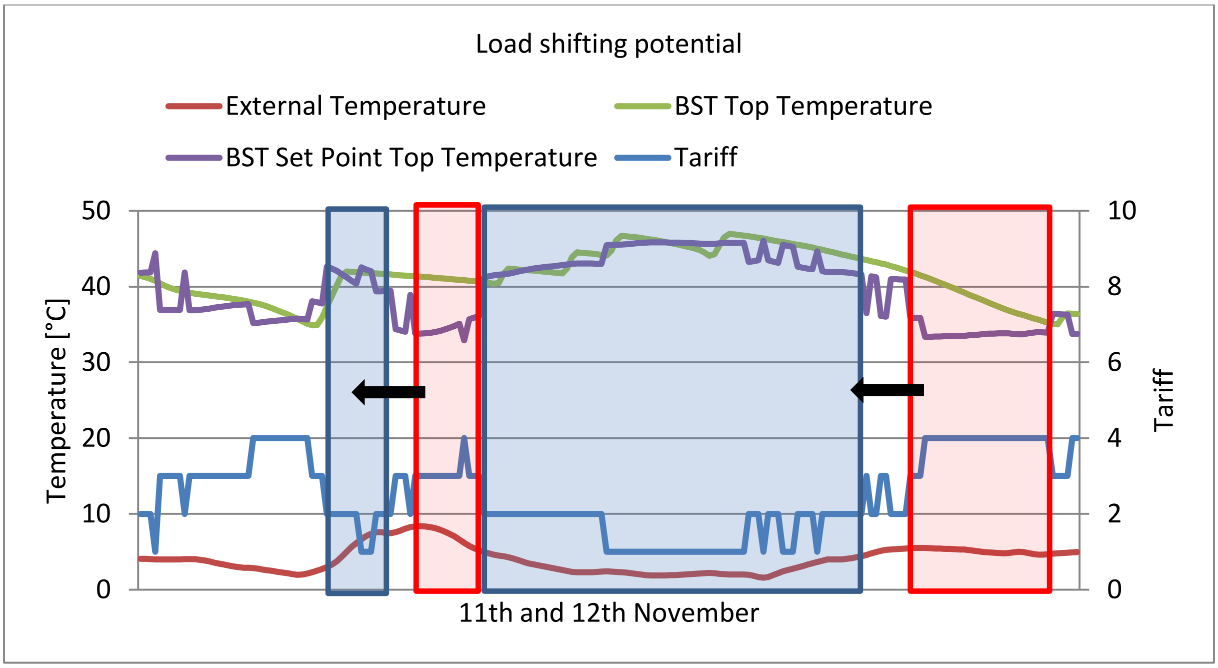

Figure 96 Load shifting potential (2)

The upper part of Figure 95 shows the temperature at the top of the buffer storage tank in the case variation 1 with 5°C applied to the ground source heat pump during two typical days of the winter period (11th and 12th November). In particular the following parameters may be distinguished:

- External temperature (red line). This parameter has been used for building up the heating curve;
- Buffer storage tank top temperature (green line). This is the real value measured by the temperature sensor placed in the buffer storage tank and it is used as control variable for the heat pump working for the space heating;
- Buffer storage tank set point top temperature (purple line). This value in the variation 1 depends not only on the external air temperature (weather compensation), but also on the Tariff as explained beforehand in Chapter 10. In particular when the Tariff is 1 and 2 the heating curve is increased by 5°C and 2,5°C respectively; on the other hand when the Tariff is 3 and 4 the heating curve is decreased by 2,5°C and 5°C respectively;
- Tariff (blue line). This parameter depends on the instantaneous value of the residual load as already mentioned in Chapter 2.

Figure 96 explains the concept of load shifting. In fact applying Demand Side Management to the space heating allows to use less the heat pump when the Tariff is 3 or 4. In fact when the Tariff is 1 or 2 the heating curve of the heating system must be increased (blue rectangles) and the heat pump works until the buffer storage tank top temperature reaches its set point value. After that when the Tariff is 3 or 4 (red rectangles) the buffer storage tank top temperature takes higher values than its set point so there is no need to use the heat pump for the space heating. Furthermore that has no negative consequences on the thermal comfort of the building as shown in the paragraph 11.5 regarding the operative temperatures.

11.7 Conclusion

In this thesis it has been demonstrated that the application of Demand Side Management to heat pumps allows to exploit better the renewable energy production of this city situated in Northern Hessen, Germany. In particular the optimization applied to the space heating permits to achieve important increases of the electricity consumption characterized by Tariff 1 and Tariff 2 if compared to the normal control of the heat pump. This phenomenon is more accentuated for the Ground Source Heat Pump because it’s possible to cover 49% of the annual electricity demand of the heat pump considering the optimization called variation 1 with 5 Kelvin. The same case considered for the Air Source Heat Pump also allows to cover 45% of the annual electricity demand for providing space heating and domestic hot water

need. It’s also important to underline that the Demand Side Management applied to the space heating should be used with some precautions. In fact it’s risky in terms of thermal behavior of the building consider optimization of the space heating that involve decreases of the heating curve higher than 5°C. In fact it has been shown that the optimization named variation 1 with 10 Kelvin provokes dangerous phenomena of under-cooling for the operative room temperature both for the Ground Source Heat Pump and the Air Source Heat Pump. In this cases the room operative temperature falls also below 18°C, while it should be higher than 20°C. On the other hand any negative consequence occurred as regards the over-heating phenomenon because the room temperature can be controlled through thermostatic valves. Furthermore it has been proven that the optimization considered for the Demand Side Management applied to the domestic hot water production causes only the increase of the overall electricity demand for supplying heating and domestic hot water need.
Moreover one of the most important result is that the use of the buffer storage tank in combination with the space heating optimization allows also to exploit the phenomenon of load shifting. In fact raising the heating curve when the renewable energy production is higher than the household electricity demand permits to use more the wind, solar and biomass production of the city. In fact the heat can be stored inside the tank and used later for heating the building through the low temperature radiators. So it’s possible to decouple the electricity and the heat demand of the building.
In addition it has been demonstrated that the Ground Source Heat Pump works with a higher annual SPF than the Air Source Heat Pump. For instance in the reference cases it is 3,75 and 2,42 for GSHP and ASHP respectively; this is due to the fact that these two systems use different cold sources. Finally it has been illustrated that the domestic hot water need is increasing its relative importance in new and well insulated buildings such as the single family house considered in this study. In fact the ratio between the annual heating demand and the domestic hot water need is 2,68.

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