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Doctoral course in Energetics - Cycle XXVII

*Wood biomass CHP in district heating systems: simulation and operation analysis*


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*He must be very ignorant for he answers every question he is asked.*

*Voltaire*

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# Abstract

This research work is focused on the development of a simulation model for the operational analysis of wood biomass CHP (combined heat and power) units supplying district heating systems. The integrated approach that has been adopted offers the possibility of considering the effect of each system component during the different operation conditions that can occur during the year. The aim of the model is to provide a support for different situations: the design of the components of the system, the analysis of real operation to match the requested performance and the local energy planning considering the effects of the actual behaviour of those systems throughout the year.

The model has been improved thanks to an analysis of real operation data, both on demand side and on supply side. The heat demand from different district heating systems has been investigated, by analysing the main differences and analogies with respect to size and climate conditions. Considering the supply side, two different ORC systems have been analysed over some years of operation with an hourly time step. These data analysis has pointed out the significant variations that can occur in DH systems when the actual operation conditions are different than the ones forecast in the system design.

The model has been used for a case study analysis to assess the optimum size of a CHP unit coupled to a heat storage system in an existing district heating network. An economic analysis has been performed in order to evaluate the current Italian incentive framework for RES (renewable energy sources) plants. A difference has been found between the optimal energetic layouts and the best economic solutions, showing that the current incentives still not promote the most efficient solutions for energy production from wood biomass.

## Keywords

Wood Biomass

Combined Heat and Power (CHP)

Organic Rankine Cycles (ORC)

District Heating (DH)

# Contents

# List of Figures

# List of Tables

# Summary

The use of wood biomass for energy production faces specific issues, resulting in slightly different plant characteristics than traditional fossil fuel plants. The electricity production deals with a low conversion efficiency, due to the use of a poor quality fuel. The main solution that can be used to increase the performance of a wood biomass system is the combined heat and power (CHP) production. For a sustainable use of local biomass that can be obtained from forests, these systems are generally small or medium sized (lower than 10÷20 $MW_{el}$) with respect to traditional fossil fuel systems. The typical applications are single industrial users and small district heating systems.

The operation conditions have a wide range of variability, as multiple parameters can affect the performance of biomass-to-energy conversion (e.g. wood biomass quality, heat demand variations, outdoor conditions, etc.). As a consequence, an energy analysis that is able to consider the different situations occurring during the operation can provide a better description of the actual performance of the system, which can significantly differ from the design performance. The knowledge of the real performance of a system is fundamental both for monitoring the existing plants and for an improved design of the future plants.

The main objective of this work is to perform an analysis on small and medium size district heating systems supplied by wood biomass CHP units. This PhD work has been developed in the framework of a wider research activity focused on energy production from wood biomass, carried out by *Sustainable Energy Analyses* research group at Politecnico di Torino.

The simulation model developed for this analysis is based on thermodynamic simulations improved using the data obtained through operation analyses of CHP units in district heating systems. The aim is to provide an integrated simulation tool, considering each component of the system with its characteristics of performance, parameters and operational logics. The need of an integrated approach lays in the strict relation between the components of the system (e.g. CHP units, boilers, heat storage systems, etc.), which influence each other.

This simulation model can be a useful support tool in different situations: in the design phase of biomass-to-energy CHP systems, in the operational phase for a performance evaluation and in the planning phase for considering the effect of different incentives on the biomass DH systems performance. The possibility of using the same approach in different phases allows a better comparison of the results and the possibility of verifying the differences between the expected and the actual performances.

This thesis illustrates in detail the main features of each aspect of the simulation model, considering both the demand side and the supply side of the DH system. The main hypotheses will be discussed,

along with the choices that have been needed for the definition of the model. A synthesis of the contents of each chapter is provided below.

Chapter 1 provides a general introduction to the themes considered in this work, with a particular focus on wood biomass and on energy conversion technologies. In the wide range of available conversion technologies, with different degrees of maturity, this work has been focused on Organic Rankine Cycles (ORC) technology, which is currently the standard for small and medium CHP systems. Nevertheless, the methodology can be extended to other technologies, taking into account their particular behaviour.

Chapter 2 is focused on district heating networks. These systems currently represent a significant share of the total residential heat consumption in many European countries. Their main characteristics have been considered, pointing out the different aspects that can affect their performance in the heat supply to the user. A case study has been analysed in detail, considering hourly demand data in order to build a consistent dataset to be used as a reference for the simulation model.

In Chapter 3 the simulation model of the ORC unit is described in detail. This part of the model is the core of the simulation tool, as the CHP is often providing the largest amount of energy through the year. The simulation has been focused on ORC units, which are currently the commercial standard for small and medium applications. A description of the characteristics of the fluids and the system configurations allows to focus on the main aspects to be taken into account for simulation and performance analysis.

The application of the simulation model to real ORC systems is described in Chapter 4. An operational analysis of two ORC case studies has been performed in order to evaluate two different cases: a small size innovative ORC unit and a medium-size ORC more representative of the standard size used in the wood biomass sector. Both analyses provide useful information on the real behaviour of those kind of systems, which can be significantly different from the one expected in design conditions.

Chapter 5 considers the other system components which are integrated into the heat production facility. The main components are the biomass boiler, which can be either coupled to the ORC unit or used as integration and backup boiler, and the heat storage systems, which are useful for the optimisation of the energy production through CHP.  Other systems can be added to the plant in order to integrate additional RES technologies, such as thermal solar panels or biomass dryers.

Chapter 6 provides a more general example of application of the model for planning purposes: a general analysis on wood biomass CHP plants is performed to evaluate the impact of the current Italian incentive framework. This example shows the advantage of having a simulation model for the comparison of economic and energy performance of an energy system, as a tool for defining sensitivity analyses on multiple parameters (e.g. biomass cost, electricity and heat price, conversion efficiency, etc.).

# Chapter 1 Introduction to wood biomass CHP systems

This research work is focused on small and medium size CHP plants (with nominal output power lower than 2 $MW_{el}$), working on wood biomass. The most diffused biomass used for CHP generation is the chipped wood, however the same approach presented in this work can be extended to other types of biomass, which are often used in larger systems or in particular situations (e.g. herbaceous biomasses, coal co-combustion, pellets, etc.). The chipped wood provides the possibility to exploit local resources in a sustainable way, with environmental, economic and social advantages. A proper forest management allows the best use of wood resources, increasing the local resources and jobs.

This first chapter will present an introduction to the current use of wood biomass in the world, as well as the main CHP technologies used for biomass to energy conversion.

## 1.1 Wood biomass use in the world

Currently, biomass is covering around 10% of the world primary energy supply with around 1,344 Mtoe (source IEA, [1]). The traditional uses, heating and cooking in the developing countries, reach 648 Mtoe, whereas 454 Mtoe are used for electricity and heat production in power plants, 339 Mtoe are supplied to the industry sector for energy transformations and the remaining part is used in other transformations, mainly for biofuels production.

Wood biomass is the oldest source of energy in mankind, mainly used for heating and cooking purposes in fireplaces. The first evidence of a controlled use of fire appeared in Kenya about 1.4 million years ago, while the oldest known hearths in Europe have been built around 500,000 years ago [2]. Chinese and Romans used chimneys since 2,000 years ago, in some cases also connected to central heating systems (developed in Rome to carry smoke from hypocausts). In the following centuries wood has been extensively used as primary energy source, before it has been substituted by coal in many industrial applications in $19^{th}$ Century.

In Europe, the final energy consumption of bioenergy reached 102 Mtoe [3], roughly 9% of the total final energy consumptions [4]. About 74% is represented by bio-heat and derived heat (half of it in the residential sector), 12% for electricity and 14% for transport biofuels. The bioenergy final energy consumption doubled from 2000 to 2012, and the NREAPs projections (European countries' National Renewable Energy Action Plans) for 2020 expect about 139 Mtoe from bioenergy. Biomass currently accounts for 88.9% of renewable final consumption in heating and cooling, and 18.7% of renewable electricity generation (of which 65% is produced in CHP plants). Despite the increase of biomass consumption, there is still a significant potential in Europe for different wood uses, as the net increment in forest available for wood supply is around 289 million $m^3$ per year, resulting from the net growth of the forests.

The European targets set by the 2020 climate and energy package [5] are currently driving a significant rise in the use of renewable sources. Considering wood biomass, this has led to an exponential increase of pellet consumption, both for heating and for power production. In 2013 the EU28 countries reached a total consumption of 18.3 million tonnes of pellet, whereas the production accounted for 12.2 Mt. The pellet consumption in Europe reached almost 80% of the total world consumption, and the shortage of pellet is mainly supplied from North America, Asia and Russia. The main European importing countries are United Kingdom, Denmark and Italy. This phenomenon should be taken into account when evaluating the benefits of the increase of energy production from renewables, as a sustainability analysis should be performed considering all the energy chain.

## 1.2 Wood biomass characteristics

Wood biomass is an organic material, a natural composite of cellulose fibres (which are strong in tension) embedded in a matrix of lignin which resists compression. It has been used for thousands of years both for energy purposes and as a construction material. The main energy conversions that will be considered in this work are the thermochemical transformations (combustion, pyrolysis and gasification), while other applications are being under development for the chemical transformation of wood biomass into other biofuels.

### 1.2.1 Composition

The major constituents of wood biomass are:

- cellulose ( $(C_6H_{10}O_6)_n$, polycondensation product of glucose molecules): it represents about 40 ÷ 45% of the weight of the biomass on a dry basis, it has a supporting function and confers mechanical strength to the structure of the plants;

- hemicellulose (composed of a complex of polysaccharides with a different chemical composition than cellulose): constitutes about 20 ÷ 35% of the weight of plant biomass on a dry basis and has the function of cementing substance of the lignified parts of the plants;

- lignin ($C_{40}H_{44}O_6$): it corresponds to approximately 15 ÷ 30% of the weight of the biomass on a dry basis, and gives rigidity to the structure;

- extracts: they consist of a wide variety of compounds (resins, waxes, fats, oils, starch, sugar, tannin substances and pigments, etc.) and may be present in the biomass in a quantity not negligible;

- inorganic material: it is indicated as a fraction of ashes and consists primarily of alkaline species (Na, K, Mg, Ca), with traces of heavy metals (Cd, Zn, As, Pb, Cu, etc.) and other elements such as S, Cl, N, P, Si, Al, etc.

The composition of the biomass used for energy production in thermochemical conversion processes is usually described in terms of water content, volatile matter, fixed carbon and ashes.

The laboratory analysis used for the determination of these fractions is known as proximate analysis, and consists of a series of experimental analyses that allow to separate and measure the different fractions included in the fuel.

The water content is the amount of water present in the biomass, which can be linked to the structure of the biomass itself (intrinsic water content) or determined by the external environmental conditions (extrinsic water content). It is usually measured in laboratory tests by calculating the loss of mass of a sample subjected to drying under standard conditions at 105°C (as required by the standard UNI EN 14774).

The volatile matter is the fraction of biomass composed primarily of carbon, hydrogen and oxygen and of minor fractions of nitrogen and sulphur, which evaporates during the phases of heating and devolatilization during the gasification of the fuel. The volatile matter is the major fraction of the biomass: it is typically from 70% to 86% by weight on dry matter for woody biomass.

The fixed carbon is the fraction of carbon that does not volatilize during the heating phase and is oxidized via heterogeneous reactions.

The ashes are the remaining solid fraction, which consists of the inorganic material (mainly salts and minerals) that are present in the biomass. The ashes are usually determined as the inorganic material that remains after the combustion of a sample of biomass at 550 ° C (as required by the standard UNI EN 14775).

An example of proximate analysis for some biomass plant is reported in Table 1.1, compared to the characteristics of bituminous coal.

| | **Volatile matter [% d.b.]** | **Fixed carbon [% d.b.]** | **Ashes [% d.b.]** |
|---|---|---|---|
| **robinia** | 80.94 | 18.26 | 0.80 |
| **poplar** | 82.32 | 16.35 | 1.33 |
| **eucalyptus** | 82.55 | 16.93 | 0.52 |
| **bituminous coal** | 33.12 | 48.18 | 18.70 |

Table 1.1 Proximate analyses for some wood biomasses and coal (in weight % on dry basis) [6]

The composition of the wood biomass can also be expressed in terms of the percentage content of carbon, hydrogen, oxygen and nitrogen (ultimate analysis), which are the main elements constituting the organic part of the biomass. The ultimate analysis of the biomass gives information on its energy content (depending on the percentage of C and H), the attitude to the combustion ratio (C / N), and the formation of pollutants in the combustion products (N, S and Cl). The woody biomass has around 50% of carbon content, 43% oxygen and 6% hydrogen (percentages by weight, on dry basis).

### 1.2.2 Moisture content

The water content of the biomass can usually be referred to as received on dry basis (H) or as received on wet basis (M), as defined in the standard UNI 17225. In this work the moisture content will be usually expressed as received on wet basis.

The moisture content of the solid biomass that has not been artificially dried varies between approximately 60% and 15% depending on the type of biomass, the length of the storage period and the mode of storage. The calorific value of the biomass is highly variable depending on the water content. The amount of water contained in the biomass, however, influence not only the calorific value, but also the combustion conditions inside the heat generators.

In the case of very high water content, thermochemical reactions have difficulty to be activated and cannot reach the temperatures necessary to ensure that the transformation of reactants into products is done in a rapid and complete process. This drawback could be overcome by the pre-drying of the fuel before the feeding of the heat generator.

In combustion systems, it is necessary to ensure an adequate residence time in the combustion chamber in order to facilitate the water evaporation. This is possible by adopting technological solutions and ad-hoc plant configurations. The higher the water content of biomass input the lower the temperatures inside the generator, with consequent increase of the residence time necessary for the completion of the chemical reactions of combustion. A high moisture content also increases the total volume of the gases produced from the process and decreases the combustion efficiency.

The moisture content has a crucial importance in gasification plants, where the termochemical process requires dry biomass (usually around 10% on wet basis) and has a low flexibility with respect to moisture variations. As a result, to ensure the conditions of chemical equilibrium of the process, it is necessary to provide a controlled drying phase.

### 1.2.3 Ash content

The ashes are constituted by the inorganic material that is present in the biomass. The inorganic constituents may be inherent to the biomass or accidentally included, or may be incorporated through the various stages of processing, in particular during the collection phase.

The most abundant elements are usually calcium, potassium, magnesium and phosphorus, present as $CaO$, $MgO$, $Na_2O$, $K_2O$, $P_2O_5$, $SiO_2$, $Al_2O_3$, $Fe_2O_3$ and $MnO$. The composition of the ashes, however, is variably depending on the type of biomass (also with significant differences from species to species), the part of the plant, the soil conditions, the possible use of fertilizers, etc.

The solid residue that is found downstream of the thermo-chemical process of biomass transformation, as a deposit in the lower part of the reactor or entrained in the flue gases, is a function of the ash present in the starting biomass, in terms of quantity, composition and characteristics (melting behaviour, etc.), and the operating conditions of the device.

In general, a low melting point of ash is related to an abundance of the alkaline elements, with the risk of slag formation and deposits. The woody fuels have a basic metals content in the ashes much low (except for bark) and they usually not show any problems related to the sintering and fusion of the ashes. In combustion of herbaceous biomass (with a melting point of the ash of about 1,000 °C) and grains of cereals (melting point <750 °C) instead, the slag formation is common. Table 1.2 reports the values of ash content and ash melting temperature for some biomass types.

| Typology | Ash content [% db] | Ash melting temperature [°C] |
|---|---|---|
| **spruce (with bark)** | 0.6 | 1,426 |
| **beech (with bark)** | 0.5 | 1,340 |
| **poplar (short rotation forestry)** | 1.8 | 1,335 |
| **willow (short rotation forestry)** | 2.0 | 1,283 |
| **bark of conifers** | 3.8 | 1,440 |
| **vine wood (wood chips)** | 3.4 | 1,450 |
| **miscanthus** | 3.9 | 973 |
| **wheat straw** | 5.7 | 998 |
| **triticale grains** | 2.1 | 730 |

Table 1.2 Ash content and melting temperature for some biomass types [6]

### 1.2.4 Heating value

The heating value of the fuel is the maximum heat that is released by its complete oxidation in adiabatic conditions by bringing all the products of combustion back to the original pre-combustion temperature, and in particular condensing any vapour produced. The heating value is usually expressed in [kJ/kg], [kWh/kg] or [kcal/kg].

There is a distinction between the higher heating value (HHV) and the lower heating value (LHV), depending on the physical state of the water in the combustion products. The higher heating value refers to the case of complete condensation of the steam contained in the flue gases. On the other hand, the lower calorific value refers to the case where all the water in the combustion products is in the vapour state, which is the typical operation condition of the biomass-fired heat generators.

Some condensing biomass boilers are currently available on the market, which allow to condensate the water in the flue gases, and in which it is therefore possible to recover an additional portion of energy from the flue gases. Such operation mode is possible only in low temperature water systems, like floor heating (when the fluid return temperature is lower than the condensation temperature of the water vapour contained in the flue gases), and paying particular attention to the problems of corrosion in the heat exchangers.

In the case of biomass it is important to pay particular attention to the reference conditions of the heating value, which can be stated on the dry basis (anhydrous biomass) or on wet basis. In the latter case, the biomass moisture content need to be specified.

The determination of the gross calorific value can be measured directly using a bomb calorimeter according to a standardized procedure (defined in UNI EN 14918). The heating value can be predicted from the ultimate analysis of the biomass with the following equation [7]:

$$HHV\ \left[\frac{kJ}{kg_{d.b.}}\right] = 0.3491\, X_C + 1.1783\, X_H + 0.1005 X_S - 0.0151 X_N - 0.1034 X_O - 0.0211 X_{ash} \quad (1.1)$$

Where HHV is the higher heating value and $X_y$ is the mass fraction of the element y. The average absolute error of prediction is 1.45%, which is the lowest among the correlations proposed in literature [7].

To get the biomass lower heating value, which is given by the higher heating value minus the energy consumed during the condensation of water and the heat released by the combustion of the hydrogen content of the biomass, the following equation can be used:

$$LHV\ \left[\frac{kJ}{kg_{w.b.}}\right] = HHV(1-M) - 2.443M \quad (1.2)$$

Where LHV is the lower heating value, HHV is the higher heating value, M is the moisture content on wet basis. A more precise LHV calculation can be performed considering also the mass fraction of the hydrogen in the biomass, but its contribution is often negligible.

The heating value of the biomass is significantly lower than for fossil fuels, and it varies depending on the type of biomass (wood species, presence of bark, herbaceous biomass, etc.) and strongly influenced by the water content (with increasing water content of the biomass the calorific value decreases). The typical values of lower heating value and the composition of some woody biomass are reported in Table 1.3.

| Typology | LHV [MJ/$kg_{db}$] | Ash [% db] | C [% db] | H [% db] | O [% db] | N [% db] | S [% db] |
|---|---|---|---|---|---|---|---|
| **wood (conifers)** | 19.1 | 0.3 | 51 | 6.3 | 42 | 0.1 | <0.02 |
| **wood (hardwood)** | 18.9 | 0.3 | 49 | 6.2 | 44 | 0.1 | 0.02 |
| **bark (conifers)** | 19.2 | 1.5 | 52 | 5.9 | 38 | 0.5 | 0.03 |
| **bark (hardwood)** | 19.0 | 1.5 | 52 | 5.8 | 38 | 0.3 | 0.03 |
| **pruning residues (conifers)** | 19.2 | 3.0 | 51 | 6.0 | 40 | 0.5 | <0.02 |
| **pruning residues (hardwood)** | 18.7 | 5.0 | 51 | 6.0 | 40 | 0.5 | 0.04 |
| **short rotation coppice (willow)** | 18.4 | 2.0 | 48 | 6.1 | 43 | 0.5 | 0.05 |
| **short rotation coppice (poplar)** | 18.4 | 2.0 | 48 | 6.2 | 43 | 0.4 | 0.03 |

Table 1.3 Heating value and composition for typical wood biomass (from UNI EN 14961-1)

The influence of biomass moisture content on lower heating value is reported in Figure 1.1, showing the linear decrease of heating value over the increase of moisture content.

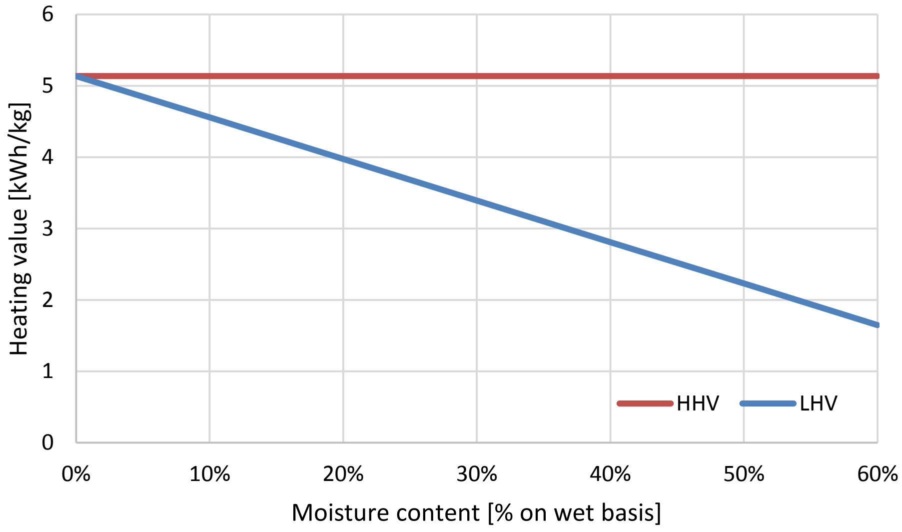


Figure 1.1 Relation between moisture content and heating values of wood biomass

## 1.3 Wood biomass to energy technologies

The use of solid biomass for energy applications involves different processes that can be combined to obtain various energy carriers, depending on the application. Wood biomass can be used for heat production through simple combustion, or more complex processes can lead to the production of electricity or to the transformation in other biofuels. The main available technologies will be briefly described in the following paragraphs, in order to provide a general background on the features of each application. The focus will be kept on the production of heat, power and cold, without considering the biofuels, which are not commonly produced from wood biomass but rather from other energy crops (e.g. sugarcane, corn, etc.). The use of biomass for energy production is promoted by the 2009/28/EC Directive, which supports the energy production from renewable sources.

### 1.3.1 Combustion, pyrolysis and gasification

The main conversion processes that involve solid biomass for energy production are the combustion, the gasification and the pyrolysis. While the combustion is largely used in a number of applications, the gasification and the pyrolysis are still in a research and pilot phase, due to their higher complexity and difficulty of developing a controlled process.

The combustion is a thermochemical process of complete oxidation that occurs in the presence of a fuel and oxygen in an amount at least stoichiometric. The quality of the combustion process depends on both the properties of the fuel and the device in which the reaction takes place. The combustion of solid biomass includes multiple intermediate chemical reactions of different nature, which are the causes of the formation of gaseous compounds and solid pollutants.

A series of physical and chemical phenomena of considerable complexity are involved, as the heterogeneous and homogeneous reactions affects larger areas and longer durations than in the case of liquid or gaseous fuels. In particular, the reaction time depends mainly on the size of the biomass and its physical-chemical properties (mainly the water content).

As the physical state of the fuel is an issue, the choice of the device in which the combustion takes place becomes essential. An optimization of the process requires a thorough design of the geometry and dimensions, the flow rate, the primary and secondary combustion air distribution, etc. Each of this parameters affect the reaction temperatures and kinetics, as well as the contact between the combustion air and the fuel. Some solid biomass types have largely variable chemical and physical characteristics, resulting in a complex management and operation of the combustion process. This is not the case for wood biomass, which is usually basically affected by its variable moisture content, being the calorific value and ash content generally quite constant.

The phenomenology of the process of combustion of woody biomass can be schematically described as the sequence of the following steps:

- heating and drying of the biomass with evaporation of the water contained in it (drying);
- pyrolysis of biomass with the release of the volatile fraction (devolatilization);
- partial gasification of the solid fraction (gasification);
- primary combustion of the gas phase (homogeneous oxidation);
- secondary combustion of heterogeneous char (heterogeneous oxidation).

The phases can be identified in the time sequence indicated or simultaneously in different zones of the heat generator, according to the characteristics of the biomass and the generator (such as the size of the wood, the type of grate, the geometry of the furnace, the introduction of air, etc.).

The step of drying occurs at relatively low temperatures (<100 ° C), by using the energy developed during the actual combustion process. If the biomass input has an excessive water content the energy required in the drying phase can become excessively high, resulting in a drop of temperature below the minimum temperature required to support combustion, and a consequent fail of the combustion process. The water content of the biomass (see point I.3.2.1) significantly influences the entire combustion process (lowering the adiabatic combustion temperature, increasing the volume of gas produced, increasing the necessary residence time and decreasing the overall efficiency of the process) hindering the completion of the oxidation reactions.

The phase of devolatilization (often also referred to as an intermediate pyrolysis phase) consists of a thermal degradation in the absence of oxygen and at temperatures indicatively between 200°C and 1,150°C. The first step is the decomposition of the biomass constituents: hemicellulose, cellulose and lignin, and at the end the gaseous products are formed. The gas resulting from the phase of devolatilization is mainly formed by carbon monoxide, carbon dioxide, steam, hydrogen, methane and other hydrocarbons.

During the gasification stage, which consists of a thermal degradation in the absence of oxygen, a part of the solid fraction evaporates and reacts with the oxygen and the gases produced in the other phases resulting in the formation of char. The char is defined as the solid residue remaining downstream of the stages of drying, devolatilization and gasification and consisting of ashes and unburned solid.

During the final phase of the combustion, the products of the intermediate stages of devolatilization and gasification are fully oxidized, giving rise to a flow of fumes at high temperature, or with high content of thermal energy consisting of: carbon dioxide, water, oxygen and excess nitrogen. A minor part of the flue gases is composed by the products of intermediate reactions described above that represents the polluting part of the flue gases.

A result of the combustion process, in addition to the flue gases, is a solid residue that has not been dragged by the gas stream, which is therefore being collected on the bottom of the furnace within which the combustion has occurred. The solid residue consists of the elements contained in the part of the ash of the biomass that have not been dragged to the stack and a fraction of carbon that has not arrived to complete oxidation.

From a well-controlled combustion of not contaminated virgin wood the main flue gases pollutants consist essentially of carbon monoxide (CO), nitrogen oxides ($NO_X$), a very small percentage of sulphur dioxide ($SO_2$) and dust (usually referred to as total particulate). In these conditions, the content of carbon is quite limited in the ashes. In the case where combustion happen in an uncontrolled manner with little or no control techniques, the percentages of CO and volatile organic compounds (VOCs) in the flue gas increase as well as the carbon content in the ash.

For a well-controlled combustion the following aspects need to be guaranteed during the process:

- combustion temperatures sufficiently high;
- sufficient supply of excess oxygen;
- good mixing between the combustion air and the gas phase of the fuel through adequate turbulence in the combustion chamber;
- residence time sufficient to ensure the completion of the oxidation reactions.

These conditions are reached thanks to the use of biomass of good quality (in terms of size and water content) and thanks to a good design of the heat generator ensuring a correct management through the control of the combustion process by means of the automatic adjustment of the operation parameters of the generator itself.

However, a combustion taking place without a good control can lead to the formation of different types of pollutants from incomplete combustion (such as CO, $C_XH_Y$, IPA, carbonaceous particles, $H_2$, HCN, $NH_3$, $NO_2$,) and pollutants and contaminants from the ashes (KCl, $SO_2$, HCl, PCDD / F, Cu, Pb, Zn, Cd), in addition to problems caused by the melting and sintering of some compounds contained in the ashes.

Pyrolysis and gasification are two thermochemical processes that can be used for the conversion of different solid matrices in liquid fuels, gaseous fuels and bases for the production of other chemicals.

Pyrolysis is the thermal degradation of the solid fuel in the absence of oxygen, without the supply of gaseous reactants. The final product that is obtained is a mix of light gases arising from the cracking of the solid, bio-oils and residual carbonaceous solids (char). Their relative percentages substantially depend on the process conditions: the maximum operating temperature, the heating rate of the fuel and its residence time within the reaction zone, as well as the size and physical form of the biomass (or solid matrix) being treated. The control of the speed of the process allows to maximize the reaction in the formation of the lighter fractions such as liquids and gases (through a fast pyrolysis at higher temperatures) or heavy products such as char and liquid (obtainable through a slow pyrolysis).

The gasification is a thermal degradation of the solid fuel that requires the presence of a gaseous substance, which can be oxygen, air or steam depending on the process. The gasification of a solid fuel is a complex process which involves multiple thermo-physical and chemical mechanisms, which are developed both simultaneously and in sequence. The same reactions take place both in homogeneous and heterogeneous phases, with a multiple-step transformation in which the intermediate products can be further recombined in multiple ways, depending on the thermodynamic conditions in the process. The reactions cannot be completely and easily controlled. As a result, a complete controlled gasification is not always an achievable objective.

The final results of the gasification process is a mix of gases with a different heating value, mainly composed by CO and $H_2$ and a minor share of $CH_4$ and inert or completely oxidised gases as $CO_2$, $H_2O$ and $N_2$. The energy content of the gas which is produced is always lower of the heating value of the fuel, due to the energy share that is consumed during the thermochemical process. The ratio between the energy content of the gas produced and the energy content of the input fuel is known as Cold Gas Efficiency, CGE. This ratio, always lower than 1, is usually used for the comparison of different gasification processes. The temperatures during the oxidation phase are generally between 1,000°C and 1,400°C, lower than in the combustion process. The main reasons are the lower oxygen supplied to the fuel and the share of heat which is used in other endothermic reactions.

### 1.3.2 Heat and electricity production

Different paths are available for the energy use of wood biomass, some of which are already a consolidated technology, while others are still in an experimental or demonstrative phase.

The most diffused conversion mode is probably the simple heat generation, which involves a combustion phase and a subsequent heat transfer to an indoor environment or to a fluid. These two phases usually occur in a heat generator, with typical efficiencies ranging from 75% to 90%, depending mainly on the kind of input biomass (wood chips, wood logs, and pellets) or the size of the heat generator. The biomass boilers will be described in greater detail in the paragraph 5.2.

The production of electricity involves two additional steps: the thermal energy needs to be converted into mechanical energy through an engine, and a further conversion into electrical energy will take place in a power generator. The complex of energy conversions for electricity production can be named thermodynamic cycle. The main distinction is done between internal combustion cycles and external combustion cycles, based on the fact that the combustion is related to the same fluid performing the cycle or to another fluid. The fuel used in CHP applications is usually wood chips, but in some cases pellets are being used as well.

The internal combustion cycles cannot be directly supplied by solid biomass, as they are generally a suitable application for liquid and gaseous fuels. The only possibility of using internal combustion engines is the use of a gasification process, which can be able to produce a synthesis gas to be supplied to the engine. The gasification process has generally an efficiency of 65÷75%, and coupled to internal combustion engines can lead to an overall electrical efficiency up to 30%. An additional amount of heat can be recovered in CHP applications. Despite the relatively high electrical efficiency, these processes are not yet available for commercial application, mainly because of the difficulty of controlling the process in a reliable, efficient and economical way.

The external combustion cycle can be easily applied to solid biomass, as the combustion can be performed in a heat generator and the resulting heat can be transferred to the fluid which operates the cycle. The physical separation between the two fluids assures that no contamination originating from the flue gases is transferred to the working fluid. These paths are currently a consolidated technology, especially for medium and large size CHP plants. The electric efficiency is generally low (up to 20% in small and medium ORC plants and up to 30% in larger steam plants), but the possibility of CHP operation can lead to interesting overall efficiencies (up to 70%÷80%).

The main CHP technologies that can be used with biomass are listed in Table 1.4, along with their average electric efficiency and heat and power ranges.

| Typology | Electric efficiency | Power range [$kW_{el}$] | Heat range [$kW_{th}$] | Technology maturity |
|---|---|---|---|---|
| **ORC turbines** | 10% ÷ 20% | 120 ÷ 1,500 | 800 ÷ 5,000 | market |
| **steam turbines** | 21% ÷ 30% | 5,000 ÷ 150,000 | 15,000 ÷ 350,000 | market |
| **steam piston engine** | 9% ÷ 13% | 400 ÷ 1,000 | 4,000 ÷ 10,000 | market |
| **gas turbine with external combustion** | 12% ÷ 16% | 50 ÷ 200 | 100 ÷ 500 | lab/pilot |
| **stirling engine** | 5% ÷ 15% | 30 ÷ 100 | 100 ÷ 200 | pilot |
| **gasifier and gas turbine** | 10% ÷ 15% | 150 ÷ 2,500 | 1,000 ÷ 10,000 | pilot |
| **gasifier and combustion engine** | 20% ÷ 26% | 150 ÷ 2,500 | 1,000 ÷ 10,000 | demonstration/market |

Table 1.4 Features of wood biomass CHP technologies [6], [8]

The technology maturity is provided as well, as the comparison between the characteristics of different technologies cannot be based only on nominal performances, but the reliability of the system and the annual hours of operation need to be accounted. The technologies that are available on the market are usually providing acceptable reliability and availability, while the others still need to improve before being ready for the commercial applications.

The production of electricity with steam turbines is an established technology in thermal power plants or plants for the combined production of heat and power (CHP - combined heat and power). In the wood biomass applications the heat generated by the combustion process is used to produce steam at variable pressures, between 20 and 80 bars, and a maximum temperature lower than 500°C. The steam is expanded in the turbine to provide mechanical energy to be converted into electricity, and after condensation it is recirculated into the generation system in a closed cycle. The steam cycles are generally the best solutions for power plants larger than 2 $MW_{el}$ and with source temperatures higher than 300°C.

The electrical efficiency of steam plants is a function of the enthalpy and pressure drop made through the turbine, which are related to the condenser temperature and to other parameters, and is generally not higher than 30%. In general, to obtain high electrical efficiencies is necessary to feed the turbine with high pressure steam, but the operation in such points requires highly resistant machinery and therefore high costs, especially in the case of turbines of small power. Due to the characteristics of the steam, its use in small size plants lead to high costs and a low efficiency of the turbine. Multiple stages are required to manage the high enthalpy difference in the turbine, with consequent technical and economic issues. The steam-cycle power systems are in fact subject to significant economic scale effects, and due to this reason the steam plants have generally a power output higher than 2÷3 $MW_{el}$.

The process used in systems with organic fluids (ORC systems) is similar to that used in the steam Rankine cycle, but heat transfer fluids with low boiling temperatures are used instead of water vapour. This allows to operate between relatively low temperatures (between 70°C and 300°C). For this reason some ORC systems have been installed in geothermal installations, solar concentration plants or powered by waste heat from industrial processes. In recent years the use of ORC systems coupled to diathermic oil boilers fired with wood biomass has seen a considerable development, with typical power range from 600 $kW_{el}$ to 2 $MW_{el}$.

The wood biomass is burned into a thermal oil boiler, which supplies the heat source to the organic Rankine cycle operating with a siloxane fluid. The cycle has the same thermodynamic principle of a traditional steam Rankine cycle, without the need of superheating that is generally used in steam cycles. The system, to increase the efficiency, may use of a regenerator on the oil, before the condenser, and an economizer for recovering the heat of exhaust gases from the biomass generator. The use of thermal oil generators entails significant reductions in operating costs compared to steam systems, and the use of organic fluids allows greater operation flexibility, which can operate between 10 and 100% of the output power.

The net electrical efficiency of the plants with ORC modules is generally less than 25%, variable according to the size and configuration of the plant. The lower efficiency of the cycle is offset by other benefits, in addition to automatic operation without supervision of the plant, such as the availability on the market of commercial modules, the low maintenance, the easy start and stop procedures and the operation down to 10 % of the rated load.

Other solutions are being developed in order to decrease the minimum size of the systems, which are generally about 600 $kW_{el}$ using siloxane fluids and thermal oil. A more detailed description of ORC systems is provided in the paragraph 3.1.

In addition to steam turbines and ORC turbines, which are currently the standard applications for biomass to energy conversion, other systems are gaining a growing interest due to their potential advantages. In particular, gasification units could provide higher electricity conversion efficiencies, but they have not yet reached a full commercial maturity, due to the difficulty of standardisation and the low operation reliability.

In recent years some technical solutions have been proposed for small size plants (< 1 $MW_{el}$) to overcome the technical problems of reliability that in the past have characterized these particular types of gasifiers (mainly related to the difficult operation of unloading of the ashes and the formation of agglomerates at the base of the grate, resulting in system shut down after a few hours of operation). These solutions provide a better performance of the grate system and a greater focus on the quality of the material entering the plant. The most diffused technology is the downdraft gasifier, which is however quite customized on each particular application. There are currently some cases of commercial gasification plants fuelled by wood chips that have been in continuous operation for more than two years (with cases of operation even higher at 7500 hours per year).

In the research of systems and facilities for small size CHP generation from biomass, some systems have been developed with a gas turbine with external heat supply. This type of plants uses a Brayton cycle with hot air, produced by a high speed turbo compressor unit. The outside air is first compressed, preheated in a recuperator, heated in a heat exchanger using combustion flue gases and expanded in the turbine. Before leaving the plant the air passes in one or more regenerators, to yield its residual heat to the fresh incoming air and any air-water exchangers for heat recovery in CHP systems.

Some companies have developed systems based on this cycle with typical powers of about 70 to 80 $kW_{el}$ (net output power) and 200 $kW_{th}$, with rated electrical efficiency of about 12% and 50% of thermal efficiency in the case of hot air and 30% in case of hot water. As the heat transfer fluid used in the cycle is air, the system requires an additional heat exchanger for the production of hot water. These data are indicative, subject to change and influenced by external conditions and temperature levels of the fluids. The interest in the development of these systems lies in the relative simplicity of operation and construction, the possibility to have small size systems (less than 100 $kW_{el}$), the exemption from qualified personnel requirements during operation and the absence of pressurized, hazardous or harmful fluids.

Some CHP systems based on small size steam cycle reciprocating engine (typically from 50 to 1,000 $kW_{el}$) are currently in operation, and they are used to supply heat users or connected downstream to the production of steam for process uses. These systems are fed with wood chips or residue from wood processing, in some cases in co-firing with fossil fuels.

The advantage of using a steam cycle reciprocating engine with respect to a steam turbine is mainly related to the possibility of designing smaller plants, which can work even with wet steam, and therefore be able to exploit large enthalpy drops without the need of an excessive superheating.

These plants are able to provide a good reliability and a fairly constant efficiency at partial loads, but they require very high investment costs considering their low electrical efficiency. They are also characterized by high noise and vibration problems.

### 1.3.3 Trigeneration systems

Trigeneration is the combined production of electricity, heat and cooling (sometimes called also CCHP). This combined output is not guaranteed by a single process or system, as in the case of CHP, but involves the use of equipment and technologies dedicated to the production of cooling energy to be connected downstream of the CHP unit. The technologies today mostly used in such applications are represented by absorption refrigerators using water and lithium bromide. The demand for heat in summer, needed to power the refrigeration cycles, is an advantage for the possibility of increase the CHP operating hours, which are usually limited to the winter season.

These types of systems are characterized by high reliability and power consumption limited to the service of auxiliary facilities. They need to be equipped with a system to dissipate the heat, usually a cooling tower. The minimum sizes of system settle in the hundreds of kW. This limit is not related to technological reasons but rather to economies of scale that induce too high investment costs for lower power sizes.

The current commercial absorption chillers have supply temperature constraints between 80°C and 120°C. It must be considered that the need to ensure such temperature levels downstream of cogeneration plants (e.g. ORC), can lead to a reduction in the electric efficiency of the CHP system itself.

Trigeneration plants can be either considered for single users (e.g. hospitals, office buildings) or to supply district cooling networks. In the case of individual users the absorption unit is typically installed near the heat generation unit, in order to reduce the energy losses. In the case of replacement of existing traditional chillers it is important to take into account the larger dimensions of the absorption groups and the higher thermal powers requested by the cooling towers of the system (up to two times for the same amount of cooling output).

District cooling networks can have different system configurations:

- Single central cooling unit and distribution of the cool fluid to the users;
- Heat supplied to the users through a district heating network and multiple absorption units installed by each user.

The choice between these two configurations is typically constrained by the need to simultaneously supply heating and cooling energy. In that case the first solution would require two different pipe networks for heating and cooling, with higher investment costs, which are otherwise avoidable through the second solution. The second case can also be applied to existing district heating networks, but it is important to evaluate the total cooling capacity (user's requirements and refrigeration units) for the sizing of the network.

### 1.3.4 Pollutant emissions from biomass combustion systems

The emissions to the atmosphere from wood biomass combustion plants are mainly composed by carbon dioxide, water, oxygen, nitrogen and a portion of pollutants, which is limited in the case of complete combustion. From the controlled combustion of virgin uncontaminated wood the pollutants in the flue gases consist of carbon monoxide (CO), nitrogen oxides ($NO_X$), a very small percentage of sulphur dioxide ($SO_2$) and dust (referred to as total particulate). In the case of incomplete combustion, in addition to higher percentages of carbon monoxide the emission of volatile organic compounds (VOCs) can become an issue.

Other compounds that are very harmful to the human health and to the environment, such as hydrochloric acid, dioxins and furans may be formed in the case of wood contamination by chlorine and in particular operating conditions of the heat generator.

The carbon monoxide is the last intermediate compound in the chain of reactions leading to the formation of $CO_2$. The completion of the oxidation reactions is guaranteed by the presence of excess oxygen in the gaseous mixture, by sufficiently high temperatures, by appropriate mixing of the compounds in the gaseous phase and by sufficiently long residence times. In the case of direct combustion of biomass, the amount of CO present in the flue gases is a good indicator of the quality of the combustion process. Low levels of CO are usually related to complete combustion and consequently high efficiency and low emissions of dust and nitrogen oxides.

Volatile organic compounds are high molecular weight compounds often referred to as carbonaceous hydrocarbons ($C_nH_m$), issued by the incomplete combustion of wood biomass. They can be mainly distinguished in methane ($CH_4$), non-methane volatile organic compounds (NMVOCs) and polycyclic aromatic hydrocarbons (PAHs). Often these three components are identified separately, because methane is a greenhouse gas whereas polycyclic aromatic hydrocarbons are classified as carcinogenic. Volatile organic compounds are intermediaries of the transformation processes of carbon and hydrogen in the fuel into $CO_2$ and $H_2O$, respectively. Generally the quantity emitted is lower than that of CO.

The nitrogen compounds that are found in the flue gases from biomass (NO, $NO_2$) are mainly produced by the oxidation of organic nitrogen that is present in the molecular structure of the fuel. More than 90% of the oxides produced by the combustion of woody biomass is constituted by nitric oxide (NO) that, in contact with oxygen in the atmosphere, it oxidizes rapidly in nitrogen dioxide ($NO_2$). The mechanisms of formation of nitrogen oxides in combustion processes are:

- "fuel $NO_X$" mechanism: through a series of reactions and in the presence of a sufficient amount of oxygen the nitrogen content in the fuel is converted into nitrogen oxides (around 90% of NO and 10% of $NO_2$);
- "thermal $NO_X$" mechanism: at temperatures above 1,300°C the nitrogen present in the air reacts with oxygen radicals and form carbon monoxide NO. The amount of NO produced increases with temperature, oxygen concentration and residence time;
- "prompt $NO_X$" mechanism: the nitrogen present in the air leads to the formation of NO through a series of reactions characterized by a greater speed and a smaller temperature dependence compared to what happens in the "thermal $NO_X$" mechanism. This mode of formation of nitrogen oxides is predominant in conditions of lack of oxygen, while it is not significant in the case of biomass combustion.

The production of nitrogen oxides in the combustion of woody biomass is due almost entirely to the nitrogen in the fuel, as the combustion temperatures are usually lower than 1,000°C. The total particulate is the set of particles emitted at the stack, mainly including:

- fly ash: carried by the gaseous flow and constituted by particles of diameter > 1 µm containing mainly Ca, Mg, Si, K and Al;
- aerosols: particles of diameter < 1 µm formed by successive evaporation, nucleation and condensation steps starting from compounds of S, Na, K, Cl, Ca, Zn present in the biomass;
- unburned carbon particles, usually referred to as soot and char.

Often the term fly ash is improperly used to indicate the total particulate, since the current Italian legislation on the management of solid waste produced by biomass plants (Legislative Decree no. 152/2006) distinguishes the bottom ashes (ash sub-grate) from the light ashes (which are derived from the dry treatment of the flue gases). The size distribution of particles emitted, distinguishing the fractions of fine and ultrafine PM, depends on many parameters, including the operating conditions of the generator.

In general the particulate can also be distinguished on the basis of:

- the formation mode: primary (directly from the combustion), condensable (from condensation of the flue gases) and secondary (by reaction with the atmosphere of precursor gases, leading to the so-called ultrafine particles);
- the chemical composition (organic and inorganic).

The majority of the particles emitted during the combustion of woody biomass has a diameter less than 10 µm (fraction indicated as PM10) and a significant fraction has a diameter less than 2.5 µm (PM2.5). In literature there are many studies to deepen the analysis of the mechanisms of particulate formation, trying to identify the emission factors for the combustion of solid biomass in different types of boilers, and to define the size distribution of the particles. The results of these studies, however, are difficult to compare, because the tests on these types of equipment are carried out in operating conditions that cannot be generalized. It can be stated that in the case of complete and well controlled combustion, the particulate emissions from combustion of solid biomass consists of particles smaller than those produced by the combustion of hydrocarbons and consists mainly of inorganic matter. During transients and in non-optimal conditions even biomass combustion may form carbon particles (soot and char) with a particle size and porosity similar to those formed by hydrocarbons, resulting in similar problems in terms of the effects on air quality and human health.

During the combustion process the sulphur tends to form gaseous compounds $SO_2$, $SO_3$ and alkaline sulphates, which may partly condense on the particles of fly ash. Experimental tests have shown a strong variability of the percentages of sulphur emitted in the gas phase or bound to the particulate as a function of the presence of alkali elements in the ash of the biomass and the regime of operation of the heat generator. Typically, a substantial fraction of the sulphur in the fuel remains in the solid residue after combustion (bottom ash), a part is emitted as sulphur oxides and a small part is emitted at the stack as particulate matter (salts). Sulphur oxides emitted at the stack during the combustion of biomass are made up for 95% of sulphur dioxide ($SO_2$) and derive from the complete oxidation of the sulphur that is present in the biomass itself.

The carbon dioxide is produced from the carbon in biomass, it is not a pollutant but is the main compound responsible for the greenhouse effect. Considering only the wood biomass combustion itself, it has a null contribution to the emissions (since the carbon dioxide emitted during the combustion phase is almost equal to the amount that the plant has absorbed during its biological cycle of growth), and this is one of the major environmental benefits related to the use of biomass energy. However, considering the $CO_2$ emissions related to the stages of cultivation, cutting and transport, this contribution cannot be zero but should be evaluated on the basis of the energy consumption linked to each phase, and its associated emissions.

### 1.3.5 Flue gas treatment

The national legislation (Legislative Decree no. 152/2006, as amended) prescribes, for thermal biomass systems with nominal output thermal power above 35 kW, maximum emission values of total dust, total organic carbon (TOC), carbon monoxide (CO), oxides nitrogen ($NO_2$) and sulphur oxides ($SO_2$). Some Italian regions prescribe in its territory lower limits than those imposed by the national law.

In the applications referred to in this discussion, the emissions of $SO_2$ and VOCs are of marginal importance. The main measures aim at containing the emissions of CO, $NO_X$ and particulate.

In wood biomass combustion plants to limit the emission of these substances is possible to counteract the formation of harmful particles through the optimization and control of the operating conditions of the heat generator (called primary measures) and/or by removing them from the flue gas through the use of appropriate treatment systems (known as secondary measures).

The primary measures pertain more specifically to the field of design and engineering (R&D) of the heat generator and the entire plant. In general, in a plant properly designed it is possible to ensure low emissions of carbon monoxide, particulate matter and nitrogen oxides thanks to the following measures:

- controlling of the size and moisture content of the input biomass;
- optimization of the combustion;
- presence of automatic control systems and combustion control based on the parameters measured inside the generator and in the flue gas;
- proper management and maintenance.

These considerations can be applied, with appropriate distinctions, both in the case of thermal devices of small size that in the case of thermal power plants.

As for the containment of CO, the possible actions coincide with the implementation of primary measures. To further reduce the levels of other harmful substances at the stack, in addition to the primary measures further action can be undertaken, through appropriate secondary containment systems and emissions reduction:

- particulate: bag filters, electrostatic precipitators, cyclones and specific devices for small plants;
- nitrogen oxides: selective catalytic reduction systems (SCR) and selective non-catalytic reduction systems (SNCR).

The particulate removal systems most suitable for combustion of woody biomass are of three types:

- cyclones and multi cyclones as pre-filters;
- electrostatic precipitators;
- bag filters.

The type and composition of the particles play a fundamental role in determining the effectiveness of the various removal systems. In particular, fundamental parameters for the choice of the most appropriate devices are the size and the size distribution of the particles, their density, the ability of cohesion and agglomeration, the shape, the electrical resistivity, etc.

Cyclones and multi cyclones

The cyclones are an inertial system for the abatement of the particulate, i.e. the main feature of the particles contributing to their removal is the mass. The flue gases are introduced tangentially into cylindrical or conical devices inside of which they get a swirling motion. The resulting centrifugal forces resulting push the solid particles on the inner surface of the cylinders from which they fall by gravity to the bottom, having now lost their kinetic energy.

Cyclones can also operate at high temperatures (above 1,000°C), introduce low load losses and do not have any moving part. For applications in woody biomass systems they not present operational difficulties. Their efficiency is high only for relatively large particles (greater than 90% efficiency for particles with d> 15 µm). For this reason they are usually used as removal systems of larger particulates, and they are coupled to other devices for fine particles removal.

Electrostatic precipitators

The principle of operation of electrostatic precipitators (also called ESP) is based on the creation of a constant electric field through which the combustion gases are conveyed. The electric field is produced by the loading of one or more filaments of a conductive material (negative electrodes), which are in turn arranged inside the plates or metal cylinders (positive electrodes). Because of the potential difference between the electrodes (15 to 20 kV), the gas molecules that pass through the filter are split into positive and negative ions, which are respectively attracted by the electrodes of opposite sign. During the migration to these surfaces they collide (by diffusion and/or interception) with the solid particles contained in the flue gases that are then also electrically charged and thus follow the same path of the ions. For the same electrostatic forces, the particles remain finally retained on the surface of the electrodes.

ESP have high removal efficiencies for particles of any size (> 99%), with a slight decrease in size around 0.5 µm. With such devices, it is possible to achieve final concentrations of particulates even below 10 mg/Nm$^3$. The operating temperatures of the electrostatic filters are higher than those of bag filters (up to 350°C) and the removal effectiveness is related to the values of electrical resistivity of the particles in a range approximately between 103 and 1010 Ohm·cm.

Such systems also require a good availability of space for both the low crossing speed of the fumes (<1.2 m / s) and for the necessary large surface area for particle retention. For usual applications in biomass combustion plants, it is estimated a value of filter surface area of more than 100m2 per m3 / s of flue gases flow rate.

The cleaning of these surfaces is performed by thin streams of water lapping on the surfaces of the electrodes (wet cleaning), or by periodical scraping or shaking of the same collection surfaces (dry cleaning). For the shaking automatic devices (hammers) are users, which transmit the vibrations to the electrodes through regular beats exercised from above.

Bag filters

The bag filters are devices for mechanical filtration. The stream of flue gases is passed through a layer of fabric (made from textile or metal fibres) on the surface of which the solid particles remain adherent due to phenomena of retention of the diffusive and electrostatic type.

The combination of these effects results in a collection efficiency particularly high, greater than 99% for particles larger than 1 µm. The filtering capacity of such devices further increases during the exercise, because of the dust layer which is formed on the surface of the sleeves and acting in turn by a filter medium for the following particles transported by the fumes. With such devices, it is possible to achieve final concentrations of particulates in the flue gas from the combustion products also generally below 10 mg/Nm3. The surfaces that are required for filtration are relatively large, given the low values of surface speed of the fumes usually achieved in these applications (12 to 24 mm/s).

The bag filters employ traditional woven textile fibre. The maximum operating temperatures are therefore relatively low (<250 ° C) and, in case of incorrect operation of the heat generator or the use of non-woody biomass, there may be operational difficulties related to the contact with incandescent particles or the deterioration of the filter due to peaks of speed of the gas inside the sleeves.

The usual cleaning system uses blowing jets of compressed air on the surface of the sleeves, in reverse to the normal passage of combustion fumes. Cleaning frequency is determined as a function of the pressure losses in the filter. This mode determines, however, a strong mechanical stress of the sleeves, which can results in breaking and need of replacement.

The filters more recently introduced on the market are made of materials of metal fibre that can withstand temperatures up to 600 ° C and that have a extended duration thanks to the improved mechanical strength.

Nitrogen oxides: selective reduction systems

The principle of selective reduction is the most suitable to reduce the emission of nitrogen oxides in biomass systems. Since approximately 95% of the $NO_X$ produced during the combustion of biomass is constituted by nitrogen monoxide NO, the removal systems are based on the selective reduction of nitrogen monoxide, which reacts easily with ammonia releasing water and elemental nitrogen.

The two main applications of this principle are:

- selective catalytic reduction (SCR) with injection of a reducing agent over a catalyst (ammonia between 220÷270°C or urea between 400÷450°C);
- selective non-catalytic reduction (SNCR) with injection of the reducing agent (urea) directly into the combustion chamber at 850÷950°C.

Such systems are not typically used in small or medium size plants, while they are generally needed in the steam generators for industrial use or for thermoelectric plants. In these cases, the solution adopted is mostly SNCR, as the SCR would be difficult to apply due to the dustiness of the flue gases.

In SNCR systems the reduction reaction of the nitrogen monoxide is realized directly in the combustion chamber or in the sections immediately downstream, where the temperature range is optimal for carrying out the abatement reactions. The reduction is carried out in the flue gas by injecting ammonia or urea ($CH_4N_2O$) in over-stoichiometric quantities.

The process is sufficiently effective only in particularly stable operating conditions, i.e. under constant conditions of flow rate and temperature of the fumes and ensuring a quite high residence time of the gas. It is also very important to optimize the injection of the reagents and their spread within the fumes. The efficiencies of reduction that can be achieved with this system are in the order of 70%.

## 1.4 Simulation of energy systems

The simulation of energy systems can be performed with a number of different tools and simulation environments, depending on the purpose of the study. Some tools are especially dedicated to particular energy systems or components, while others have a more general range of applications.

The main tools that are usually used in the simulation of CHP systems are listed in Table 1.5. This list does not pretend to be exhaustive, as there are a number of different solutions, especially for commercial design of specific applications.

| Software | | Description |
|---|---|---|
| **Matlab** | non-free | High-level language and interactive environment for numerical computation, visualization, and programming |
| **Engineering Equation Solver** | non-free | Popular equation-based solver that includes a database of fluid thermodynamic and transport properties. |
| **Cycle-Tempo** | non-free | Tool for the thermodynamic analysis and optimization of systems for the production of electricity, heat and refrigeration |
| **Aspen Plus** | non-free | Chemical process optimization software for the design, operation, and optimization of manufacturing facilities. |
| **TRNSYS** | non-free | Simulation program primarily used in the fields of renewable energy engineering and building simulation for passive as well as active solar design. |
| **Gatecycle** | non-free | Plant performance monitoring software for design and off-design performance analysis of combined cycle plants, fossil boiler plants, nuclear power plants, combined heat-and-power plants and many other energy systems. |
| **Modelica** | open source | Object-oriented, declarative, multi-domain modelling language for component-oriented modelling of complex systems |
| **RETScreen** | free | Excel-based clean energy project analysis software tool for determining the technical and financial viability of potential renewable energy, energy efficiency and CHP projects. |

Table 1.5 Simulation tools for the analysis of wood biomass CHP systems

Some tools allow for precise dynamic simulations of each components, and generally require a deep knowledge of a number of parameters. Other models have a higher degree of approximation, but at the same time can be applied to a broader range of applications.

The aim of this work requires the use of a tool that can be used for the simulation of different system components with a high degree of customisation. The interest is rather in steady-state simulation than in dynamic behaviour, as the objective is the analysis of annual operation performance considering hourly time steps. This choice has been limited by the common availability of operational data, which are usually stored as hourly values, and by the intention of considering a representative operation of the systems, ignoring the dynamic variations that have usually a marginal effect over one year of analysis.

Some simulation tools (e.g. Aspen Plus, Gatecycle or TRNSYS) have a very high precision in the component simulation, providing also the possibility to perform dynamic simulation in design and off-design conditions. Nevertheless, they are also somewhat limited in the type of components that can be simulated and also require a number of detailed parameters to provide reliable results. Moreover, as the components are already built-in, there is no possibility of controlling the equations that are used for the model.

Other tools (e.g. Matlab, Engineering Equation Solver and Modelica) are not specifically designed for system simulations, as they are rather general purpose programming language and/or calculation solvers. Using these software requires a complete definition of the model of the component that needs to be simulated, and often results in more approximated results. However, the higher degree of customisation that can be reached allows the user to have a deeper knowledge of how the simulation model is actually working.

For this reasons, in this research work EES has been used for the simulation of the ORC unit, while Matlab has been chosen for the definition of the whole system.

The use of EES has been mainly driven by its comprehensive thermodynamic library that includes all the working fluids that are commonly used in ORC plants. Moreover, the tool provides the possibility of defining a system of equations that can be solved with different available inputs. This feature allows to use real operation data, whose availability can differ from one system to another. The use of EES has been applied to the ORC unit, as it provides an acausal simulation environment for the calculation of the working points, resulting in reliable outputs. The results of the EES simulations has been then integrated in the main model developed with Matlab.

The choice of Matlab has been based on its wide range of applications, and the possibility of defining different scripts with a high degree of customization. Moreover, Matlab allows for a good compatibility with other tools, and provides a developing environment that can be used for multiple purposes. Finally, the choice of Matlab with respect to other similar tools (e.g. Modelica) has been made for the familiarity of the candidate with this environment.

# Chapter 2 Demand side: District heating systems analysis

The most common uses of heat from wood-fired CHP systems are the industrial sites and the small or medium DH networks. The former are usually limited to the sites where the biomass is already available as by-product, i.e. the wood treatment utilities or the furniture factories. Other industry sectors can use wood biomass due to its lower price with respect to other fuels or for the presence of incentives for energy production from renewable sources. The heat load can be variable from site to site, depending of the heat required by the industry process, and in most cases the heat is required at high temperatures, often as process steam.

The larger amount of heat produced from solid biomass CHP plants in Europe is supplied to DH networks. In 2010 EU27 countries consumed more than 7.5 Mtoe of heat from solid biomass plants supplied through DH networks [9]. Heat only plants produced 2.9 Mtoe, while the remaining part has been produced in CHP plants. The countries with the larger amount of heat production are Sweden (2.6 Mtoe), Finland (1.5 Mtoe), Austria (9.3 Mtoe) and Denmark (8.9 Mtoe).

Italy has a different situation, related both to climate and geographic conditions and availability of wood biomass, resulting in smaller DH networks, usually related to a local availability of wood biomass. There are currently 86 DH networks working on wood biomass, of which 16 equipped with CHP systems (as of 2012, [10]). The plants have 425 MW of installed thermal capacity, and 25 MW of CHP electric capacity. A total length of 910 km of network serves about 16.000 final users, providing about 2 TWh of heat and 200 GWh of electricity produced from 750.000 t of chipped wood [11].

In this research work the operation data of some DH systems have been analysed and compared, in order to build a preliminary dataset to be applied to the CHP simulation model. This dataset can be used for a preliminary study when demand data are not available with an acceptable time step detail. However, the heat load is conditioned by several aspects, and the definition of a reliable model for the heat load forecasting is not trivial. Hence, in this work the real data of some systems will be analysed and compared, in order to find the main drivers of the heat load variability and to investigate the relation between the heat demand and the main key parameters.

The current chapter presents the activity that has been performed on the demand side analysis. The paragraph 2.1 provides a general introduction to DH systems, while in paragraph 2.2 some additional information is provided about the description of the case studies and the typical performance analyses and indexes that are considered in DH systems. Finally, paragraph 2.3 provides a detailed description of the operational data that have been evaluated in the case studies and the results that have been obtained.

## 2.1 District Heating Systems description and key considerations

District heating networks are used to supply heat to multiple users taking advantage of a centralized production, which can allow for higher conversion efficiencies and easier flue gas treatments. Moreover, the size of centralized plants often allow CHP generation, resulting in an increased overall efficiency of the system.

District heating systems have a wide variability over multiple aspects:

- The size of the system;
- The type of fluid and the network configuration;
- The technology and fuel used for heat production;
- The integration of renewable energy sources;
- The distribution and characteristics of the users.

The DH systems can be applied to a wide range of building stocks sizes, from small towns (some hundreds users, starting from 100.000 $m^3$ of buildings) to large networks supplying big cities (thousands of users, up to several million $m^3$ of buildings). This aspect has usually consequences on the other features of the system, as the small grids have much different needs, limits and peculiarities with respect to large cities.

The fluid used in DH system can vary depending on the application: hot water, superheated water, steam, thermal oil. The choice of the fluid is also strictly related to the choice between direct networks or indirect networks. The first case, usually applied to very small systems, has a single fluid loop which supplies directly the radiators in final users. The indirect system uses two separated networks, with a heat exchanger as interface. Direct systems are better for small systems, while in larger systems the direct configuration would have unacceptable heat losses, pressure drops and management issues. The major part of the current DH networks in Europe are using hot water or superheated water in an indirect configuration.

The water temperature required is related to multiple factors, including the length of the grid, the temperature requirements from the users and the heat generation technologies. Traditional networks working with hot water have usually supply temperatures between 80°C ÷ 90°C and return temperatures in the range 45°C ÷ 60°C. New DH systems using RES technologies are using lower water temperatures in order to maximize the contribution of solar or geothermal sources. However, return temperatures lower than 40°C require that the users can use low temperature heating systems (e. g. radiant floors). A low return temperature can also increase the efficiency of fossil-fired CHP systems, by recovering additional heat from the flue gases.

The technology used for heat production is related to the specific site. Many options are available: fossil fuels CHP or boilers, waste incinerators, nuclear power plants, biomass boilers or CHP, geothermal sources, heat pumps, solar systems, waste heat from industry processes, etc. Large DH systems are often supplied by different sources, which can maximize the overall efficiency by taking advantage of the peculiarities of each (e. g. by using the CHP as base-load and the boilers for peak-load). Heat storage systems are used to match the demand fluctuations without the need of a continuous variation of the generation units.

The conversion efficiency is dependent on the energy source, the size of the unit and the conversion technologies. Natural gas fired CHP systems have generally higher efficiency, while waste recovery and solid biomass systems have lower performances, especially at small sizes. An important parameter to be taken into account during the design phase is the heat to power ratio, which represents the available heat output per unit of power output. Typical values range from lower than 1, for combined cycle, to more than 4, for biomass ORC plants. This factor is a crucial aspect when considering the size of CHP unit for a given DH network, as the energy production should be sized to match a part of the heat demand of the DH network.

The use of renewable sources in DH networks is gaining interest, thanks to the possibility to match the European RES and climate targets in the heat sector. The main renewable source currently used for DH is biomass, both from wood and municipal solid waste. However geothermal and solar sources are gaining interest, and several systems are already in operation [12]. The DH system in Marstal (Denmark) is supplied by 100% RES, thanks to a solar collector field (providing about 30% of the annual energy), a wood biomass ORC system and a $CO_2$ heat pump [13].

Finally, the characteristics of the users is probably the main aspect influencing the heat demand that has to be fulfilled by the system. The final users can be residential, commercial, public authorities or industrial users. These sectors have different heat load characteristics, both in term of annual energy requirements and daily load profiles. Industrial users usually require a definite load profile, which is dependent on the process schedule. Other users, typically needing energy for ambient heat, can have a much variable and unpredictable load profile. The daily variation is also related to the use of the buildings: residential users typically have different operation hours than commercial buildings, schools and offices. A DH system can have a mix of user's typology, but usually industrial and civil sector requirements are too different to be supplied from the same network in an efficient way. Moreover, in the Italian framework DH systems are not economically sustainable for industry users, due to the low cost of natural gas for industries.

## 2.2 District Heating Systems operation: main features

Each space heating generation system needs to be able to meet two main requirements: to supply the maximum heat required by the users, and to operate in a wide range of output thermal power in order to match all the different conditions over the year. Considering DH networks, the optimal system layout, the choice of the components and the operation settings need to be defined considering the actual heat demand variations that the system will need to deal with. The two main heat profiles that affect the DH operation are the variations that occurs throughout the day and the year.

The real operation data of some DH systems located in Piedmont Region have been collected in order to analyse the heat demand behaviour. The DH systems that have been considered in this work are listed in Table 2.1.

| Parameter | **Unit** | **Case 1** | **Case 2** | **Case 3** | **Case 4** |
|---|---|---|---|---|---|
| Location | | Torino | Leini | Sestriere | San Sicario (Cesana Torinese) |
| Elevation | m (a.s.l.) | 239 | 245 | 2035 | 1354 |
| Min. Outdoor Temperature | °C | -8 | -8 | -18 | -14 |
| Connected buildings | $Mm^3$ | 53.40 | 0.52 | 0.98 | 0.35 |
| Prevalent building type | | residential | residential | holiday houses | holiday houses |
| Installed thermal power | $MW_{th}$ | 1,766.0 | 13.5 | 31.6 | 13.9 |
| Heat source | | Natural gas | Biomass | Natural gas | Natural gas |

Table 2.1 DH systems main features

The first two cases are located in the plain and supplying residential and tertiary buildings, whereas Sestriere and San Sicario are two mountain villages, mainly composed by holiday houses and hotels. Starting from the available operation data, the main operational features of DH systems will be described in the present paragraph, and the paragraph 2.3 will focus in detail on the comparison between the operational analyses of these systems.

The detail of the analysis that has been performed is set by the availability of operation data with a narrow time step. For the cases of Leini, Sestriere and San Sicario the data are available on a daily basis. For the DH system of Torino the dataset is much more detailed, as the information is available each 6 minutes over ten heating seasons. Therefore, the analysis of this case will be much more interesting, providing the possibility of investigating both seasonal and daily variations. The other cases provides however useful information for the comparison of different climate conditions and types of final users.

### 2.2.1 Description of the DH case studies

The four DH systems considered have significant differences in terms of size, typology of users, network temperatures, climate conditions and generation units. However, not all these aspects have an influence on the heat profile, and therefore useful comparisons can be performed.

Torino

The DH system of Torino is among the largest in Italy and Europe, supplying today about 54 million cubic meters of buildings with 474 km of network length [14]. The heat is provided by two cogeneration sites, Torino Nord and Moncalieri, where high efficiency natural gas combined cycles supply the main part of the annual energy required by the network. Other sites (BIT, Mirafiori Nord and Politechnic) provide natural gas integration boilers that are used during peak hours. The location of the generation sites can be seen in Figure 2.1.


Figure 2.1 DH system of Torino [14]

Each site has different generation units, which are listed in Table 2.2. This is the current configuration of the system, which has seen a progressive expansion throughout the years. The larger site is Moncalieri, where two natural gas combined cycles (NGCC) are operated in CHP mode with up to 90% of overall efficiency. The current configuration has been completed after the refurbishment of the power plant, as the two current NGCC have been installed in 2005 and 2009. Three natural gas boilers are available as integration and backup, providing an additional thermal power of 141 MW.

The other large generation site, located in the North part of the city, has been set in operation at the end of 2011. The CHP unit is a NGCC providing an electrical output of 400 MW and a thermal output of 220 MW. Three steam generators are operating as backup/integration boilers, with a total thermal power of 340 MW. Six heat storage systems are available, for a total volume of 5,000 $m^3$ of water.

Other sites are producing heat through natural gas boilers (BIT and Mirafiori Nord), heat storage systems (Martinetto) or both (Polytechnic). These sites are located near to the city centre, in order to supply additional heat to the grid in a better position with respect to the users. The availability of multiple generation plants distributed over the DH area also reduces the need of high supply temperatures in the main generation station, which would be necessary to serve the users located far away from the unique generation plant.

| Site | Unit | Electrical Power [MW] | Thermal Power [MW] | Volume [$m^3$] |
|---|---|---|---|---|
| **Moncalieri** | 2 x NGCC | 800 | 520 | |
| | natural gas boilers | | 141 | |
| **Torino Nord** | NGCC | 400 | 220 | |
| | natural gas boilers | | 340 | |
| | heat storage systems | | | 5,000 |
| **BIT** | natural gas boilers | | 255 | |
| **Mirafiori Nord** | natural gas boilers | | 35 | |
| **Polytechnic** | natural gas boilers | | 255 | |
| | heat storage systems | | | 2,500 |
| **Martinetto** | heat storage systems | | | 5,000 |
| **TOTAL** | | **1,200** | **1,766** | **12,500** |

Table 2.2 Generation sites and units

The total buildings connected to the DH system has been constantly increasing over last decade, from about 25 million $m^3$ in 2004, to 40 million $m^3$ in 2010 and 54 million $m^3$ in the current situation (as of 2014).

The network is operating with superheated water, with usual supply temperature of 120°C during the winter season, with a return temperature of 70°C. The DH system operates also during the summer season, in order to supply domestic hot water and heat to some users working all year round (e.g. hospitals). The heat losses of the system are estimated to be around 14%.

The operation of the DH system of Torino has been analysed from October 2001 to April 2011 [15]. The following data have been collected with a time step of 6 minutes for each generation unit:

- thermal power supplied to the grid,
- water mass flow rate,
- water supply and return temperature,

- outdoor temperature.

The aggregation of the data of all the generation units allows to calculate the total heat supplied to the DH network at each time step. At the same time, the detail of each generation unit helps to identify management logics and operation behaviours.

The DH system Manager provided also information about the total volume of buildings supplied by the system, with a monthly update. These data allow to calculate a specific heat load, in order to compare the DH system behaviour over multiple years. Moreover, the specific heat load can be useful to be compared with other DH systems with similar climate conditions.

Leini

Leini is a little town of about 15,000 inhabitants in the outskirts of Torino. More than 500,000 cubic meters of buildings (mostly residential structures and commercial and public administration buildings) are supplied by a 12-km DH system powered by two biomass boilers (5 MW each) and a natural gas emergency boiler (3.5 MW). The annual energy supplied by the system is about 17 GWh, with a consumption of more than 9,500 tons of chipped wood.

Figure 2.2 shows the biomass storage in the heating plant of Leini. The main problem of the system is that the two biomass boilers currently in operation are not supported by any heat storage system, resulting in a highly variable operation with consequences on the combustion efficiency and on the generation performance of the boilers.

Figure 2.2 Biomass storage in the Leini heating plant

Sestriere

The DH system is located in a touristic mountain village at 2,000 m above sea level. The inhabitants are less than 1,000, but the total sleeping accommodations reach almost 5,000 in the hotels and exceeds 12,000 in the holyday houses. As a result, the heat demand has among the main drivers the presence or absence of the tourists. The DH system currently supplies 177 users, for a total heated volume of 980,000 m$^3$, of which about 60% of residential users (including holyday houses) and the remaining part of public and commercial utilities (mainly hotels and touristic structures). The network length is 13.7 km, and the operation temperatures are around 90°C – 60°C during the winter season and 70°C – 40°C during the summer season.

The heat is generated using natural gas as primary source, both by CHP production through engines and through backup and integration boilers. There are currently five engines, for a total electric power of 8.25 MWel and a total thermal output of 8.735 MWth. The eight natural gas boilers have a total available heat output of 22.9 MWth, and therefore the maximum thermal power available to the DH network slightly exceeds 30 MWth. The nominal efficiency of the boilers is 0.91, while the engines have an electric efficiency of 0.39 and a thermal efficiency of 0.44.

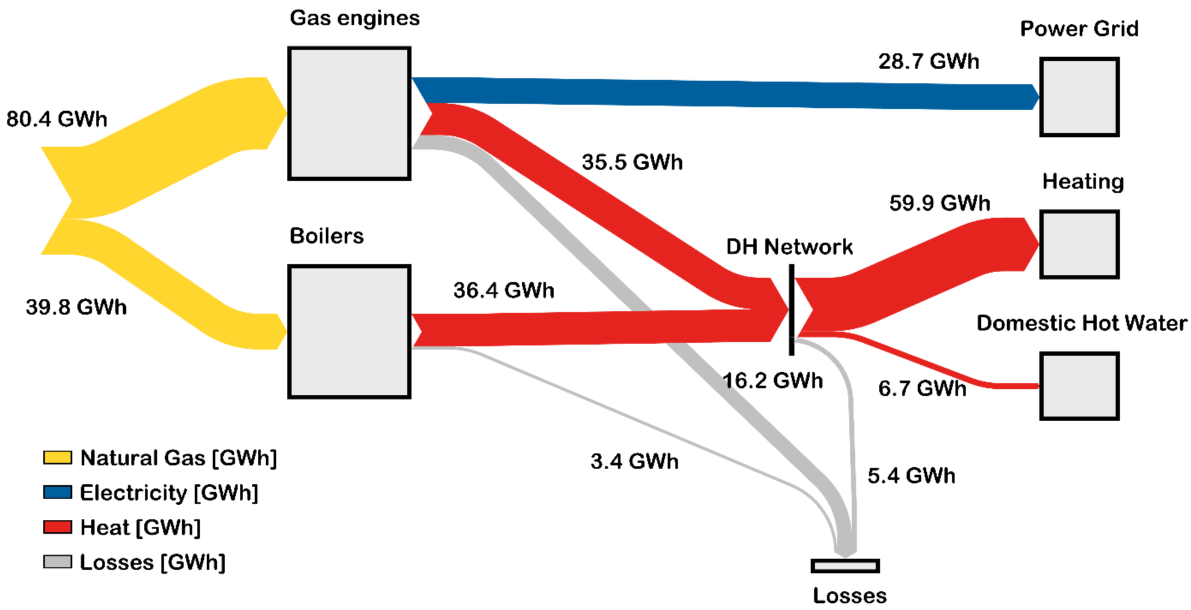


Figure 2.3 Sankey diagram for Sestriere DH network (Sestriere, 2013)

Figure 2.3 shows a Sankey diagram of the different energy flows in the DH system over a year of operation (considering the year 2013, data from [16]). The annual primary energy consumption is about 121.2 GWh, for an electricity gross production of 28.7 GWh and 71.9 GWh of heat supplied to the DH network. The larger part of the heat is consumed for space heating purposes, while the domestic hot water requires a small share (about 10% of the final heat supplied to the users). The network thermal losses are about 8% of the total annual heat provided to the network. The global annual overall efficiency of the system, considering both electricity and heat production, is equal to 79.3%.

<u>San Sicario</u>

The DH system of San Sicario is located in the village of Cesana Torinese, in the Province of Torino. In operation from 2005, this system supplies 54 residential and commercial buildings, for a total volume of 355,000 m$^3$. The generation plant is located at 1,670 m above sea level, and it is composed by three natural gas engines providing the larger share of the heat supplied to the network, and by three natural gas boilers used for integration and backup. Each engine has a nominal output power of 1 $MW_{el}$ and a nominal heat output of 1.194 $MW_{th}$. The nominal electric efficiency is 38.9%, and the heat recovery lead to a global conversion efficiency of 87.8%.

Considering the most recent operation data available (year 2013, from [16]), the heating plant supplied to the network about 23.6 GWh of heat, whereas the heat supplied to the final users has been 21.1 GWh, of which about 10% of domestic hot water and the remaining part for heating purposes.

### 2.2.2 Annual heat profile

The annual heat profile of a DH system is related to the climate variations along the year. There is a proportionality between the outdoor temperature and the thermal energy required to heat a building, although there are other phenomena that have an effect on heat consumption (e.g. radiation, wind speed, etc.). The seasonal variations need to be taken into account during the design phase of the DH system, as the heat load requested to the generation units can vary depending on the season of operation.

Large DH systems have usually multiple generation units, and therefore some of them are shut down during mid seasons, in order to avoid part load operation. This solution cannot be applied to small DH systems, as the installation of multiple boilers or CHP units under a certain size becomes economically unacceptable. Some types of unit are more appropriate for partial load operation, while others have a better performance running constantly at nominal load. Moreover, each unit has a proper inertia, resulting in different ability to react to fast changing loads.

The heat load of the DH system of Sestriere in the year 2013 is showed in Figure 2.4, providing an example of the differences between winter and summer operation and the share of heat provided through natural gas engines and natural gas boilers.

The difference between winter and summer season is significant: while in February the system reaches daily heat productions up to 400 ÷ 450 MWh, between July and September the daily consumption is lower than 100 MWh. However, the energy consumption during summer season remains noticeable: due to the critical climate conditions the buildings require to be heated also during summer, especially at night. Moreover, a significant part of the users is represented by hotels, which can require tap water throughout the year. As the chart shows the heat production, which is supplied to the DH network, the DH thermal losses are included.

Figure 2.4 shows also the share of heat produced with engines and boilers. The CHP units are generally providing a base load, trying to keep a constant operation in order to optimize their performance. The peak load is supplied using the integration boilers, which can match the variations required by the users with a higher flexibility. This system can rely to five engines and eight backup/integration boilers, with the possibility to operate them near their nominal power output in a wide range of conditions, resulting in better conversion efficiencies and an optimization of the primary energy consumptions.

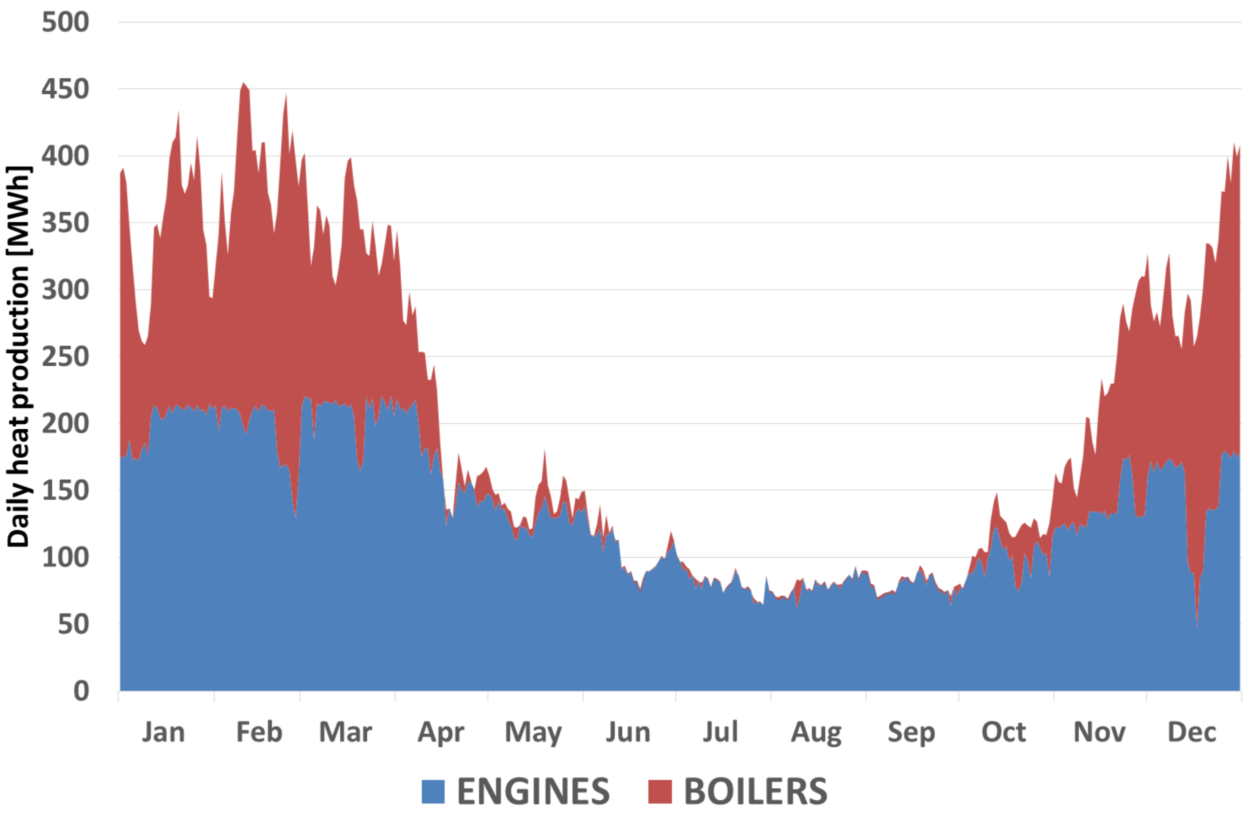


Figure 2.4 DH daily heat production, shares of natural gas engines and boilers (Sestriere, 2013)

Some DH systems are in operation only during the heating season, the domestic hot water being provided through other generation systems (e.g. electric boilers, solar panels). The DH systems that are operated year-round have usually a low efficiency during the summer season, as a significant share of the total heat supplied to the grid is lost to the ground. While in the winter season the network losses are a minor share of the total delivered heat, during the summer the amount of losses become a significant share of the total heat required by the network. Some studies are trying to limit this drawback by installing centralized solar collectors, in order to avoid the use of fossil energy during the summer season [13].

### 2.2.3 Daily heat profile

Another source of variability in heat demand can be seen on a daily basis, i.e. the heat demand has a considerable variability depending on the hour. The daily heat profile of a DH network is strongly related to the type of users, and the behaviour of different user categories. The main share of the heat consumption is caused by space heating, and therefore it is related to the difference between the ambient temperature and the outdoor temperature. While the latter is determined by the climate and cannot be controlled, the internal ambient temperature is a result of the regulation and control of the building heating system.

The set point temperatures of the buildings can vary depending on the hour of the day, the day of the week and the category of user [17]. In residential buildings, there is usually a night setback control, which means that the set point for the indoor temperature is lowered at night. In service buildings, e.g. schools, the ventilation is usually off when no people is using the building (at night, during the weekends, etc.). The absence of ventilation significantly reduces the heat consumption of the building. Some specific users require higher temperatures (e.g. hospitals) or lower temperatures (e.g. industrial buildings), resulting in different consumption profiles.

The behaviour of the network depends on the aggregate of the buildings supplied by the DH system. While different buildings could theoretically compensate each other, DH systems are often supplying a large amount of residential buildings with the same operational logics and hourly settings. As a result, the higher consumption usually occurs at the beginning of the morning, when all the systems which are controlled with setback at night switch to day-load operation, and therefore the building need to be heated up to the daily set point temperature [18].

These considerations results in variable heat load during the day, as can be seen in Figure 2.5. The plot shows some real daily heat load profiles in different months for the DH system of Torino in the year 2010. The three profiles provided in the plot are real profiles of chosen working days, obtained with a time step of 6 minutes.

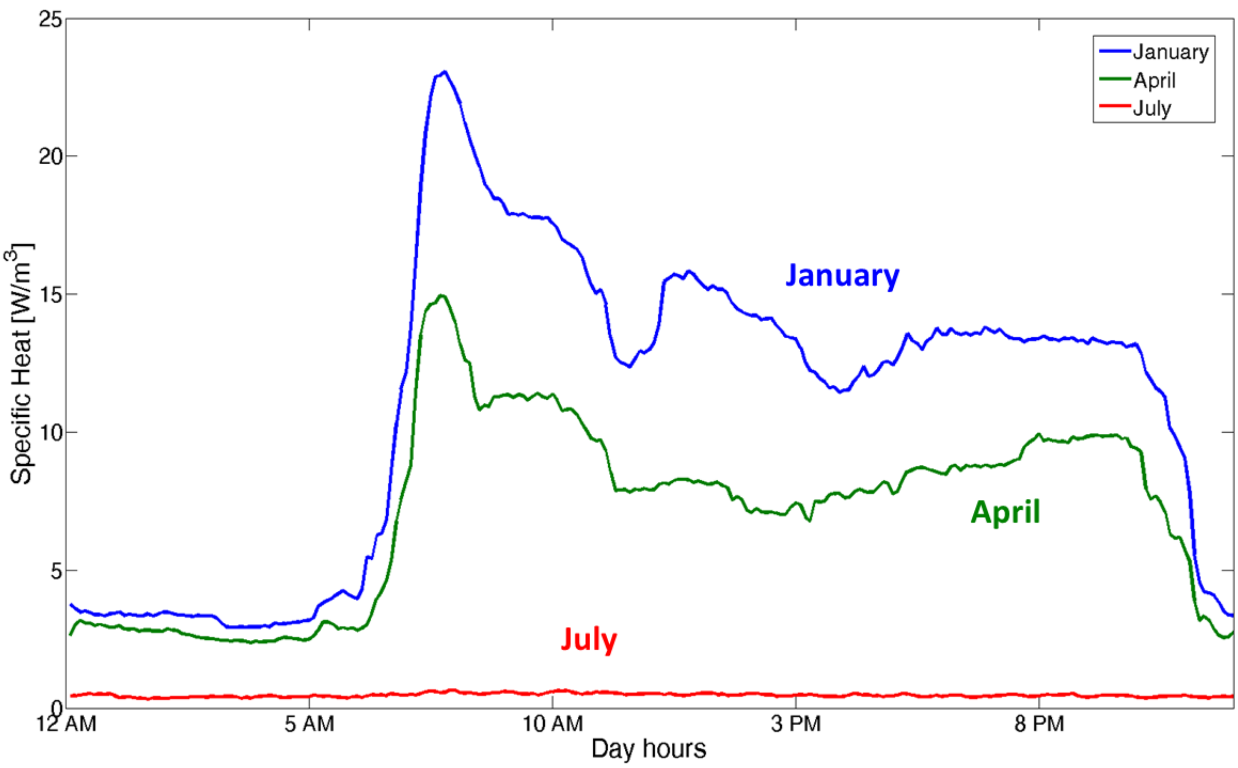


Figure 2.5 Specific heat supplied to the DH network of Torino (real values, 2010).

The load is represented as specific heat, divided by the total volume of buildings connected to the network, in order to be comparable across the years (due to the changing of total amount of buildings connected to the network) or to other DH systems. January and April show a similar behaviour, with the main peak during the morning and a minor peak around midday. July is quite flat because during summer the heat is used for domestic hot water and other uses that are not related to space heating. The plot represents the heat supplied to the DH network from all the generation units, i.e. it includes the network heat losses. The plot shows that the peak load can reach up to 60% more than the average daily load, resulting in the need of sizing the heat generation units considering the annual peak load.

### 2.2.4 Heat load duration curve

A simplified way of considering the operation of a DH system is the heat load duration curve analysis. The duration curve is similar to a load curve, but it is an arrangement of the annual hourly heat consumptions from the higher to the lower. This representation is more convenient for analysing the amount of time required for each heat load. The area under the load duration curve represents the sum of the heat supplied by the DH system.

The duration curve can be built over different periods of time and with different time steps. In DH systems the most used refers to an annual load duration curve with an hourly time step. This curve is often used in the design phase of the system, supporting the evaluation of the share of energy to be supplied by CHP units, MSW plants or boilers.

Considering the DH system of Torino, the hourly heat load duration curves are plotted in Figure 2.6 for different heating seasons.

The plot shows specific heat consumptions, i.e. the ratio between the heat consumption and the total volume of the buildings supplied by the DH network. This correction is necessary in order to compare different years of operation, as the DH system has increased from about 22 $Mm^3$ of connected buildings in the year 2001 to about 39 $Mm^3$ at the beginning of 2011.

The plot shows that the DH system has a comparable behaviour over the year, with some differences that are caused by the changing weather conditions from a year to another. Two heating seasons, which have been plotted with dashed lines in Figure 2.6, show a larger deviation from the others. This behaviour can be related both to climate conditions and to the connection of new buildings to the grid during the year, resulting in a slightly imprecise calculation of the specific heat, as the information of the connected buildings is available on a monthly basis.

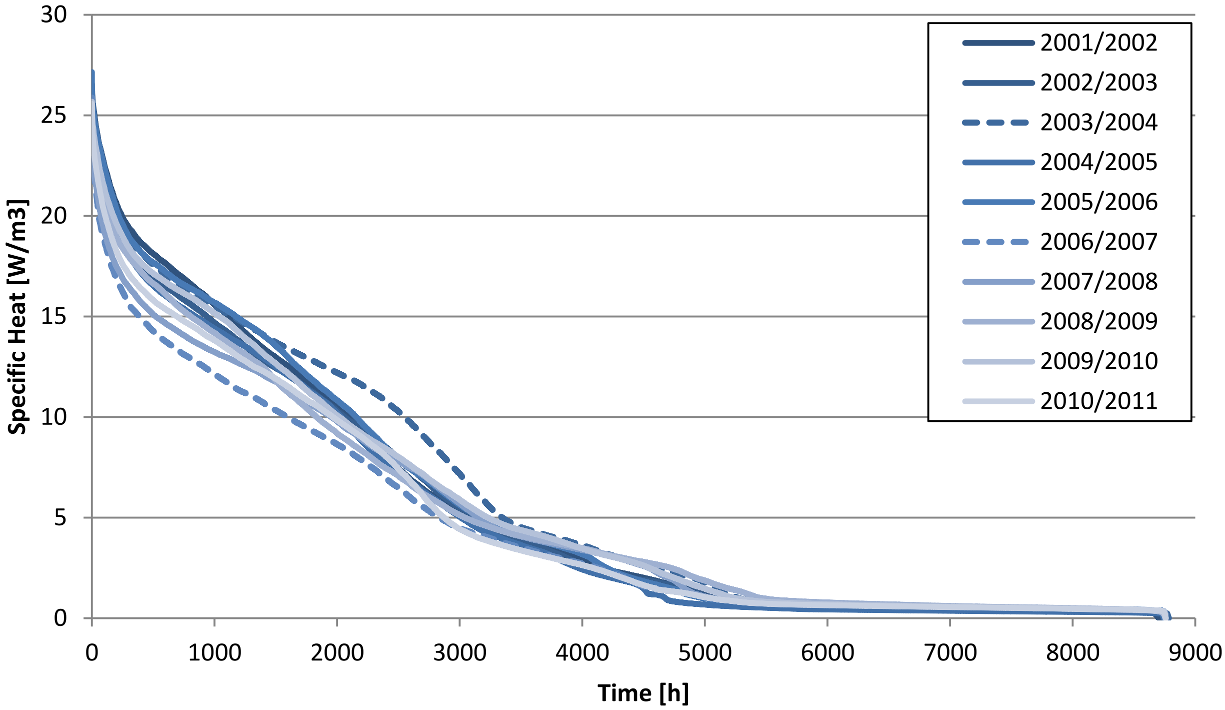


Figure 2.6 Specific heat load duration curves for the DH system of Torino

The heat load duration curve can also be used to analyse the share of energy produced by each generation unit. Figure 2.7 shows the contribution of each type of heat generator to the fulfilment of the total heat requested by the users in the DH of Torino in the year 2010.

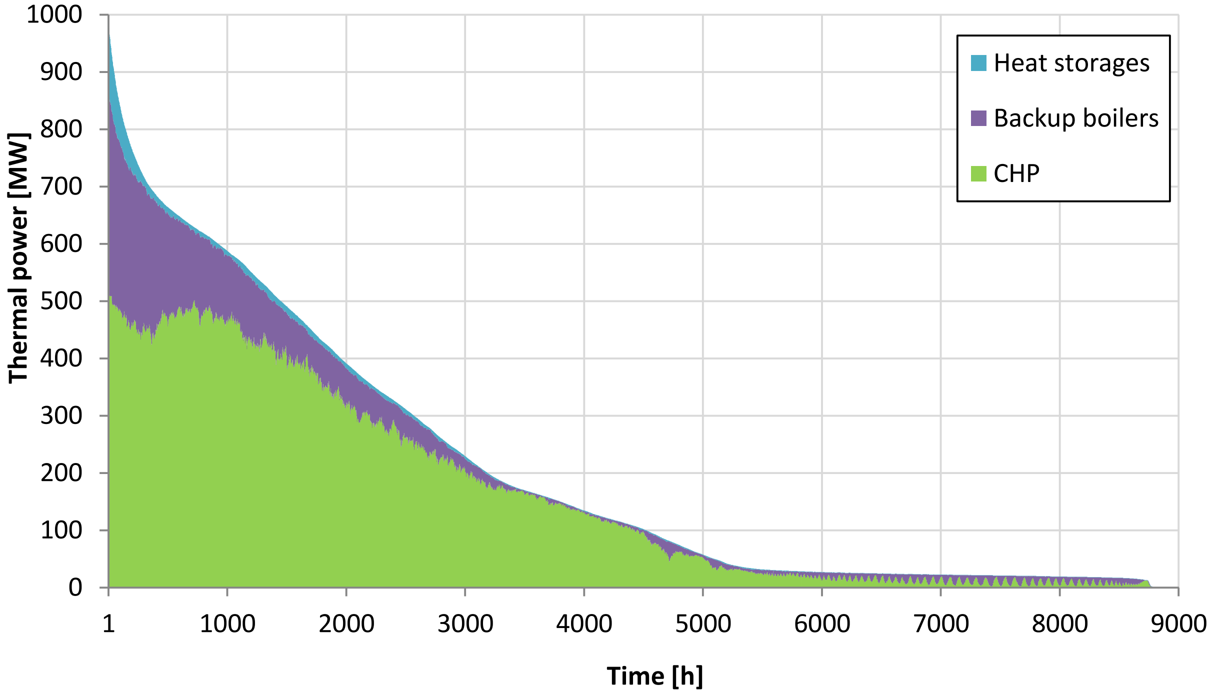


Figure 2.7 Heat load duration curve, different production units (Torino, 2010)

The main share of the energy is produced by CHP units (natural gas combined cycles), which provide the base load. The peak load is generated by the backup and integration boilers, and through the heat storage systems that are loaded during the night using the CHP units and unloaded during the morning peak.

However, the heat duration curve does not provide information about the inertia of the generation units, as the consecutive hours represented on the plot can be located in different periods along the year. As a consequence, this analysis can provide a first approximation of the DH operation logic, and a useful tool for comparing different years with an intuitive graphic format, but further evaluations need to be performed considering the daily operation logics.

## 2.3 District Heating Systems operation: comparisons and results

The possibility of comparing the operation of different DH systems can lead to a first definition of a possible dataset of heat load to be used in the integrated simulation model developed in this research work. This dataset is an attempt to provide a generalization of the behaviour of a general district heating system operating in Northern Italy. The main parameter that has been considered for the formulation of the model is the total volume of connected buildings. However, other key factors have an importance on the heat load, such as the climate conditions, the typology of buildings, the building density, etc. For this reason, further research can be performed, but there is a need of a much wider set of DH systems in order to obtain statistical results.

The access to DH operation data is not trivial, both for privacy reasons and for the availability of the data (especially in small DH systems). In this work four different cases are compared (see Table 2.1), with different years of operation depending on the data availability. This analysis lead to the definition of a dataset representing the users' thermal load in a DH network, to be used as a general case if a more precise heat load is not available. This dataset has be intended to be scaled only on the size of the system (volume of buildings), as a scaling on the climate conditions would need a wider set of systems with much different locations, which have not been obtained for this work.

### 2.3.1 Comparison between different systems: daily analysis

The significance of the case study need to be proven in order to apply the results to other similar systems or to use them for forecast and design purposes. The availability of high-detailed data with a narrow time step is not easy, due to multiple reasons. Not all the DH managers have a continuous measurement of the heat production, and often the network is supplied by different generation units. In some cases, these units are also located in different places, resulting in a simultaneous heat supply in different points of the network. Moreover, the DH operation results in a huge amount of data, which are usually considered only during the live control of the system. Therefore the stored data are rarely analysed by the DH managers.

The operation analysis is focused on the case studies presented in the previous chapter. A synthesis of the main characteristics affecting the operation of these systems is provided in Table 2.3.

| Parameter | **Unit** | **Case 1** | **Case 2** | **Case 3** | **Case 4** |
|---|---|---|---|---|---|
| Location | | Torino | Leini | Sestriere | San Sicario (Cesana Torinese) |
| Elevation | m (a.s.l.) | 239 | 245 | 2035 | 1354 |
| Degree Days | DD | 2617 | 2722 | 5165 | 4775 |
| Design Outdoor Temperature | °C | -8 | -8 | -18 | -14 |
| Connected buildings | Mm3 | 53.40 | 0.52 | 0.98 | 0.35 |
| Prevalent building type | | residential | residential | holiday houses | holiday houses |
| Network length | km | 467 | 12 | 14 | 5 |
| Heat supplied (in 2013) | $GWh_{th}$ | 1,923.1 | 14.3 | 66.5 | 21.1 |
| Specific heat | $kWh/m^3$ | 35.4 | 29.2 | 68.0 | 59.5 |
| Installed thermal power | $MW_{th}$ | 1,766.0 | 13.5 | 31.6 | 13.9 |
| Equivalent hours (in 2013) | $h_{eq}$ | 1,090 | 1,060 | 2,100 | 1,520 |
| Network losses (in 2013) | % | 17% | 19% | 8% | 10% |
| | | | | | |
| Available daily data | | 2002-2010 | 2009-2011 | 2006-2009 | 2007-2013 |

Table 2.3 DH systems comparison [16]

The data available for the DH analysed are the following (daily basis):

- average outdoor temperature (°C)
- minimum outdoor temperature (°C)
- maximum outdoor temperature (°C)
- rain (mm of water from h 0 to h 24)
- global horizontal radiation (MJ/d)
- wind speed (m/s)
- height of snow on the ground (m)
- heat supplied to the DH network (MWh).

The heat supplied to the DH network has been converted into specific heat by dividing it by the connected volume of buildings, in order to compare the different systems.

The weather data have been obtained from the Weather database provided by ARPA Piemonte [19], using different weather stations (Torino Giardini Reali, Torino Buon Pastore, Caselle, Sestriere, San Sicario, Cesana Thuras) in order to provide the dataset needed for calculations. The main results of weather data describing the different locations are listed in Table 2.4.

The difference between urban and mountain systems appears clear, especially with respect to temperatures, radiation and snow. The higher radiation is related to a cleaner atmosphere and to the significant part of diffuse radiation provided by the snow during winter and spring seasons. It has to be noticed that the weather data of Leini and Sestriere are the results of only 3 years of analysis (due to limited DH operation data available for those sites), and therefore could be not significantly representing the usual weather conditions of those sites. However, for the aim of this study only the weather data measured during the considered DH operation are of interest.

| Parameter | Unit | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Location | | Torino | Leini | Sestriere | San Sicario (Cesana Torinese) |
| Average Outdoor Temperature | °C | 13.7 | 13.0 | 4.7 | 3.8 |
| Minimum Outdoor Temperature | °C | -10.8 | -13.1 | -16.5 | -20.6 |
| Maximum Outdoor Temperature | °C | 40.6 | 35.9 | 25.6 | 26.6 |
| Average annual rain | mm | 849 | 1050 | 620 | 836 |
| Average wind speed | m/s | 1.3 | 1.7 | 2.5 | 0.9 |
| Average daily radiation | $MJ/m^2$ | 12.1 | 13.9 | 15.2 | 14.2 |
| Average snow level on the ground | cm | 0 | 0 | 28 | 47 |
| | | | | | |
| Reference years | | 2002-2010 | 2009-2011 | 2006-2009 | 2007-2013 |

Table 2.4 Main weather information obtained from weather stations [19]

A first comparison of the system performances can be made on the specific heat load duration curves. The available curves are shown in Figure 2.8, using a single colour for each DH system.

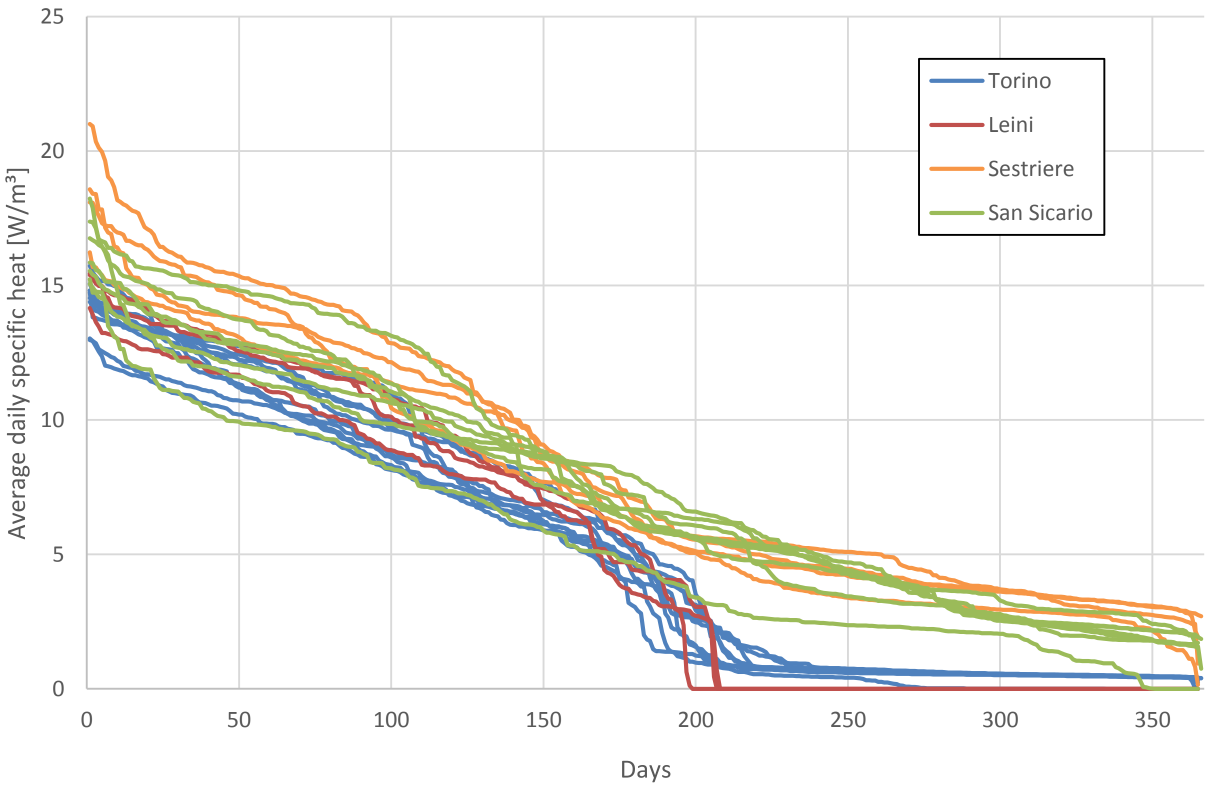


Figure 2.8 Specific heat load duration curves for the case studies

The load duration curves are usually represented on an hourly basis, and they appear more smooth (e.g. see the hourly duration curves for Torino, Figure 2.6). Two main differences stand out from the figure: the *"mountain systems"* (Sestriere and San Sicario) have generally higher specific consumptions than the *"urban systems"* (Torino and Leini) related to climate conditions and the former also show a summer heat consumption that is not present in the latter systems (Leini is totally off during summer, being the domestic hot water provided through distributed generation systems).

The same information can be observed in Figure 2.9, which shows the chronological heat loads for the case studies in available years. However, duration curves are usually more readable and suitable for a graphic comparison, as the effect of anomalous climate conditions for some days tends to decrease the clarity of the representation. Moreover, the *"mountain systems"* have a high dependence of the load on the day of the week, which can produce further oscillations in a chronological plot. This effect, combined with the possibility of very low minimum outdoor temperatures (down to -21°C in Sestriere) results in an higher peak of the duration curves with respect to *"urban systems"*.

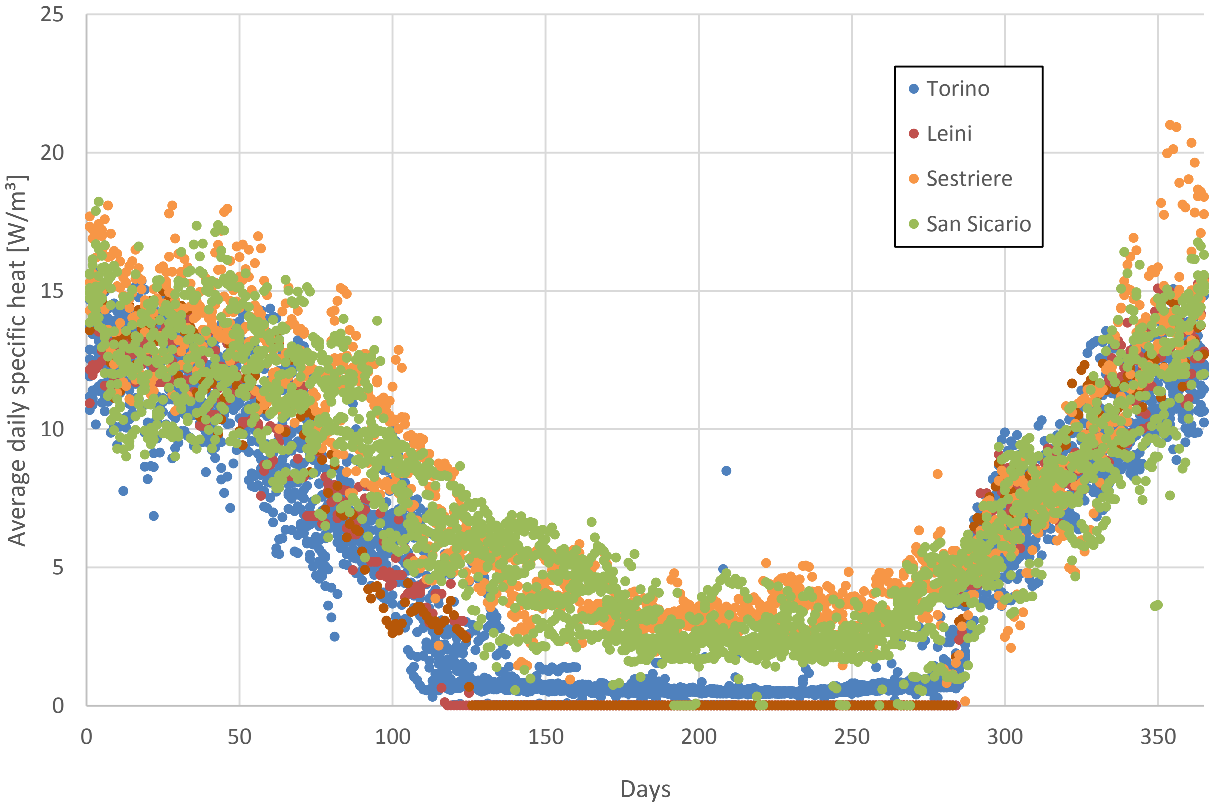


Figure 2.9 Specific heat load chronological curves for the case studies

The specific daily heat consumptions can be plotted versus the average daily outdoor temperatures: the results are shown in Figure 2.10. Once again, the main difference in the point distributions can be seen between the urban systems and the mountain systems.

In the first case the heat consumption shows a good linear correlation with outdoor temperature during the winter season, the oscillations being caused by inertial phenomena and other weather variables (e.g. radiation, snow, etc.) and other non-predictable parameters (e.g. users behaviour). The behaviour of the two DH systems appears totally comparable, despite two order of magnitude of difference between the sizes of the systems. This result shows that the specific heat consumption is a suitable tool for the estimation of the heat load of DH systems with other sizes but similar climate conditions and type of users.

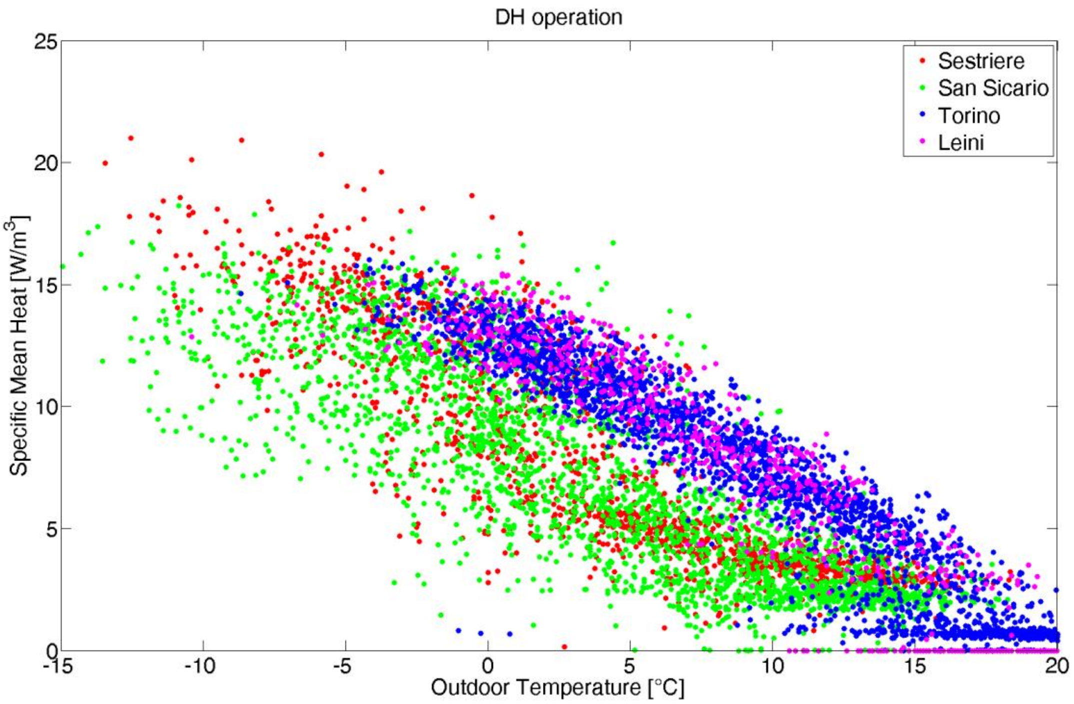


Figure 2.10 Daily heat supplied to the network for different systems

The summer season has low values in the case of Torino, due to the domestic hot water and some hospitals, whereas the DH of Leini is not in operation during the summer season.

In the two other cases the data appear more scattered, and in the meantime the temperatures reach much lower values, due to the climate conditions of the sites. The scattering of the points is related to the characteristics of the users: the buildings connected to Sestriere and San Sicario DH systems are mainly holiday houses and hotels. Their consumptions are highly variable depending on the season, the day of the week, etc. Therefore a simple correlation with respect to climate conditions is not possible, as a part of the variability is related to external factors.

A multiple regression model has been performed to assess the effect of the other weather data of Table 2.4. Although some aspects have been found to increase the accuracy of the correlation (especially solar radiation), a multiple regression model adds a significant degree of complexity to the model, with a lower correspondence with the physical phenomena that it should describe.

These daily operation analyses show that there is a linear relation between heat demand and outdoor temperature, especially for urban DH systems. Other parameters are however affecting the scattering of the points, and in some cases this effect is so high that the linear relations appears no more evident.

Focusing on residential DH systems, additional information can be retrieved from a daily load analysis, which is performed in the case of Torino.

### 2.3.2 Hourly operation analysis

The heat variability over the years has been described in the previous chapters. Another major source of variability is related to the daily behaviour (see paragraph 2.2.3). Although the variability of each day is well represented in Figure 2.11, a general load pattern can be identified among the different daily profiles.

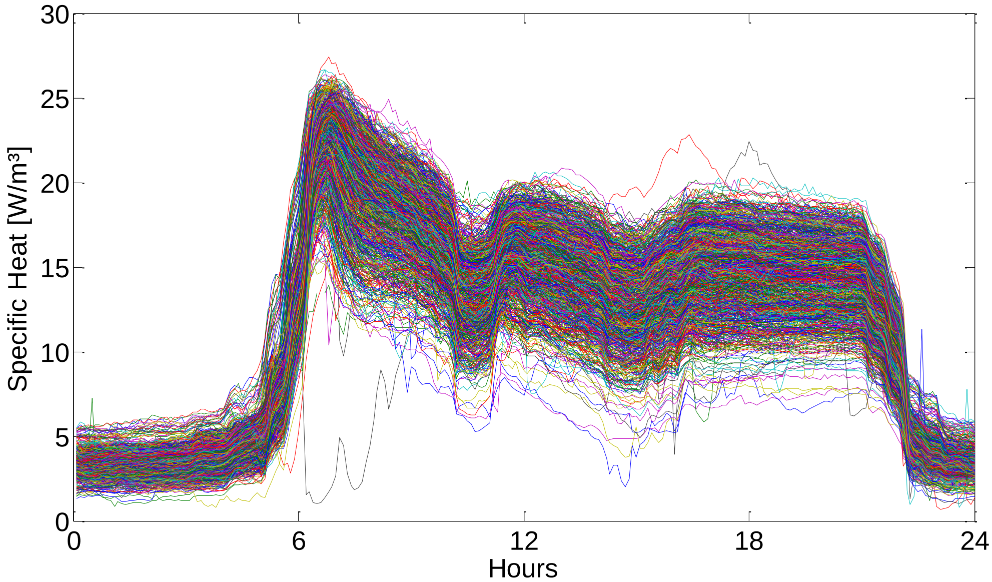


Figure 2.11 Daily DH heat profiles (Torino, December 2009 – February 2010).

This variability is related to different factors, but the most important is probably the control setting of the heating systems of the buildings. The control logic is usually not constant over the day, as the day-time and night-time set-point temperatures are different. As a result, the first hours of the day show a significant peak of the heat demand. This aspect can be clearly appreciated in Figure 2.12: the higher consumptions are in the morning hours (5 AM to 9 AM), while at night (10 PM to 4 AM) the specific heat is much lower. The average consumption during the day is lower, as the morning peak is caused by the need of re-heating the building to the daily set point.

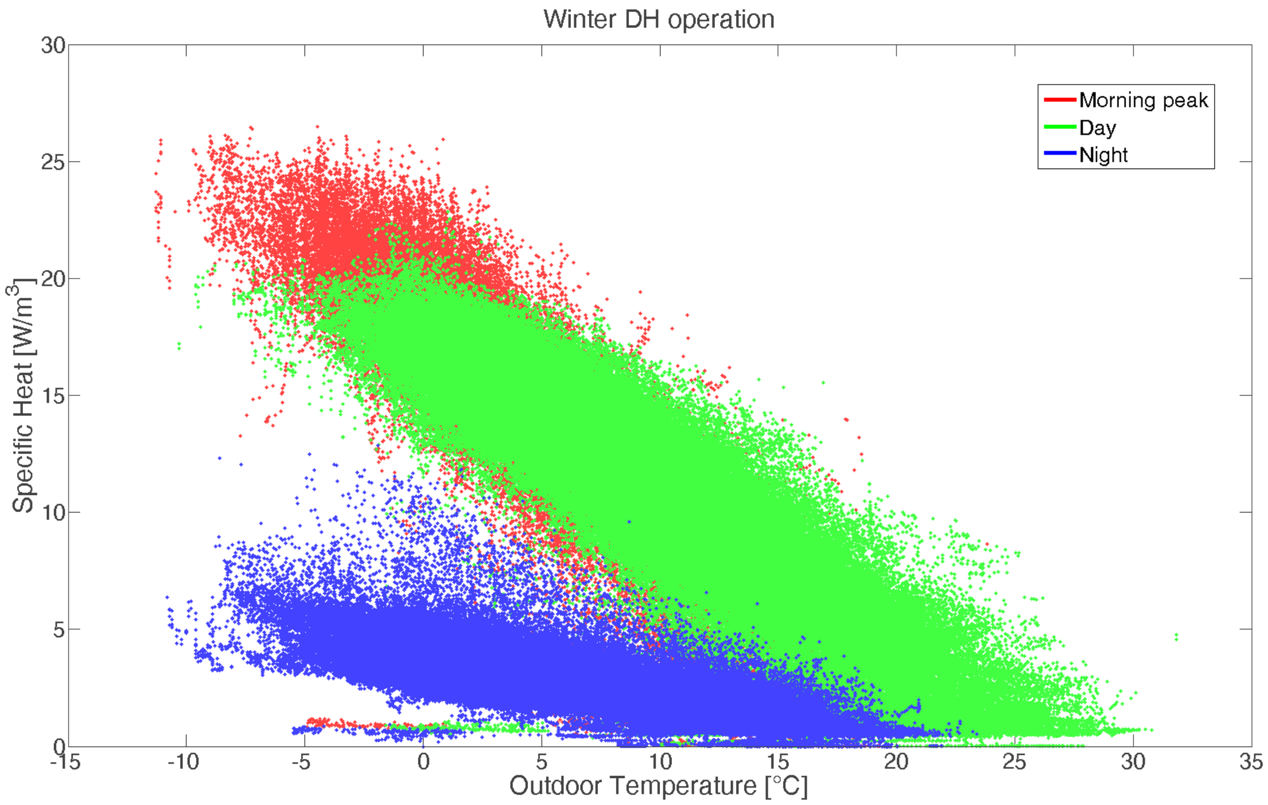


Figure 2.12 Specific heat vs Outdoor temperature for different hours of the day.

However, this variability is not trivial to describe with a model, as also the inertia of the buildings plays a major role at these time steps. The definition of an exact model of the heat demand of the buildings is beyond the scope of this PhD work.

Therefore, the real specific heat data have been used as a standard dataset in the simulation model, in order to describe a standard urban DH system starting from the knowledge of the volume of buildings supplied by the DH system.

This choice allows to take into account multiple real operation conditions, which could not be obtained through the definition of an approximated model. These data will be used as an input for the simulation, calculating the operation of the heat generation units starting from the heat demand that is required by the users of the DH network.

### 2.3.3 DH Network heat losses

The heat profile that has been described previously, represents the heat produced by the generators, thus including the heat losses in the network. An aspect to be taken into account during the analysis of a district heating system is related to the heat losses of the network, which in some cases rise to significant levels.

The heat losses of a DH system are related to multiple parameters:

- total length of DH network pipes;
- size of the pipes;
- typology of the pipes and insulation materials;
- supply and return temperatures of the water;
- ground temperature and depth of installation.

Some parameters are related to design and installation phases, and they are often dependent on economic considerations related to the investment cost of the system. On the other hand, other aspects are related to the operation conditions of the DH network. Heat losses during operation can be different from expected design calculations. This can be related to the fluctuating temperature difference between the hot water and the ground. Moreover, the deterioration of the insulation coating over the time is a crucial issue for old systems. In some cases, additional sources of losses can be related to faulty connections between pipes leading to a progressive deterioration of the coating.

Figure 2.13 shows the distribution of the average annual grid losses of the Italian DH systems (from [11], year 2012).

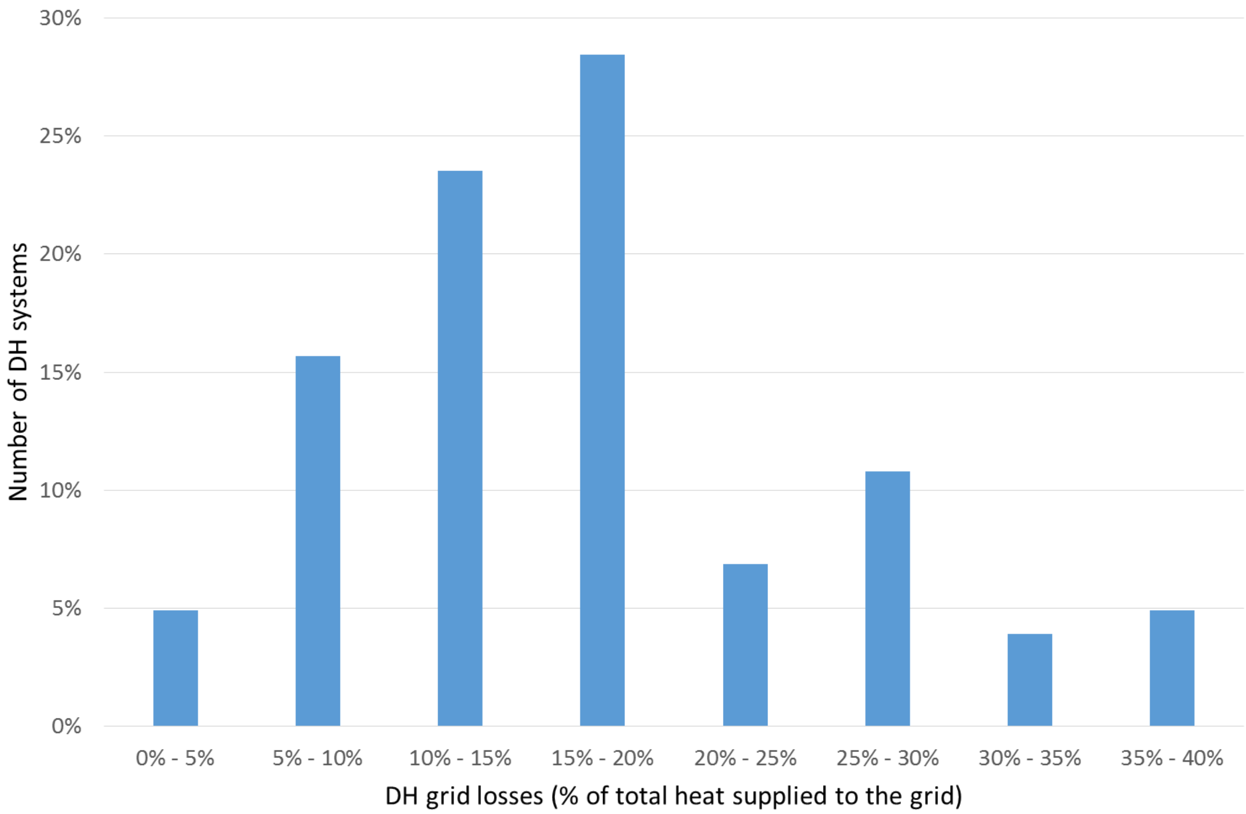


Figure 2.13 Average heat losses in Italian DH systems (author's calculation from [11])

Almost 75% of the DH systems have losses lower than 20% of the total heat supplied to the grid. However, some systems reach values near 40%. The DH networks with higher losses are all supplying small volumes of users (lower than 0.5 ÷ 1 $Mm^3$). One possible reason can be the lower density of the buildings supplied by the DH system. Other causes can be related to the operation conditions of the network, or to the design parameters of the insulation of the pipes.

Considering the entire energy production in DH systems in Italy, the average losses have been about 16% in the year 2013. Table 2.5 shows the evolution in the last six years. The average performance of DH networks appears to worsen over the years. This can be related to the increasing number of DH networks in mountain regions, which are typically supplying smaller buildings with a low density with respect to the DH network. Another possible reason can be the increasing of real measurements of the energy produced and supplied, resulting in the availability of data that were previously obtained from approximated estimations.

| | Heat produced [GWh] | Heat supplied to users [GWh] | Heat losses [%] |
|---|---|---|---|
| **2013** | 10,966 | 9,200 | 16.1 % |
| **2012** | 9,533 | 8,005 | 16.0 % |
| **2011** | 8,645 | 7,322 | 15.3 % |
| **2010** | 8,999 | 7,746 | 13.9 % |
| **2009** | 7,786 | 6,734 | 13.5 % |
| **2008** | 7,095 | 6,257 | 11.8 % |

Table 2.5 Heat produced and supplied to final users in Italian DH systems [16]

The average losses for Italian systems are in line with the data available from other European countries [20]. Denmark has an average of 16% of losses as well, while in Sweden this share increases to 18%. Finnish DH systems appear to have a better performance, losing only 8% of the total heat produced by the generation units.

While annual heat losses can be calculated as the difference between the total heat produced and the final energy supplied to the users, which are both measured, there is little information about the variation of the losses during the year. Some considerations can be performed for the DH system of San Sicario, where a daily measurement of the produced and supplied energy is available for some years of operation (from 2010 to 2012). The Table 2.6 shows the annual performance of the DH system, by listing the total heat produced, the heat supplied to the users and the heat losses.

| | Heat produced [GWh] | Heat supplied to users [GWh] | Heat losses [GWh] | Heat losses [%] |
|---|---|---|---|---|
| **2010** | 25.09 | 22.07 | 3.01 | 12.0% |
| **2011** | 22.51 | 19.89 | 2.63 | 11.7% |
| **2012** | 23.97 | 20.39 | 3.58 | 14.9% |

Table 2.6 Comparison of heat produced and supplied to final users in San Sicario

The analysis of daily measurements can provide some additional information. The behaviour of the heat demand during the year is shown in Figure 2.14, revealing the presence of significant variations from one year to another. The DH system of San Sicario supplies a mountain village that has a ski resort, resulting in a considerable variability of the heat consumptions. As a consequence, the heat load during the winter cannot be correlated to the outdoor temperature or other parameters, as the behaviour of the users has a major effect and cannot be taken into account in a simple way. This aspect needs to be kept in mind during the following considerations.

The major discontinuities are related to some particular holydays or weather conditions. February has usually high consumptions due to the presence of skiers, and the lower consumptions of 2011 are due to ten days of anomalous weather conditions, with outdoor maximum temperatures that reached 12°C (the first two weeks of February had an average of 0°C in 2011, of -8°C in 2010 and -12°C in 2012). The unusual peak in the second half of July 2011 was due to anomalous cold conditions, as the temperature dropped to average daily values lower than 10°C. Finally, the four blue dots that can be noticed in the beginning of April 2010 are due to the weekend of Easter, which saw a higher presence of tourists and consequently a higher heat request.

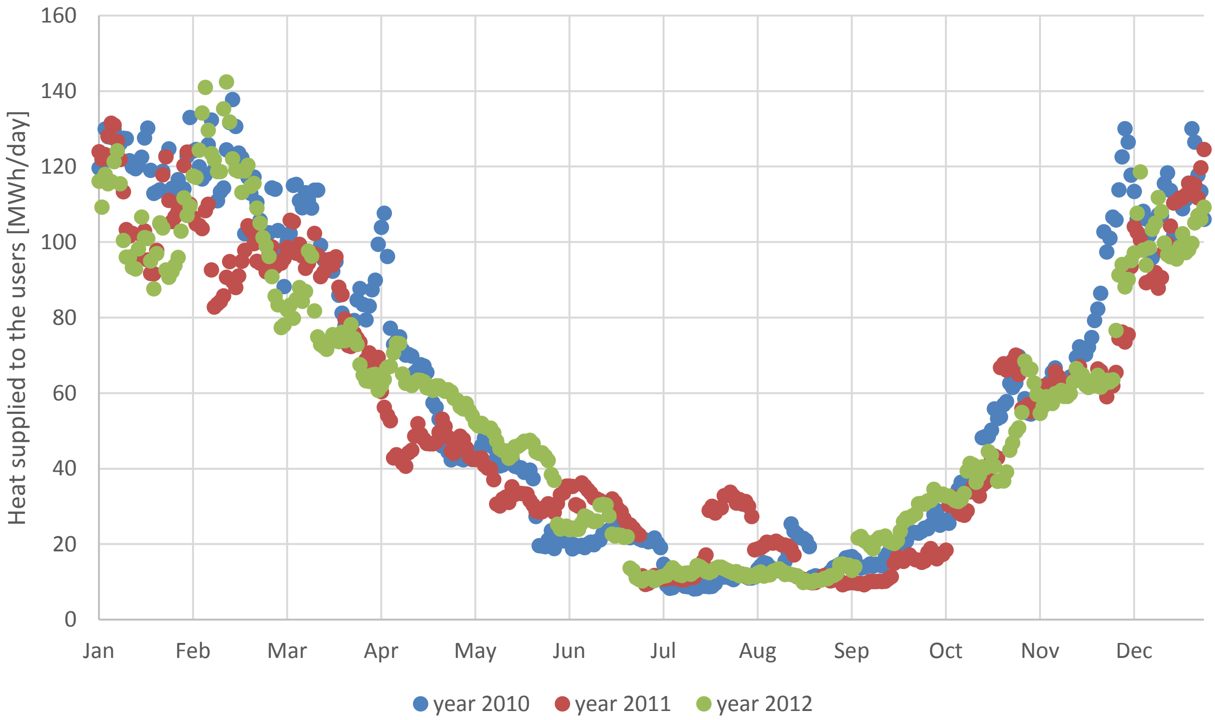


Figure 2.14 Daily heat supplied to the users in San Sicario DH system

The variation of the network heat losses over the year is shown in Figure 2.15, considering the total daily heat losses, and in Figure 2.16 as a share of the total heat supplied to the network. During the winter the absolute losses show a certain scattering and the losses share is more comparable among different years, and in the summer season just the opposite happens.

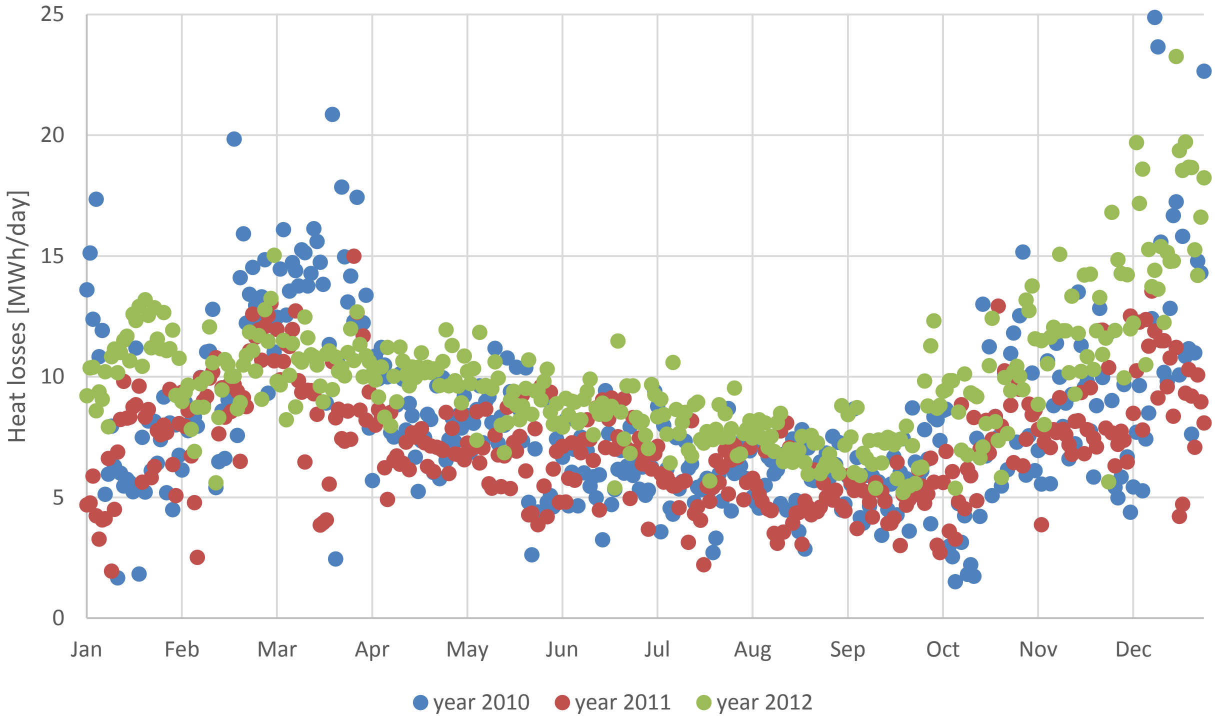


Figure 2.15 Daily heat losses in San Sicario DH system (absolute values)

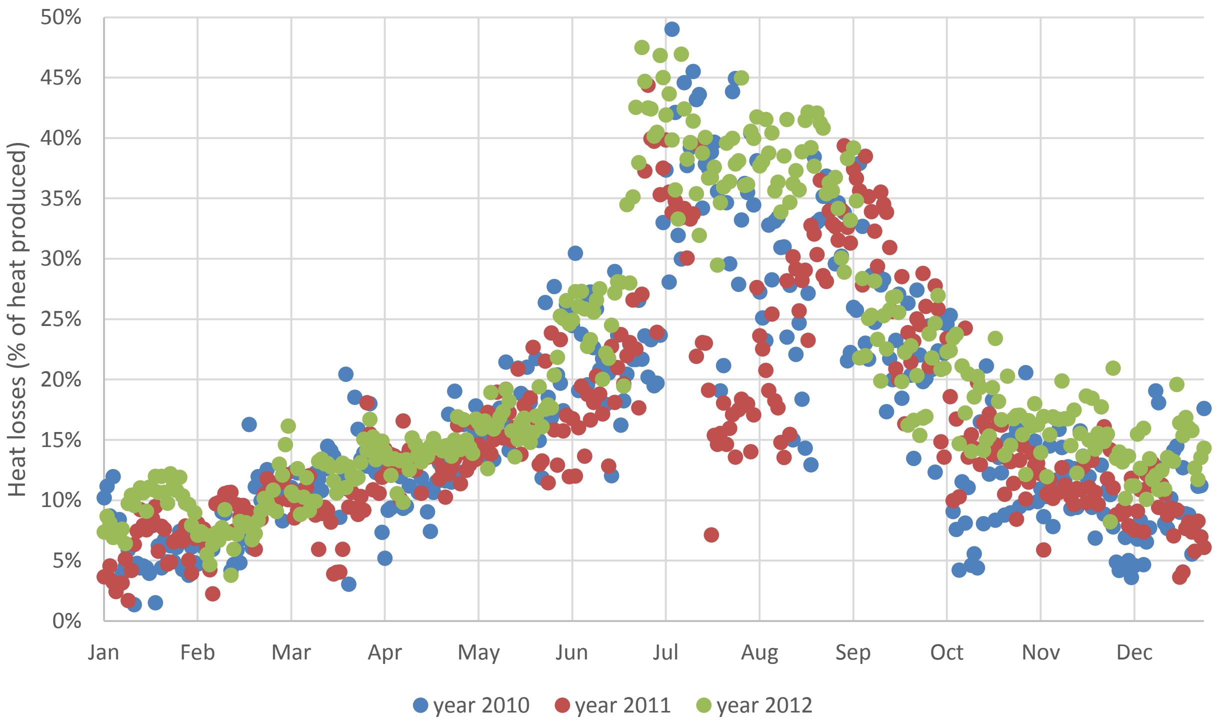


Figure 2.16 Daily heat losses in San Sicario DH system (% of heat produced)

As mentioned before, the losses are caused by a number of aspects, and is not trivial to find reliable correlations with few parameters. However, a general trend show lower losses during the summer season, due to a narrower temperature difference between the ground (that is warmer than during winter) and the network temperatures (that are colder than during winter). Obviously, the increase of the share in summer is due to the low heat supplied to the users. It deserves to be mentioned that during the summer the heat losses can reach 35% ÷ 45% of the total heat supplied to the DH network, and therefore other solutions could be evaluated in order to increase the overall system efficiency (e.g. distributed solar collectors for domestic hot water). However, in this particular climate conditions the system may need to provide some space heating also during summer (see July 2011 in Figure 2.14), and therefore the total shut down of the system is probably not possible.

# Chapter 3 Supply side: Organic Rankine Cycle simulation

The CHP technology most diffused in wood biomass CHP systems lower than 2 $MW_{el}$ is based on Organic Rankine Cycles. This is a well-proven technology, which is also applied to other heat sources: geothermal, solar thermal and waste heat recovery. Biomass systems are currently the first ORC application in terms of number of installed units [21]. ORC manufacturers are on the market since the beginning of the 1980s, and the number of installed units is increased with an exponential trend in last ten years.

## 3.1 Description of the ORC systems

The current ORC systems are the result of an evolution of more than a century, with different applications and approaches, resulting in a variety of technical solutions that can provide an interesting concept for the energy generation from renewable sources.

### 3.1.1 Brief history of ORC systems

The Rankine cycle is named after William J. M. Rankine (Scotland, 1820 – 1872), a Scottish engineer, mathematician and physicist, who was a founding contributor to the science of thermodynamics. The Rankine cycle is the most diffused process used by steam-operated heat engines to generate power. The larger part of electricity in the world is currently generated through steam turbines power plants.

The first applications of Rankine cycles with fluids different than water appeared already in the 19$^{th}$ century. Thomas Howard patented the idea of using ether as working fluid for an engine in 1826 [22]. However, the first commercial application of non-aqueous fluids has been the use of naphtha engines to power small boats, called naphtha launches. The naphtha, a form of gasoline, was used as cycle fluid, as lubricant and as fuel for the boiler. Although the first naphtha launch has been built in United Kingdom (the *Zephyr* invented by Alfred Yarrow), the solution had some success only in the US due to a law that required all steam boats to carry a licensed engineer at all times. This regulation had little effect on commercial crafts, where the cost of such an engineering was marginal, but small boat's owners were interested to non-aqueous solutions. The Gas Engine & Power Company of New York claimed to have sold over five hundred ORC engines based on the design of Ofeldt in 1890 [23].

However, the modern ORC technology has been developed in the 20th century. The first concept of a solar ORC engine has been proposed by Frank Shuman in 1907, who boiled ether at temperatures of about 120°C with a 110 $m^2$ flat solar collector. A great contribution to ORC technology has been provided by Professor Luigi D'Amelio (1893 – 1967), chair of thermal and hydraulic machinery at the University of Naples. In 1936 he won a prize proposing a solar power plant for the irrigation of arid areas in Libya, using monochloroethane as working fluid [24], with a net power output of 4 $kW_m$ and a net conversion efficiency of 3.5%. He also outlined the main principle of ORC systems, including turbine design and working fluid selection. His experience led also to the development of a prototype for the conversion of low-temperature geothermal energy in a laboratory of the University of Naples, and a geothermal ORC pilot power plant of 11 $kW_m$ on the island of Ischia in 1940.

The first commercially operated ORC plants were based on geothermal energy sources. In 1952 a plant of 200 $kW_{el}$ supplied power to a mining company using water at 91°C as heat source, but was shut down after some years. In Kamchatka peninsula in 1967 a geothermal ORC plant using refrigerant R12 supplied electricity to a small village and heat to some greenhouses, having a rated power of 670 $kW_{el}$ [25]. After the energy crisis of the 70's, which led to the investigation of alternatives energy sources, many geothermal ORC units were developed, also increasing in size. Other renewable sources were investigated, especially industrial waste heat and exhaust gases from engines.

The first biomass-fired ORC plant has been installed in Bière, Switzerland, in 1998, supplying electricity and cogenerated heat to a barrack of the Swiss Army [26]. The generator was a 2-stage axial turbine with a rated power of 300 $kW_{el}$, and the cycle used the siloxane MDM.

In last decades the ORC has become a reliable and well diffuse solution for energy generation, and a number of different companies are currently available on the market. The ORC market has been supported by the increasing interest in the development of renewable energy sources and the increase of energy efficiency through the recovery of waste heat, especially in industrial applications and large engines (e.g. for marine propulsion).

### 3.1.2 State of the art of commercial ORC systems

The ORC systems can vary considerably due to the high number of technical options that can be chosen, like different fluids, expanders, system layouts, etc. These options are often tailored on the specific application, based on the type of thermal source, heat source and cooling fluid temperatures, plant capacity, and other relevant parameters. The design of an ORC unit requires an iterative approach which considers multiple aspects, as the design of each component of the system can strongly depend on the others. There is a large amount of possible combinations of technical solutions, resulting in the need of increase the automatic optimization of the design phase. The main degrees of freedom lay in system configuration, working fluid, and design of expander, pump and heat exchangers.

Current commercial applications are using saturated and superheated cycles, whereas only in some rare cases supercritical cycles have been chosen. The main limit lays in the consumption of the feeding pump, as in supercritical cycles it increases at unacceptable levels. In case of expansion in a turbine a little degree of superheating is required, in order to avoid the presence of liquid droplets during the expansion. There are basically three different cycle layouts, represented in Figure 3.1. The first layout is the simpler, being composed by an evaporator, a turbine, a condenser and a feeding pump. In the second layout a regenerator is used, in order to recover a part of the heat of the working fluid at the turbine exit. The choice between layouts "A" and "B" is mostly related to the kind of working fluid. The third layout is a further evolution, which involves the use of a pre-heater, working in parallel with the regenerator. A part of the fluid is heated through an additional external heat source: this component allow to recover heat at lower temperatures than in the evaporator, and therefore it is particularly useful in some applications (e.g. biomass combustion).

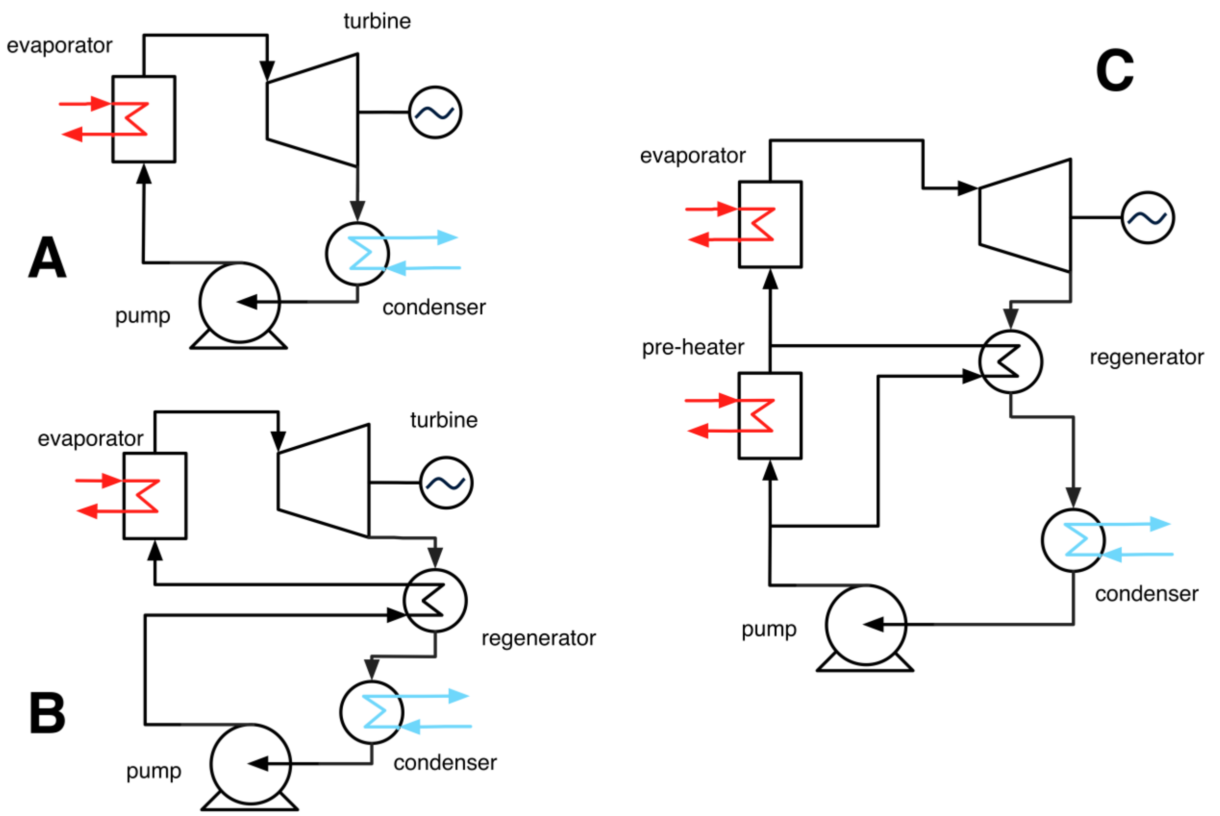


Figure 3.1 Different cycle layouts: A) single cycle B) with regenerator C) "split" cycle

The expanders that are used in commercial ORC systems are almost always turbines, while volumetric expanders (scroll or screw) are used at very small scales in pilot systems. The current turbines applications span from about 100 $kW_m$ to some $MW_m$, with turbine inlet temperatures from about 110°C to 350°C. The volumetric expanders are applied to the lower limits of power and temperatures, with the possibility of lowering the technical limit, both in terms of temperature and available heat. They have generally lower isentropic efficiencies, but as they are generally derived from refrigerant compressors they can be cost-effective. Moreover, they have the advantage of allowing the presence of a part of liquid during the expansion process.

Medium to large size ORC systems are generally using axial turbines in single or multi-stage configurations. Their isentropic efficiency can range from 80% to almost 90% in nominal conditions. Smaller systems are usually provided with radial inflow turbines, achieving higher efficiencies with a single stage. In these cases the optimal rotational speed can be of 10,000 ÷ 40,000 rpm.

The ORC pump is usually larger than in comparable steam plants, as the ratio of pumping work over turbine work is higher. In some cases this requires to design ad-hoc pumps in order to reach acceptable efficiencies. This is one of the crucial aspects limiting the use of supercritical ORC applications, which would require too high pumping consumptions.

The heat exchangers can be the most significant part of an ORC system. The evaporator, which is used to transfer heat from the primary source to the working fluid, can be directly or indirectly connected. In the latter case an intermediate thermal loop is needed, often using thermal oil. The choice of the optimal configuration depends on many aspects. Direct connection increases the efficiency of the cycle, as it allows higher temperature and pressure at the turbine inlet. The use of an intermediate loop can avoid hot spots, which can lead to working fluid decomposition, and in some cases is required by safety regulations or contractual concerns. However, it requires an additional pumping consumption and a temperature loss. The main parameter is often the operating cost, which lead high temperature cycles to be operated with indirect connection due to the higher cost of the working fluid with respect to thermal oil. The use of regenerator depends on the kind of working fluid and cycle parameters. As the turbine outlet can be far from condensing conditions, a significant amount of heat can be recovered in a regenerator in order to increase the cycle efficiency and to lower the size of the condenser, which can be a very critical aspect in the system design. As in steam plants, the water-cooled condensers have generally higher performances, resulting in higher cycle efficiencies. They require the availability of a water sink, which is not always available. The wet cooling is also used when the ORC system is connected to a district heating network or to another heat circuit, usually at medium or high temperatures (up to 80÷90°C). The air coolers are often used with an intermediate water loop, in order to limit the total working fluid volume and possible leakages.

The characteristics of an ORC unit are strongly related to the type of application. Currently the commercial ORC systems have four main applications for CHP generation:

- low temperature geothermal plants;
- solar plants;
- waste heat recovery from industries or engine exhaust gases;
- biomass plants.

Geothermal plants are using ORC where the size or the temperature of the source have not the characteristics requested for the installation of a steam plant. These systems usually require a high customisation, due to the large variability of the characteristics of the heat source. As a result, a variety of working fluids and system configurations can be used. The available output power can reach several $MW_{el}$, and in some cases they substitute steam plants for safety or regulatory reasons. These systems are affected by a significant auxiliary consumption, which can reach up to 50% of the gross output power [27]. In some large plants with high temperature geothermal heat source the heat at the condenser is recovered for supplying district heating networks, increasing the overall efficiency of the system. However, in this case the electric efficiency is slightly lower.

Solar plants have been among the first applications of ORC systems (see paragraph 3.1.1). ORC units are used in concentrated solar power plants, which can be composed by parabolic dishes, solar towers or parabolic trough. These systems are usually coupled to steam Rankine cycles or to Stirling engines (at smaller scales). ORC technology is being applied to low temperature solar plants, but it still remains a marginal sector with little commercial investments. The larger example of ORC solar plants has been built in 2006 in Arizona, with a cycle efficiency of 20%, but a total solar to electricity efficiency of 12.1% [21]. The main competitor of solar plants is the photovoltaic technology, which is currently the most diffused solution for power production from solar radiation, with higher efficiencies and lower costs than solar thermodynamic power plants.

Waste heat recovery is among the most interesting applications of ORC systems, both from the industry sector and from the engine exhaust gases. Industrial systems require high customisations, depending on the characteristics of the thermal source, in terms of temperatures, fluids and operation cycles. The use of engine exhaust gases is more suitable to be standardized, and some manufacturers propose ORC units that have been designed to be coupled to a specific engine. These units are typically with a small output power (from 50 to 150 $kW_{el}$) and small efficiencies. They are generally providing an additional output power for stationary engines for electricity generation, but a number of units have been used in ship engines.

Solid biomass ORC systems have usually a certain degree of standardization. The advantage of using ORC is the possibility of lowering the average size of the plant, in order to perform a sustainable gathering of local biomass, with limited environmental impacts and costs related to the transport phase. These units are usually using a higher temperature source than the other applications, thanks to the possibility of reaching high temperatures with biomass combustion. At the same time, the optimal use of the biomass requires the flue gases to exit the boiler at the lower possible temperature, avoiding the condensation of the acid compounds originated during the combustion process. For this reasons, usually a preheater is installed in parallel with the regenerator, in order to recover a part of heat also at lower temperature. These configuration is sometimes referred to as "split system". Current biomass systems are usually designed with an intermediate loop with thermal oil, to avoid the design of a specific boiler which would need to be tailored on the characteristics of the working fluid and to avoid the formation of hot spots. However, some manufacturers are starting to develop a direct connection with the working fluid, in order to allow a higher temperature in the turbine.

Biomass ORC systems are the only application which actually use a fuel that has a cost, both from economic and environmental point of view. Whereas sun or geothermal heat are free of charge, the input biomass need to be gathered, transported and supplied to the system. In biomass ORC systems the efficiency of the conversion process becomes particularly important. In most cases, a simple power generation from biomass is neither environmentally nor economically sustainable. While in other ORC applications the cogeneration is quite uncommon, when considering biomass systems it becomes the most diffused configuration. The CHP production is promoted by the European Union, for the opportunity to save energy and combat climate change (EU Directive 2004/08/EC and amending Directive 92/42/EEC).

### 3.1.3 ORC manufacturers and market

The current ORC market is the result of an evolution started in the beginning of the 1980s. A list of some main manufacturers is provided in Table 3.1.

| Manufacturer | Applications | Power Range [$kW_{el}$] | Heat source temperature [°C] | Technology |
|---|---|---|---|---|
| **ORMAT, US** | Geo., WHR, solar | 200-70,000 | 150-300 | Fluid: n-pentane and others, two-stage axial turbine, synchronous generator |
| **Turboden, Italy** | Biomass, WHR, Geo. | 200-2,000 | 100-300 | Fluids: MDM, MM, Two-stage axial turbine |
| **Adoratec/Maxxtec (*now Siemens*), Germany** | Biomass | 315-1,600 | 300 | Fluid: MDM |
| **Opcon, Sweden** | WHR | 350-800 | <120 | Fluid: Ammonia, Lysholm Turbine |
| **GMK, Germany** | WHR, Geo., Biomass | 50-5,000 | 120-350 | 3000 rpm Multi-stage axial turbines (KKK) |
| **Bosch KWK, Germany** | WHR | 65-325 | 120-150 | Fluid: R245fa |
| **Turboden PureCycle, US** | WHR, Geo. | 280 | 91-149 | Radial inflow turbine, Fluid: R245fa |
| **GE CleanCycle, US** | WHR | 125 | >121 | Single-state radial inflow turbine, 30,000rpm, Fluid: R245fa |
| **Access Energy, US** | WHR | 125 | 95-170 | Single-state radial inflow turbine, 30,000rpm, Fluid: R245fa |
| **Cryostar, France** | WHR, Geo. | n/a | 100-400 | Radial inflow turbine, Fluids: R245fa, R134a |
| **Triogen, NL** | WHR, Biomass | 160 | >350 | Radial turbo-expander, Fluid: Toluene |
| **Electratherm, US** | WHR, Solar | 50 | >93 | Twin screw expander, Fluid: R245fa |
| **Exergy, Italy** | Biomass, WHR, Geo., solar | 125-22,500 | n/a | Fluids: perfluorocarbons, siloxanes, radial outflow turbine |
| **Zuccato Energia, Italy** | WHR | 30-50 | >94 | Fluid: HFC mix, Radial, fixed nozzles, directly coupled to generator |

Table 3.1 Main ORC manufacturers (integrated from [21]).

The main players are currently Ormat and Turboden (Mitsubishi Group), which are representing together more than 90% of the current installed power and number of units [21].

Ormat Technologies Inc. is the larger ORC manufacturer in terms of installed power, with more than 1,850 MW of geothermal and recovered energy power plants in more than 75 countries [28]. The company was established in 1965 as Ormat Turbines Ltd., in Yavne, Israel, by engineer Lucien Bronicki (Chairman and CTO) and wife Yehudit "Dita" Bronicki (CEO).

Turboden is the leading company for installed units, with 298 plants in 32 countries, for a total power of 408 $MW_{el}$ [29]. The prevalent application is biomass, with more than 250 units. The company was founded in 1980 in Milan by Mario Gaia, Professor of Energy at the Politecnico di Milano, and today Managing Director of Turboden. In 2009, Turboden became part of UTC Corp., a worldwide leader in development, production and service for aero engines, aerospace drive systems and power generation gas turbines. In 2013 UTC exits the power market forming strategic alliance with Mitsubishi Heavy Industries and Pratt & Whitney Power Systems (now PW Power Systems) with the affiliate Turboden become an MHI group company. Today Turboden s.r.l. and PW Power Systems, Inc. are Mitsubishi Heavy Industries group companies, with whom MHI is able to provide a wider range of products and services for thermal power generation systems.

Other companies are emerging and gaining a share of the market, due to the growing interest in renewable technologies and energy efficiency. Calnetix Technologies patented a small-size ORC unit which can be used for heat recovering at low temperatures. An integrated power module, consisting of a high-speed turbine expander and three integrated proprietary technologies, which produces 125 $kW_{el}$ of gross power with input temperatures down to 120°C [30]. Another emerging company that invested in small-size ORC systems is Triogen, which had developed a 160 $kW_{el}$ ORC unit for waste heat recovery with direct coupling with the flue gases. The lack of an intermediate cycle allows for higher efficiencies, as the toluene used as working fluid can enter the turbine at higher temperature. The system has a nominal electric efficiency of 17% in full-electric mode and 14% when producing hot water at 80°C. These units are mainly installed in biogas or natural gas engines, but some applications are running on solid biomass [31].

## 3.2 Choice of the working fluid

The degree of freedom given by the possibility of choosing the working fluid is among the main advantages of ORC systems. The fluids suitable for ORC applications can be chosen from a wide variety of substances, and new fluids and mixtures are still under development and research. The choice of the optimal fluid need to take into account multiple aspects. The most significant are the following (reported from [21]):

- Thermodynamic performance: depending on the application, the main goal is to optimize the cycle efficiency or the available output power. These features are both depending on multiple independent fluid parameters, such as critical temperature, critical pressure, density, specific heat, etc. During the design phase usually the cycle performance simulation is performed with different fluids comparing the results.

- Saturation vapour curve: an advantage of ORC systems is the possibility of using a fluid with a positive or isentropic saturation curve. A negative saturation vapour curve (‘‘wet’’ fluid) leads to droplets in the later stages of the expansion, with the need of a significant superheating of the vapour (as it usually happens with steam power plants). In the case of a positive saturation vapour curve (‘’dry’’ fluid), a regenerator is usually used in order to increase cycle efficiency.

- Vapour density: this parameter is of key importance, especially for fluids showing a very low condensing pressure (e.g. siloxanes). A low density leads to a higher flow rate, with the consequent need of increasing the size of the heat exchangers for lowering the pressure losses. The increase of heat exchangers is basically an economic problem, as they usually are among the most expensive components of the system. On the other hand, higher flow rates can simplify the design of the turbine.

- Evaporation pressure: the evaporation pressure has an impact on the cost of the system components, especially pipes and exchangers. As in steam power plants, the maximum pressure need to be acceptable compared to the additional component costs.

- High temperature stability: the maximum primary heat source temperature is usually limited by the chemical stability of the working fluid, which can deteriorate at high temperatures. This is one of the reasons for the use of an intermediate loop between the primary heat source and the working fluid, in order to avoid hot spots that could threaten the fluid stability.

- Conductivity and viscosity: low viscosity and high conductivity results in high heat transfer coefficients in the heat exchangers. An additional advantage of low viscosity lays in low friction losses through the cycle.

- Safety level: safety involves two main parameters— toxicity and flammability. This is a common issue as synthetic fluids can be both toxic and flammable. There are different regulations and classifications for safety issues, depending on the country. This parameter should be taken into account when comparing different fluids, also considering the safety measures that are required both in case of emergency and in normal operation of the plant (e.g. additional required security systems).

- Environmental aspects: the two main environmental aspects are measured by the Ozone Depleting Potential (ODP) and the Greenhouse Warming Potential (GWP). The ozone depleting potential is expressed in terms of the ODP of the R11, set to unity. The Montreal Protocol has banned fluids that can damage the ozone layer, therefore the current refrigerants have an ODP almost null. The global warming is an increasing concern for the environment. However, no regulation is currently limiting the use of any fluid based on its GWP value. GWP is measured with respect to the GWP of $CO_2$, chosen as unity.

- Availability and cost: last but not least, the cost and availability need to be taken in serious consideration. This is an aspect which is usually forgotten during thermodynamic simulations for energy performance, but can become determinant for the commercial application of some units. Due to the relatively small size of the ORC sector, fluids already used in other applications (e.g. refrigerants) are usually preferred for their lower prince and higher availability.

The scientific literature covers a wide range of fluids, and many comparison and thermodynamic analysis have been performed. However, commercial applications are generally limited to few working fluids:

- R-134a: is refrigerant used in geothermal systems or in low temperature heat recovery. Its IUPAC name is 1,1,1,2-Tetrafluoroethane, and its molecular formula is $CH_2FCF_3$. It is a haloalkane refrigerant with thermodynamic properties similar to R-12 but with insignificant ozone depletion potential. It is primarily used as a refrigerant for domestic refrigeration and automobile air conditioners. Its significant global warming potential (100-yr GWP = 1430) has lead it to be banned from use in the European Union, starting with cars in 2011 and phasing out completely by 2017.
- R-245fa: low temperature working fluid used mainly for waste heat recovery. Also known as 1,1,1,3,3-Pentafluoropropane, with the formula $C_3H_3F_5$, is used primarily as spray foam insulation and marketed by Honeywell under the Genetron 245fa brand name. It is a colourless gas with no ozone depletion potential and nearly non-toxic. It has a 100-years GWP of 950, which is relatively high.
- MM: it is an organosilicon compound mainly used in waste heat recovery, also called hexamethyldisiloxane, or HMDS (with the formula $O[Si(CH_3)_3]_2$). It is a volatile highly flammable colourless liquid, which can cause serious eye irritation. It is primarily used as a solvent and as a reagent in organic synthesis.
- MDM: it is a linear siloxane widely used in biomass CHP plants, also known as octamethyltrisiloxane, or OMTS (with the formula $C_8H_{24}O_2Si_3$). It is a flammable colourless liquid, not classified for human health effects, mainly used in coatings, sealants and personal care products.
- Toluene: also known as methylbenzene (IUPAC name), its main applications in ORC systems are for waste heat recovery and some biomass CHP plants. It is a colourless liquid widely used as an industrial feedstock and as a solvent. It is a mono-substituted benzene derivative, consisting of a $CH_3$ group attached to a phenyl group (molecular formula: $C_7H_8$). It is highly flammable and it has a moderate degree of toxicity.
- n-pentane: used in waste heat recovery, medium temperature geothermal and the only commercial solar ORC. It is an alkane with five carbon atoms, with the molecular formula $C_5H_{12}$. It is a colourless and odourless liquid, and it is mostly employed as specialty solvents, with properties similar to butane and hexane. It is highly flammable, and exposure could cause irritation but with only minor residual injuries.

The simulations performed in this work have been applied to these fluids. However, the methodology can be transposed to each other compound without significant differences.

The main thermodynamic characteristics of these fluids are listed in Table 3.2, while Figure 3.2 T-s characteristics for some ORC fluids. Figure 3.2 shows their saturation curves compared with water in a T-s diagram. All the fluids have dry or isentropic vapour saturation curves, and the critical temperatures are lower than for water, which has a $T_{CR}$ of 374°C, showing a large variability from about 100°C (for R-134a) to over 300°C (for toluene).

| Fluid name | Chemical formula | Molar mass [g/mol] | $T_{CR}$ [°C] | $p_{CR}$ [bar] | $\rho_{CR}$ [kg/m³] | Boiling temperature [°C] |
|---|---|---|---|---|---|---|
| **R-134a** | $CF_3CH_2F$ | 102.0 | 101.1 | 40.6 | 511.9 | -26.1 |
| **R-245fa** | $CF_3CH_2CHF_2$ | 134.1 | 154.0 | 36.5 | 516.1 | 15.1 |
| **MM** | $C_6H_{18}OSi_2$ | 162.4 | 245.6 | 19.4 | 258.6 | 100.3 |
| **MDM** | $C_8H_{24}O_2Si_3$ | 236.5 | 290.9 | 14.1 | 256.7 | 152.5 |
| **Toluene** | $CH_3$-$C_6H_5$ | 92.1 | 318.6 | 41.3 | 292.0 | 110.6 |
| **n-pentane** | $CH_3$-3($CH_2$)-$CH_3$ | 72.1 | 196.6 | 33.7 | 232.0 | 36.1 |

Table 3.2 Thermodynamic characteristics of the most common ORC fluids (data from Refprop [32]).

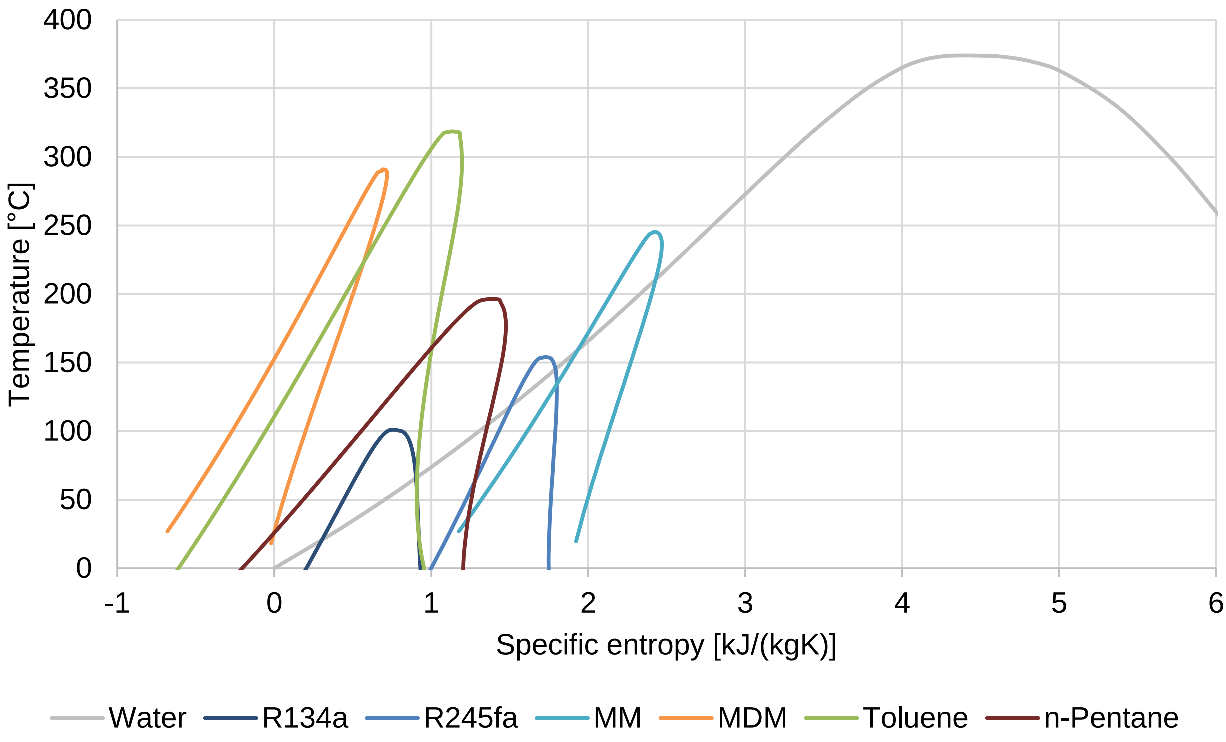


Figure 3.2 T-s characteristics for some ORC fluids.

The environmental and hazard characteristics have been listed in Table 3.3. The hazard classification has been reported as defined by the US National Fire Protection Association (NFPA 704) Hazard Identification System. Three main aspects are considered: health, flammability, and reactivity. Each classification uses a numbering scale ranging from 0 to 4. A value of zero means that the material poses essentially no hazard; a rating of four indicates extreme danger. There are other hazard classifications, depending on the country of application or on the specific sector where the fluids are used.

| Fluid name | ODP | GWP | Health code* | Flammability code* | Reactivity code* |
|---|---|---|---|---|---|
| **R-134a** | 0 | 1430 | 1 | 0 | 1 |
| **R-245fa** | 0 | 1030 | 2 | 0 | 0 |
| **MM** | 0 | n/a | 1 | 4 | 0 |
| **MDM** | 0 | n/a | 1 | 3 | 0 |
| **Toluene** | 0 | n/a | 2 | 3 | - |
| **n-pentane** | 0 | 20 | 1 | 4 | 0 |

* These codes are referred to US National Fire Protection Association (NFPA 704)

Table 3.3 Environmental and hazard aspects of the most common ORC fluids.

## 3.3 Simulation model developed in EES

A stationary simulation model of the ORC system has been developed in order to analyse the cycle performance and the main system parameters. This analysis allows a comparison of multiple system configurations, cycle parameters and different working fluids. Moreover, one of the aims in developing this simulation tool is the possibility of applying it in different situations. The same simulation tool has to be suitable both for an optimization analysis during the design phase and an operation analysis, where some cycle parameters are available from real measurements (e.g. cycle temperatures, output power, operating pressures, etc.) and the others need to be calculated through thermodynamic properties and component features. For this reason, the model should have the flexibility of providing performance results with different available inputs. However, the available operation data are often not enough, and therefore they need to be integrated with design parameters or literature values.

The model has been developed with EES (Engineering Equation Solver) software. EES is a general equation-solving program developed by F-Chart Software [33]. A major feature of EES is the high accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability. This aspect has been used to extend the performance analysis on the working fluids described in the previous paragraph. The model can be applied to any working fluid; this selection has been done for a better comparison with current commercial systems rather than an evaluation of all the possible working fluids.

Considering the purpose of the study some simplifications have been necessary. The aim of the tool is not a precise simulation of a given system with particular design conditions, but rather a parametric analysis of the system performance. For this reason the model performs a stationary simulation of each component of the system, without taking into account dynamic behaviour. Since the tool is intended to be used for an assessment of general performance, pressure drops and heat losses in the exchangers have not been considered (in accordance with [34], [35] and [36]). This approximation appears to be adequate for the purposes of this work.

The thermodynamic analysis has been developed considering a simple cycle with regenerator. The operation points of the cycle are reported in Figure 3.3. The working fluid exits the evaporator and reaches the turbine inlet (point 1), and after the expansion it enters the regenerator in the hot side (2). After the regenerator, the fluid enters the condenser as superheated vapour (3), and inside the condenser, it reaches the conditions of saturated vapour (4) and saturated liquid at the exit of the condenser (5). The working fluid is then pumped through the cold side of the regenerator (6), and after being pre-heated, it enters the evaporator as sub-cooled liquid (7) to be transformed in saturated liquid (8), saturated vapour (9) and slightly superheated vapour that exits the evaporator and enters the turbine (1). The needing of superheating the vapour up to point 1 is due to the necessity of avoiding liquid particles in the turbine, which would cause possible damages to the blades and a decrease of the efficiency.

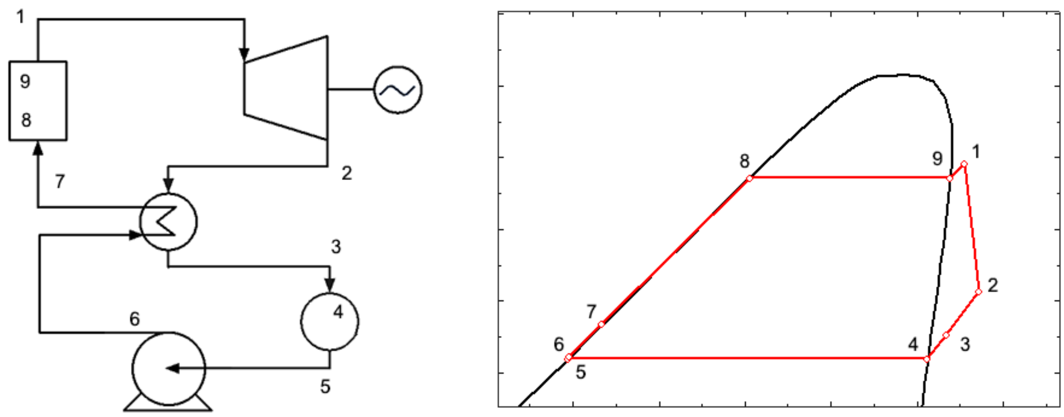


Figure 3.3 Operation points of the simulation system.

Each component has been defined through its thermodynamic properties, and energy balances for each component and for the whole system have been used for the calculation of the parameters of the cycle. The following paragraphs report the main equations that have been used for the simulation.

<u>Turbine</u>

The expansion of the working fluid in the turbine is represented by the transformation 1-2 in Figure 3.3. The conditions of the fluid at the inlet of the turbine are usually known from cycle design conditions, as they are dependent on the available primary heat source and on the choice of the working fluid. As pressure and temperature are known, it is possible to calculate enthalpy and entropy in the given point. The outlet enthalpy can be defined considering the isentropic efficiency of the turbine $\eta_{t,is}$ defined as follows:

$$\eta_{t,is} = \left(\frac{h_1 - h_2}{h_1 - h_{2is}}\right) \tag{3.1}$$

The isentropic efficiency depends on the turbine design, in nominal conditions it's typically in the range from 80 % to less than 90 %. In off-design operation the turbine efficiency is usually lower, as the best performances are achieved when the fluid expands in the turbine in nominal conditions.

The outlet pressure is related to the condensing temperature of the cycle, which depends on condensing parameters (e.g. external temperature for power systems and DH temperatures for CHP systems). The electric power generated by the turbine can be defined by the following equation:

$$P_e = \dot{m}_f (h_1 - h_2) \eta_{t,m} \eta_{t,e} \tag{3.2}$$

Where $\eta_{t,m}$ and $\eta_{t,e}$ are the mechanical efficiency of the turbine and the electrical efficiency of the power generator. An additional source of losses can result from the gearbox for high-speed turbines that need to be coupled to a 3,000 rpm generator. For this reason in some cases electronic systems are used instead of gearboxes, due to their higher conversion efficiencies.

Condenser

The condenser of the cycle can be connected to a cooling tower or it can be wet cooled (usually at relatively high temperatures if the condensing heat is used for thermal users). In the first case the temperature of the water is dependent on the ambient conditions. The nominal design conditions are the standard ISO conditions with 15°C and 60% of humidity, but depending on the site the outdoor conditions can be highly variable throughout the year. In the case of district heating network the water temperatures are usually set at supply design values of 75°C ÷ 90°C. These values can vary during operation, as a consequence of the heat load requested by the users.

A typical pinch point of 10°C has been set in the condenser, in order to calculate the pressure and the temperature of the fluid in the points 4 and 5 (see Figure 3.3). Before condensation, the working fluid that enters the condenser as superheated vapour (point 3) needs to be cooled down to saturated conditions.

The total heat to be exchanged in the condenser is given by:

$$Q_{cond} = \dot{m}_f (h_5 - h_3) \tag{3.3}$$

Pump

The calculation of the working fluid conditions at the outlet of the pump is based on the definition of the isentropic efficiency $\eta_{p,is}$. The output pressure is a design parameter of the system, while the enthalpy can be calculated from the equation of the isentropic efficiency of the pump:

$$\eta_{p,is} = \left( \frac{h_{6is} - h_5}{h_6 - h_5} \right) \tag{3.4}$$

The power consumption of the pump is calculated by defining the mechanical efficiency $\eta_{p,m}$ and the electric efficiency of the motor $\eta_{p,e}$:

$$P_p = \dot{m}_f(h_6 - h_5)\eta_{p,m}\eta_{p,e} \tag{3.5}$$

The pumping efficiencies can be significantly low, especially when pumps are not specifically designed for the cycle but they are adapted from other applications. The pumping efficiency can become in some cases a critical issue for the overall cycle performance, resulting in the need of a ad-hoc pump design.

<u>Evaporator</u>

The superheated water from the biomass boiler supplies the heat to the working fluid in the evaporator. Three different regions can be observed:

- the pre-heating region, where the liquid coming from the regenerator reaches the saturation conditions (points 7-8 in Figure 3.3) ;
- the evaporation region, where there is a coexistence of liquid phase and vapor phase (points 8-9);
- the super-heating region, where the saturated vapor is superheated before exiting the evaporator (points 9-1).

The pinch point in the evaporator is in correspondence of the point 8, and it has been set to 15°C. The heat losses and the pressure drops have not been considered.

The total heat to be supplied to the fluid in the evaporator is given by:

$$Q_{evap} = \dot{m}_f(h_1 - h_7) \tag{3.6}$$

The larger share of the heat is usually required by the evaporation, as the fluid is rarely much superheated in ORC systems.

<u>Regenerator</u>

The regenerator performs a pre-heating of the fluid before entering the evaporator, by recovering a part of the heat from the vapour at the turbine exit. This operation increases the efficiency of the cycle, by requiring less heat from the evaporator. The regenerator effectiveness, $\varepsilon_{reg}$, is defined as follows:

$$\varepsilon_{reg} = \left(\frac{T_7 - T_6}{T_2 - T_6}\right) \tag{3.7}$$

Through the assumption of $\varepsilon_{reg}$ it is possible to calculate the temperature of the working fluid at the inlet of the evaporator. The regenerator effectiveness is usually in the range 0.70 ÷ 0.75, depending on the size of the system, the working fluid and the operation temperatures. In paragraph 4.2.4 an effectiveness of 0.71 has been measured for a real ORC unit in a wide range of operation conditions.

The energy balance on the entire component is defined by:

$$h_2 - h_3 = h_7 - h_6 \tag{3.8}$$

This equation is valid under the assumption of no energy losses in the regenerator, which appears acceptable within the current approximations.

Depending on the characteristics of the fluid and the available primary heat source, in some cases an economizer can be installed in parallel with the regenerator. These systems are usually known as "split", and they allow to recover an additional part of heat when using high-temperature working fluids (e.g. siloxanes). This layout usually allows to lower the stack temperature of the flue gases, increasing the conversion efficiency of the boiler.

System performance

The ORC performance is usually calculated by mean of gross electric efficiency and net electric efficiency. The gross efficiency represents the ratio between the power produced by the generator and the heat provided at the evaporator:

$$\eta_{el,gross} = \frac{P_{el}}{Q_{evap}} \tag{3.9}$$

The net electric efficiency takes into account also the consumption of the pump and of the other auxiliary components of the system, and it can be defined as follows:

$$\eta_{el,net} = \frac{P_{el}-P_{aux}}{Q_{evap}} \tag{3.10}$$

In some ORC systems the pump requires a considerable power input if compared to the output power, reaching in some cases a share of 20%. Geothermal systems have higher auxiliary consumptions, due to the brine pump consumptions.

In the case of CHP systems also a thermal efficiency is defined, in order to consider the heat supplied from the condenser to the thermal users:

$$\eta_{th} = \frac{Q_{th}}{Q_{evap}} \tag{3.11}$$

The thermal efficiency should consider the real useful heat supplied to the user: if a part of the heat from the condenser is dissipated into the environment, this should be taken into account when assessing the thermal efficiency. In some plants only a part of the condensing heat is recovered, while in others systems the heat is dissipated during the summer season. In this case the average annual thermal efficiency becomes significantly lower.

The total ORC efficiency can be defined as the sum of electric efficiency and thermal efficiency:

$$\eta_{tot} = \eta_{el} + \eta_{th} \tag{3.12}$$

This value can reach in some cases 97%, demonstrating a good capacity of energy conversion in ORC systems.

It is important to remark that these efficiencies are describing only the ORC cycle performance, i.e. from the evaporator to the power output. When coupling ORC units to biomass systems the efficiency of the boiler becomes a crucial issue, being the component affecting the higher amount of energy losses (see paragraph 5.2.1).

Another significant index when considering a CHP unit is the heat-to-power ratio:

$$\lambda = \frac{Q_{th}}{P_{el}} \tag{3.13}$$

This index is generally used in the design phase, to compare the energy needs of the user and the energy that can be provided by the CHP unit. The heat-to-power ratio is highly dependent on the kind of CHP unit: ORC systems have generally high ratios (often > 4), whereas for gas engines the ratio is around 1. This index points out that ORC systems are usually best when the primary energy consumption is heat rather than electricity.

## 3.4 System performances and main parameters

The model described in 3.3 can be used for a parametric simulation of the system, to evaluate the main constraints affecting the ORC performance. The ORC electric efficiency is mainly related to three main aspects: the choice of the working fluid, the evaporation temperature and the condensation temperature. These three aspects are somewhat correlated, as the operation temperatures are determined by the primary heat source and the heat sink, but also by the characteristics of the working fluid.

The commercial units available show a wide range of electric efficiencies, depending on size, fluid, application, etc. The nominal gross efficiencies of some commercial units are reported in Figure 3.4 with respect to nominal output power.

The chart is based on an extract of available commercial units (data from technical papers, see [29], [30], [31] and [37]), in the range 50 $kW_{el}$ ÷ 3 $MW_{el}$. Larger units are available for some tailored geothermal applications, and therefore will not be considered in this study. The units are quite heterogeneous in terms of working fluids, applications, system layouts and reference temperatures. However, the value of the gross electric efficiency for CHP systems is in the range of 18% ÷ 20% for units larger than 300 $kW_{el}$ of output power. The full electric systems reach higher levels, up to 25%, thanks to the possibility of taking advantage from a lower condensation temperature. Smaller systems (< 300 $kW_{el}$) have efficiencies down to 7%, due to the lower evaporation temperatures and the size of components resulting in some technical constraints.

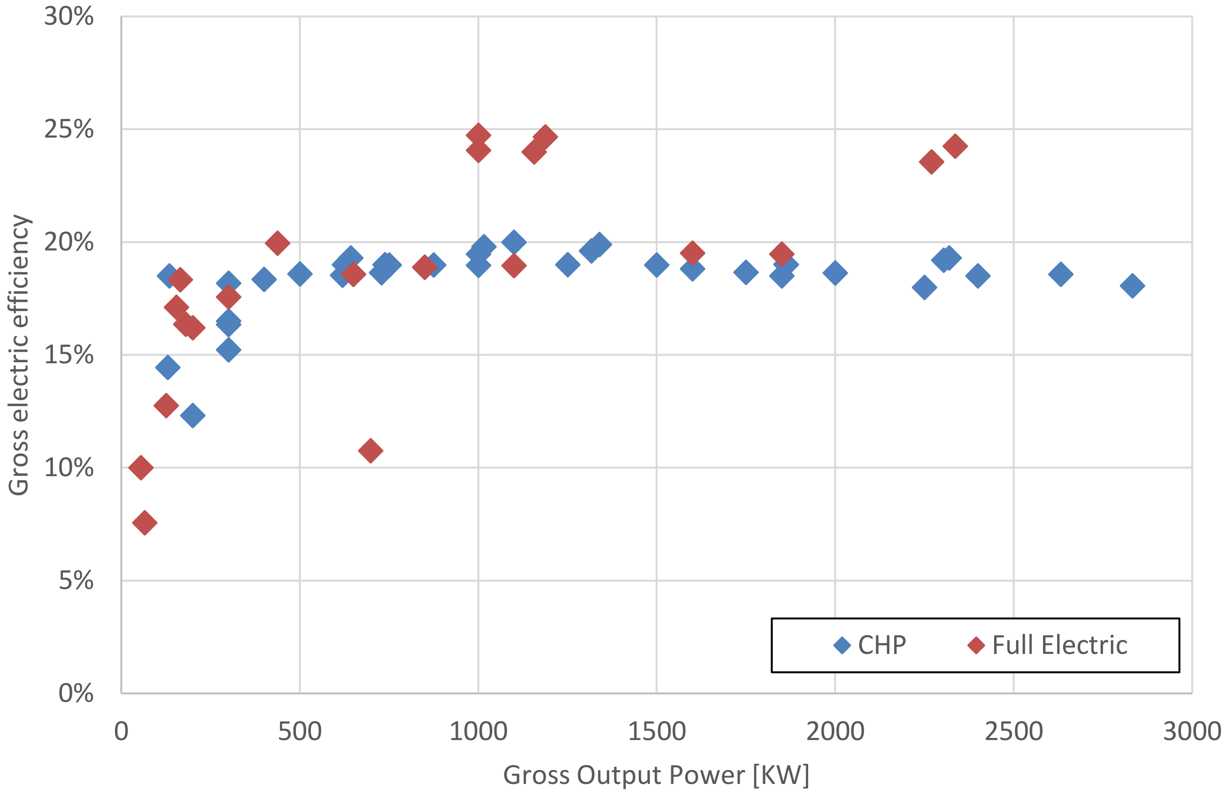


Figure 3.4 Gross electric efficiency over gross output power for some commercial units.

The simulation model has been used to evaluate the effect of the different parameters from a thermodynamic point of view. As discussed in 3.2, the working fluid choice is usually an iterative process that have to be performed for each application. The aim of this paragraph is to provide some general trends and considerations, as it is not possible to perform a general analysis that can be valid for each application. The model has been applied considering an ORC layout with regenerator (layout B of Figure 3.1), and considering a sub-critical cycle for the working fluids reported in Table 3.2. The R134a has been excluded, as with its critical temperature of around 100°C is not suitable for the temperature range of interest.

Figure 3.5 reports the results of the ORC simulations performed with different fluids, showing a comparison of the maximum gross electric efficiency that can be reached in an ORC unit. The efficiency increase associated with increasing turbine inlet temperature is evident, with some little differences between working fluids, depending on their characteristics and saturation curves. Real efficiencies are usually lower, due to the pressure drops and energy losses that have not been considered in the thermodynamic simulation. The temperature range of each curve is related to the typical application of each organic fluid. The results of Figure 3.5 have been obtained with an isentropic efficiency of 0.8 and a condensing temperature of 30°C. The toluene shows higher performances on a wider range. However, it has to be observed that MM and MDM plants in wood biomass applications are usually designed in "Split layout", obtaining higher conversion efficiencies.

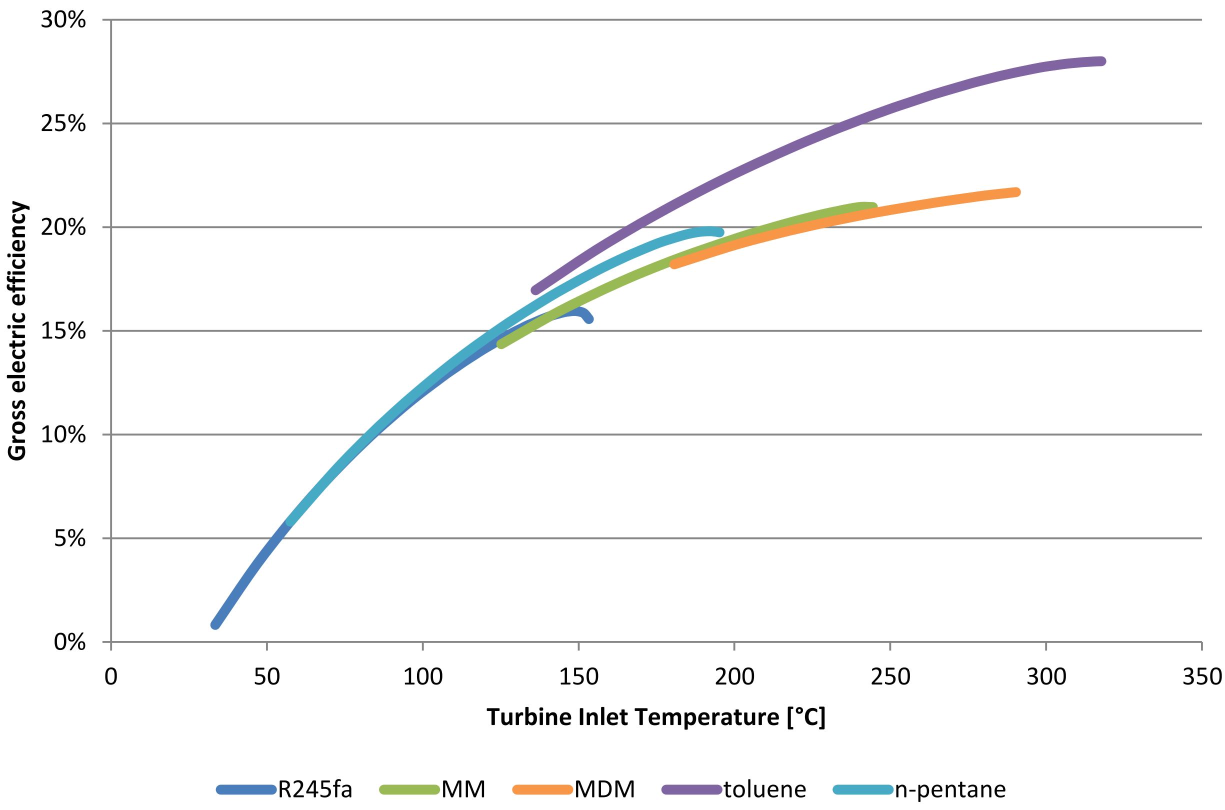


Figure 3.5 Simulation of ORC maximum electric efficiency w.r.t. turbine inlet temperature.

The evaporation temperature is usually fixed by the available heat source, and it is generally one of the most important design parameters for the choice of the working fluid and system layout. For this reason, the evaporation temperature should be kept as constant as possible during the operation of the system, although in some cases the primary heat source can have a variable temperature profile.

On the other hand, the condensation temperature is hardly constant during the operation of the system, and these variations can affect significantly the average performance of the ORC system. This issue can be related to multiple causes, depending on the type of condenser that is used.

Full electric ORC systems have lower condensing temperatures, the lower limit being generally associated with ambient temperature for air-cooled systems or water temperature for wet-cooled condensers. For this reason the electric efficiency varies throughout the year, and can be significantly depending on climate conditions. The availability of a water reservoir for condenser cooling gives generally interesting advantages, as it happens with steam cycles. However, even if the systems are generally smaller than steam cycles, potential environmental impacts on water need to be taken into account.

CHP systems use the available heat at the condenser to supply a heat user, which can be a district heating network, an industrial facility or another kind of user. The nominal temperature of heat supply is a design factor (usually in the range 60°C ÷ 90°C), but this value can vary over the year depending on the kind of load control of the system. A detailed analysis of the electric efficiency variation with respect to DH temperature is reported in [38].

The effect of the condensing temperature on the cycle efficiency has been investigated through some additional simulations. Figure 3.6 shows the results of the simulation, with a comparison between gross electric efficiency and condenser inlet temperature for two different fluids commonly used in biomass systems, MDM and R245fa. The evaporation temperature has been set accordingly to the maximum obtainable for each fluid, and it is the main cause for the huge difference between the performances of the two fluids. The condenser pinch point has been set to 10°C, whereas the working fluid entering the condenser has generally a little degree of superheating.

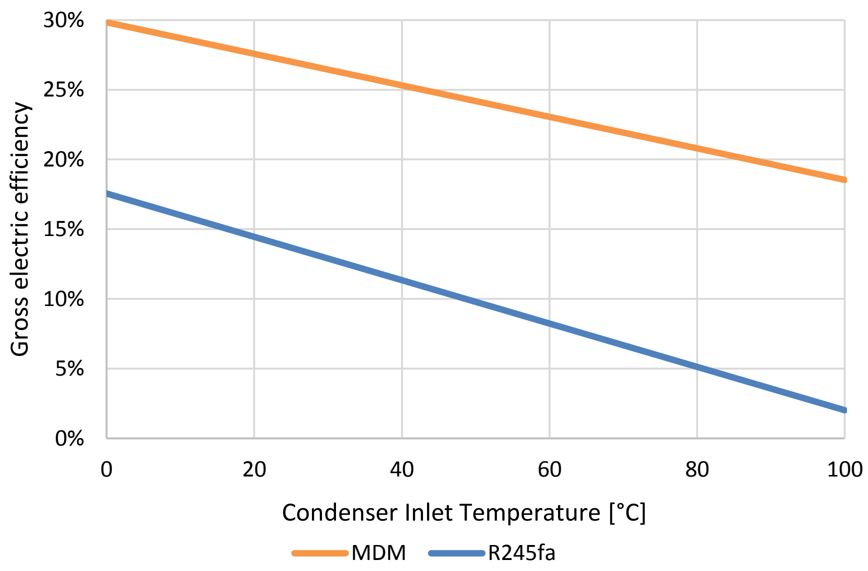


Figure 3.6 Relation between ORC electric efficiency and condenser temperature.

It is clear that the use of R245fa for CHP applications appears to provide a low electric efficiency, unless the heat is supplied at very low temperature (e.g. for low temperature space heating). The choice of MDM, which can benefit from the higher evaporation temperature, allows to obtain an suitable electric efficiency even with relatively high temperatures at the condenser. However, this analysis is a simplified thermodynamic evaluation of the performance: other aspects need to be taken into account when designing a real system, such as the minimum available size, the need of an additional thermal loop, etc.

Another aspect to be considered is the consumption of the feeding pump, which can in some cases become a significant issue in ORC systems. Figure 3.7 shows a simulation of the auxiliary consumption of the pump, for the same cases of Figure 3.5. The pumping consumption has been calculated considering an isentropic efficiency of 0.7 and a pump efficiency of 0.9. The worst fluid appears to be R245fa, while toluene and MDM show the best performance. The pumping consumptions are generally the larger part of the energy required for the auxiliary system components.

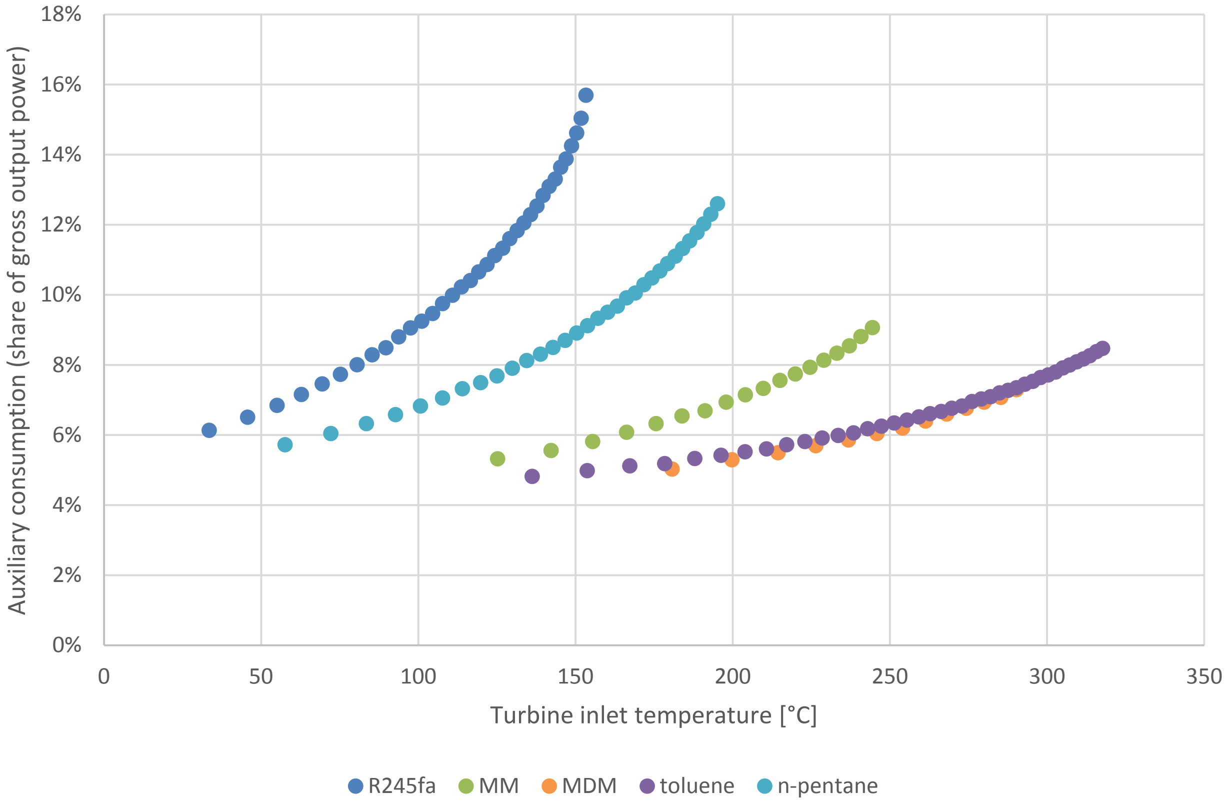


Figure 3.7 Simulated auxiliary consumption of the pump.

# Chapter 4 Operation analysis of real ORC systems

A part of the study has been devoted to the operation analysis of two real ORC systems currently in operation. This choice has been done in order to apply the simulation model to real systems, considering real operation data where multiple external factors can affect the performances and the operation conditions. The analysis of real plants allows to deal with multiple aspects that are usually not considered in standard simulations.

Two different case studies have been included in the analysis. A critical point has been the availability of operation data with a narrow time step, together with the plant owner's willingness of performing operation analyses. The case studies considered represent two typical applications of the ORC technology to biomass to energy conversion systems.

The first case study is composed by two small size ORC units (125 $kW_{el}$ each) connected to a superheated water boiler running on biomass residuals from pruning activities and green waste management. The type of ORC installed in this plant is one of the first installed in Italy in this range of power output for biomass systems.

The second case study is related to a more standardized ORC unit, with a nominal output power of about 1 $MW_{el}$. The ORC unit is part of a biomass CHP system installed in an industrial facility.

In the following paragraphs these two case studies will be described in detail, providing the main data available from the operation of the system, together with the results of the analysis.

## 4.1 Case study 1 – small size ORC unit

The first application considered for operation analysis is located in a wood residual facility, where the biomass residuals from pruning activity and green waste management are burned in a superheated water boiler. The energy conversion system, including 2 small size ORC units (125 $kW_{el}$ each), has been developed in the project Biogenera [39]. The research activity has been performed on more than two year of operation data of the ORC units (see also [40], [41] and [42]).

### 4.1.1 Description of the case study

The CHP system considered in this case is composed by a 2 $MW_{th}$ biomass boiler connected to two ORC units. A part of the heat production is used to supply a small district heating system that provides heat to some residential users and to the offices of the facility. The fuel is composed by wood biomass by-products, resulting from the activities of pruning and green management in the surrounding municipalities. As a consequence, the fuel is quite heterogeneous, and it is generally of low quality, with a variable heating value and a significant moisture content. Before the installation of the boiler, this biomass was not used for any energetic purpose and was sent to disposal.

The layout of the system is reported in Figure 4.1. The biomass boiler produces superheated water at 150 °C and 5 bar in nominal conditions, having a net heat output of 2,088 $kW_{th}$ with a nominal efficiency of 85.8%. The boiler has been specifically designed for the combustion of wood biomass residuals, with a moisture content higher than usual woodchips. The combustion chamber is slightly oversized, and it is covered with refractory bricks. The flue gases from the boiler pass through a multiple-cyclone and a fabric filter to remove particulate matter. The filters guarantee emission limits below 5 mg/$Nm^3$ for dust, 200 mg/$Nm^3$ for $NO_X$ and 80 mg/$Nm^3$ for CO (at 11% $O_2$).

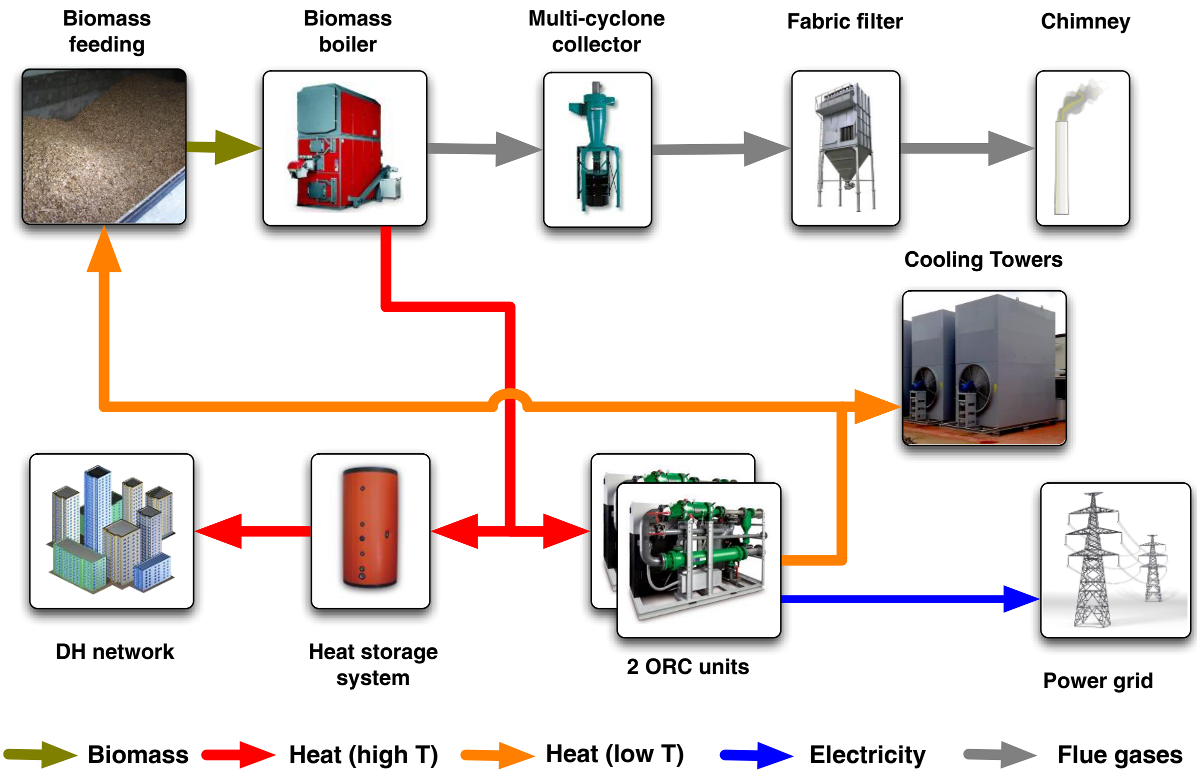


Figure 4.1 Biomass-fired CHP system layout.

The boiler supplies heat both to the district heating network and to the ORC units. The connection with the district heating network is made through a heat exchanger, providing heat to the grid with a supply temperature of 80°C and a return temperature of 60°C. Two heat storage tanks of 110 $m^3$ each allow to stock the excess heat produced by the boiler during off-peak hours.

The ORC units have a nominal output gross power of 125 kWel, with a nominal gross efficiency equal to 12.8%. The operating fluid is 1,1,1,3,3-Pentafluoropropane (HFC-R245fa). A patented integrated power module is composed by a high-speed turbine expander (26,500 rpm) plus a high-efficiency alternator in one sealed unit. The entire electricity production is directly supplied to the national grid, in order to maximize the economic benefits from the incentives.

Part of the heat recovered from the ORC condensing units is used to pre-heat the input biomass, in order to decrease its moisture content and to allow for better combustion conditions. The remaining part of the heat needs to be dissipated by two cooling towers.

Each ORC unit has a gross output power of 125 $kW_{el}$, requiring 980 $kW_{th}$ in the evaporator. The heat is supplied to the cycle through superheated water at 143 °C, while the organic fluid at the turbine inlet in nominal conditions is at 122 °C and 15.5 bar. The regenerator allows a pre-heating of the fluid to about 40 °C before entering the evaporator, increasing the cycle efficiency. The main design conditions of the unit are listed in Table 4.2.

| | **Unit** | **Value** |
|---|---|---|
| Gross output power | $kW_{el}$ | 125 |
| Gross ORC electric efficiency | - | 12.8% |
| Turbine inlet temperature | °C | 122 |
| Turbine inlet pressure | bar | 15.5 |
| Evaporator thermal power | $kW_{th}$ | 980 |
| Water inlet temperature in the evaporator | °C | 143 |
| Water outlet temperature in the evaporator | °C | 127 |
| Condenser thermal power | $kW_{th}$ | 820 |
| Organic fluid inlet temperature in the condenser | °C | 36 |
| Organic fluid outlet temperature in the condenser | °C | 21 |

Table 4.1 Nominal conditions of the ORC unit

### 4.1.2 Operation data analysis

The ORC units are equipped with a monitoring system, which retains in a database the hourly average value for multiple parameters. The available parameters are shown in Figure 4.2. Data are available for each unit over two complete years of operation, 2011 and 2012, while in 2013 some data have been lost due to database errors.

The water temperature is measured both in the evaporator and in the condenser inlet and output, while water mass flow is monitored only in the condenser. The working fluid properties (temperature and pressure) are measured only at the inlet and outlet of the condenser. The gross output power produced by the turbine, the external temperature and the humidity are available as well. These data are measured each second, but only hourly average data are stored in the ORC databases. Consequently, in some cases hourly averages can be the result of very different operation conditions and not be representative of the real behaviour. This aspect needs to be taken into account while using the data for performance calculations: some points can describe unreal operation conditions, and they need to be excluded from the sample. For this reason, the operation conditions lower than 50% of the electric load have not been considered in the analysis.

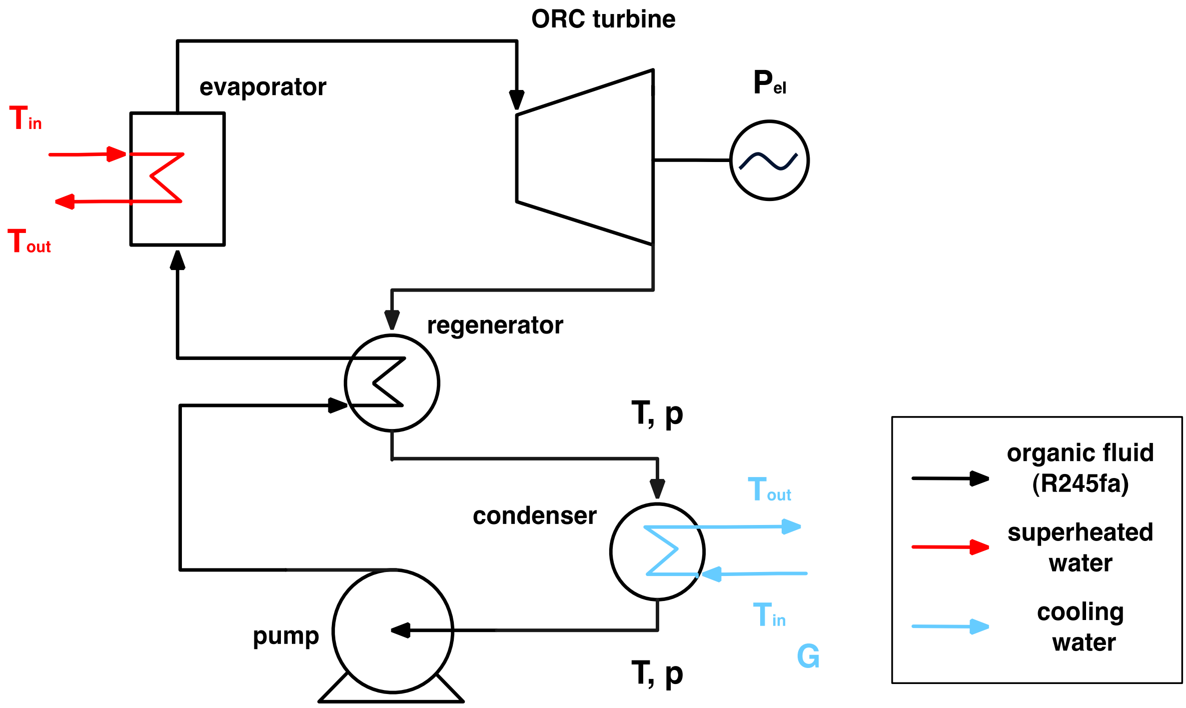


Figure 4.2 Available operation data.

### 4.1.3 Simulation of the system

The cycle has been analysed from a thermodynamic point of view, using the simulation model described in the paragraph 3.3.

Some approximations have been necessary, due to the lack of detailed design data. The thermal losses and pressure drops in the heat exchangers have been ignored. The heat exchangers have been studied considering a pinch point of 15°C for the evaporator and 10°C for the condenser, without calculating exchanging surface and exchanging coefficients. Therefore, only design conditions can be simulated with these assumptions. A detailed analysis of the off-design operation is not possible to be performed with the available design and operation data. However, the degree of approximation appears acceptable given the context of this study.

The main simulation assumptions related to the nominal performance of the components are listed in Table 4.2. The values of the efficiencies have been chosen considering literature values for similar units, as detailed information on the ORC system has not been provided.

Figure 4.3 shows the nominal conditions of the ORC cycle. The red line represents the R-245fa, while the blue lines show the water conditions in the evaporator and in the condenser. The specific entropy axe is showing fluid entropies, while water has been scaled in order to match the organic fluid conditions. This chart has been obtained by running the simulation in nominal conditions, as not all the points were known from design data.

| Parameter | | **Value** |
|---|---|---|
| Turbine isentropic efficiency | $\eta_{t,is}$ | 0.75 |
| Turbine mechanical efficiency | $\eta_{t,m}$ | 0.95 |
| Electric generator efficiency | $\eta_{t,e}$ | 0.97 |
| Pump isentropic efficiency | $\eta_{p,is}$ | 0.60 |
| Pump electric motor efficiency | $\eta_{p,e}$ | 0.98 |
| Pump mechanical efficiency | $\eta_{p,m}$ | 0.80 |
| Regenerator effectiveness | $\varepsilon_{reg}$ | 0.50 |

Table 4.2 Main simulation assumptions

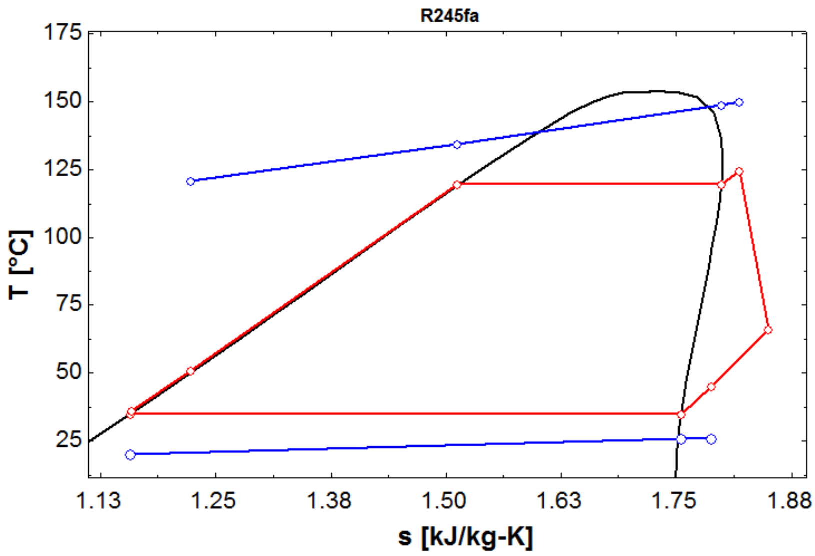


Figure 4.3 Nominal ORC conditions.

### 4.1.4 Operation data results

The operation data of the two ORC units has been analysed from January 2011 to March 2013, considering hourly averages stored from the monitoring system of the turbines.

Figure 4.4 shows the gross output power from each units over the operation analysis. The two main aspects are the high variability of the power with respect to nominal conditions (125 $kW_{el}$), and the good availability of the system throughout the years (> 7,500 h/y). The annual energy production ranges from 745 $MWh_{el}$ to 815 $MWh_{el}$, the ORC 2 being slightly better. The average output load is between 98 $kW_{el}$ and 105 $kW_{el}$.

The operation at partial load appears to be rather a consequence of the evaporator temperatures than a control choice of the plant operator. As can be seen in Figure 4.5 the evaporator inlet temperature is often lower than the nominal value of 143°C, resulting in off-design conditions of the evaporator and the other components of the cycle. The same plot shows that also the return temperature is significantly lower than the design condition (127°C).

The limitation of the output power by the evaporator inlet temperature is well represented by the correlation plot of Figure 4.6: there is a noticeable upper limit in the cloud of points. The scattering of the points is due to other parameters that can affect the output power, as the heat supplied in the evaporator, the external temperature, the condensing temperature, etc.

The most probable cause is related to the lower performance of the biomass boiler, due to the high moisture content of the input biomass. This has been noticed by the plant owner, which also reported some problems with the heat exchangers fouling. However, operation data of the boiler are not available, and this assumption cannot be demonstrated with performance measurements.

Another effect of the off-design operation lays in the inlet temperature to the condenser, which is higher than expected (see Figure 4.7). The design value has been set to 36°C, but the actual operating point lays between 40°C and 60°C. This means that a significant part of the heat available in the fluid is not recovered by the regenerator, and is being dissipated at the condenser. This is probably related to the inability of the regenerator to recover all the available energy at the turbine outlet. Unfortunately, also in this case the lack of detailed design information prevent to confirm this hypothesis with more accurate calculations.

The electric gross efficiency of the ORC unit has been calculated using the simulation model. The results are represented in Figure 4.8, showing the relation between electric efficiency and load of the ORC units. The values are lower than the nominal efficiency of 12.8%, and they are significantly decreasing with the load. Moreover, there is a noticeable variability in the cloud. In this case ORC 2 appears to have generally lower performances than ORC 1.

Once again, the low performance at partial load is also related to the evaporator temperature: as discussed before the output power lower than expected is a consequence of evaporator conditions rather than a system control. A global efficiency of the system cannot be calculated, as there are no reliable information on the operation of the biomass boiler and the biomass has a high variability. Dedicated measures of calorific values, together with real biomass flow rate, would be needed in order to calculate the global efficiency of the system.

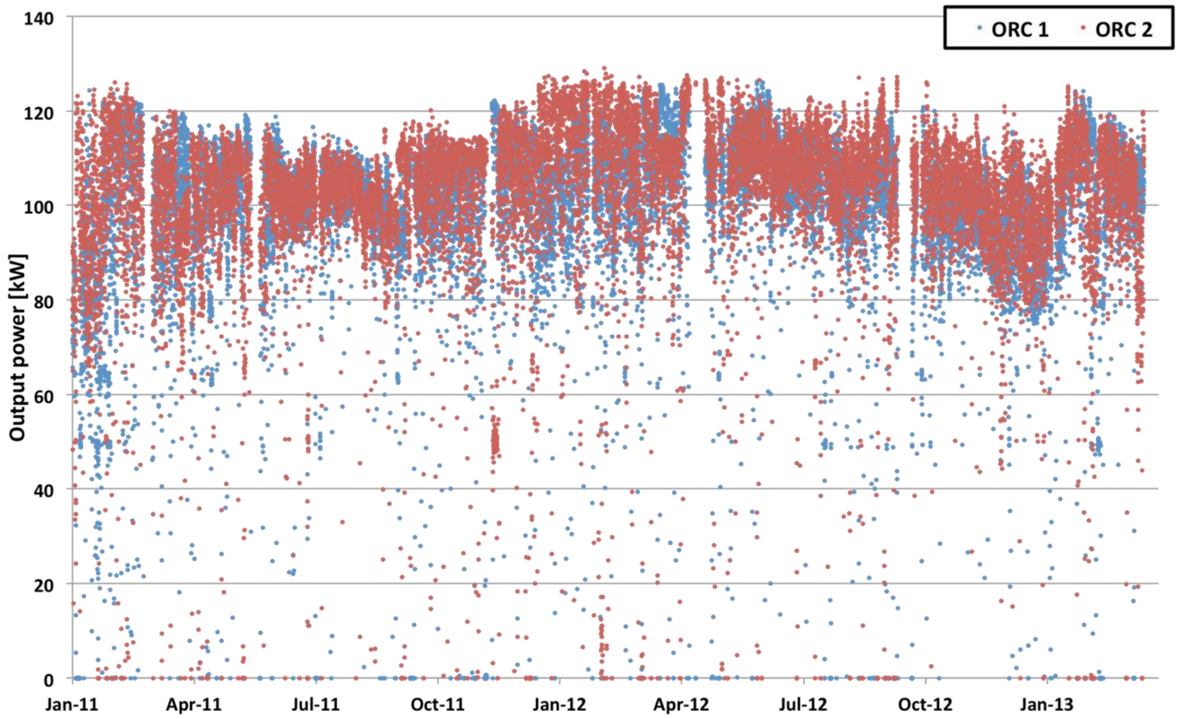


Figure 4.4 Hourly output power from ORC units.

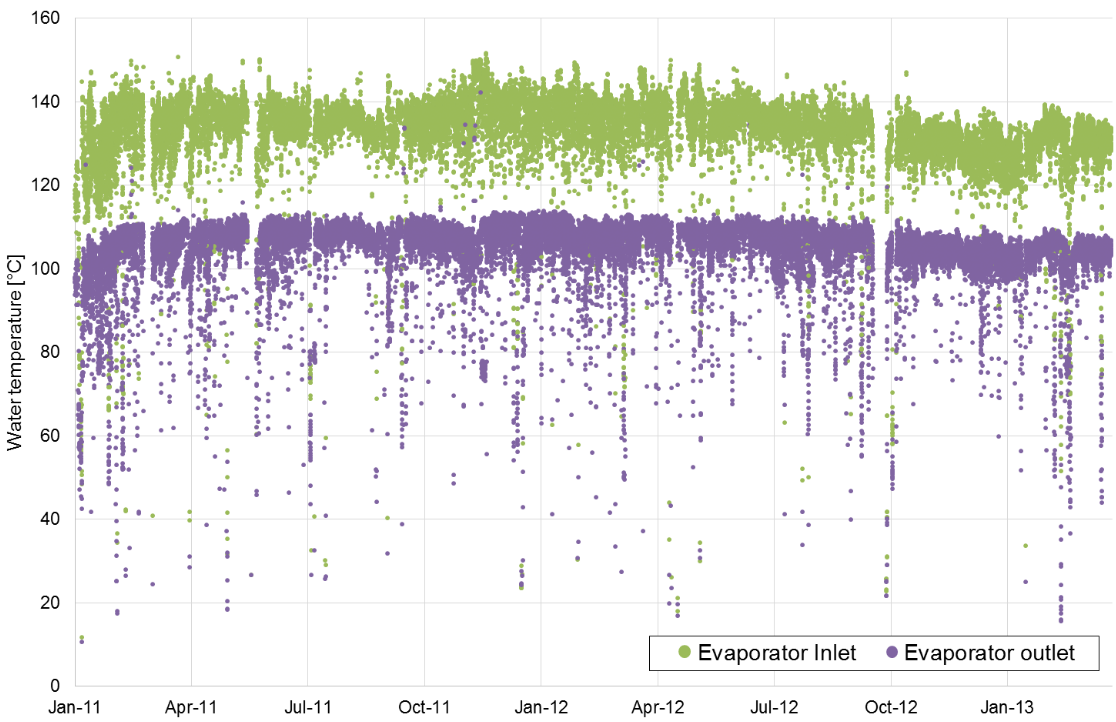


Figure 4.5 Evaporator inlet and outlet temperatures (ORC 1).

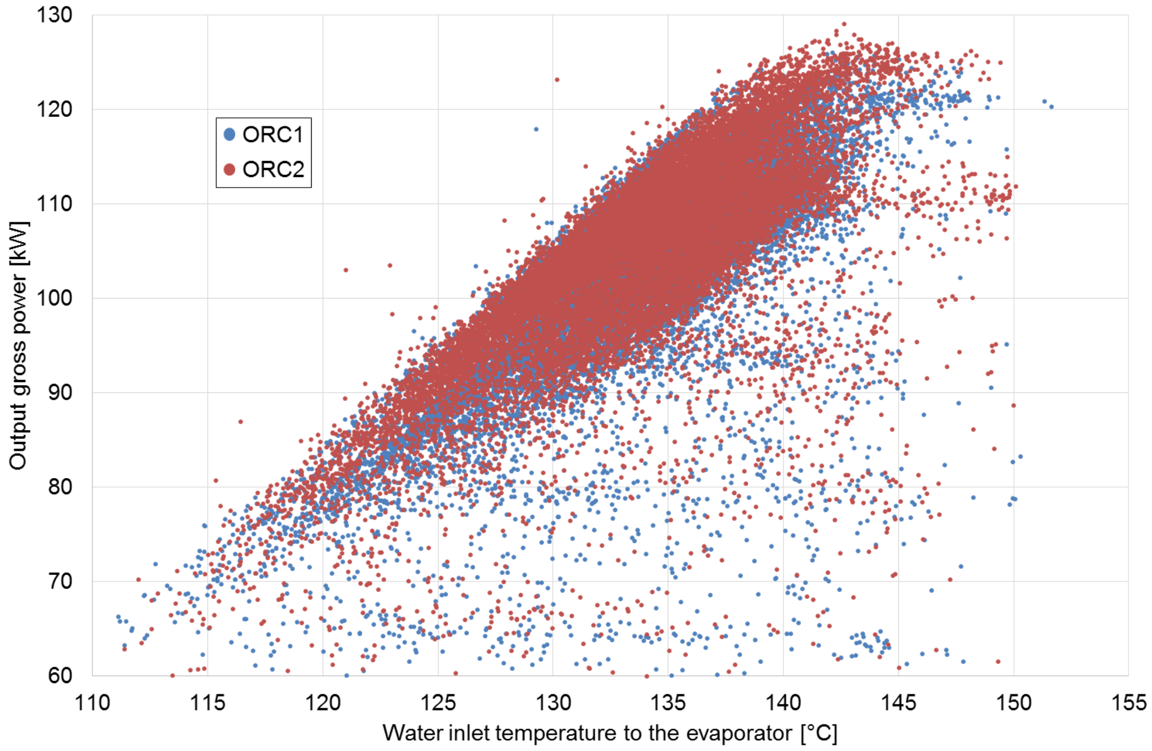


Figure 4.6 Output gross power vs water inlet temperature to the evaporator.

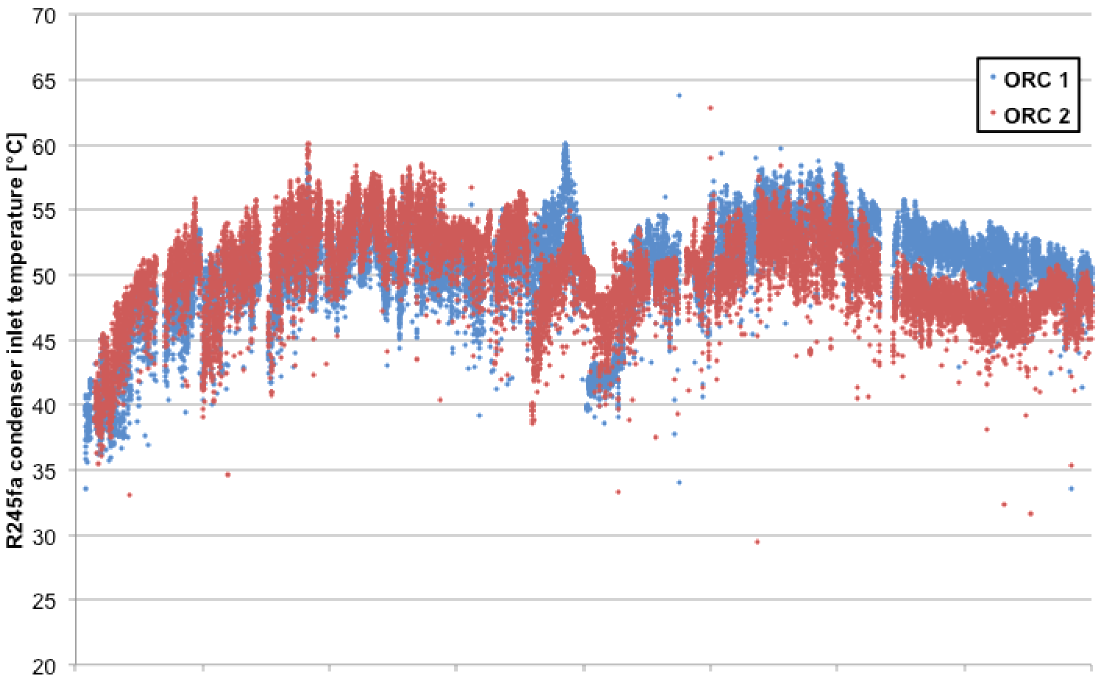


Figure 4.7 Condenser inlet temperature.

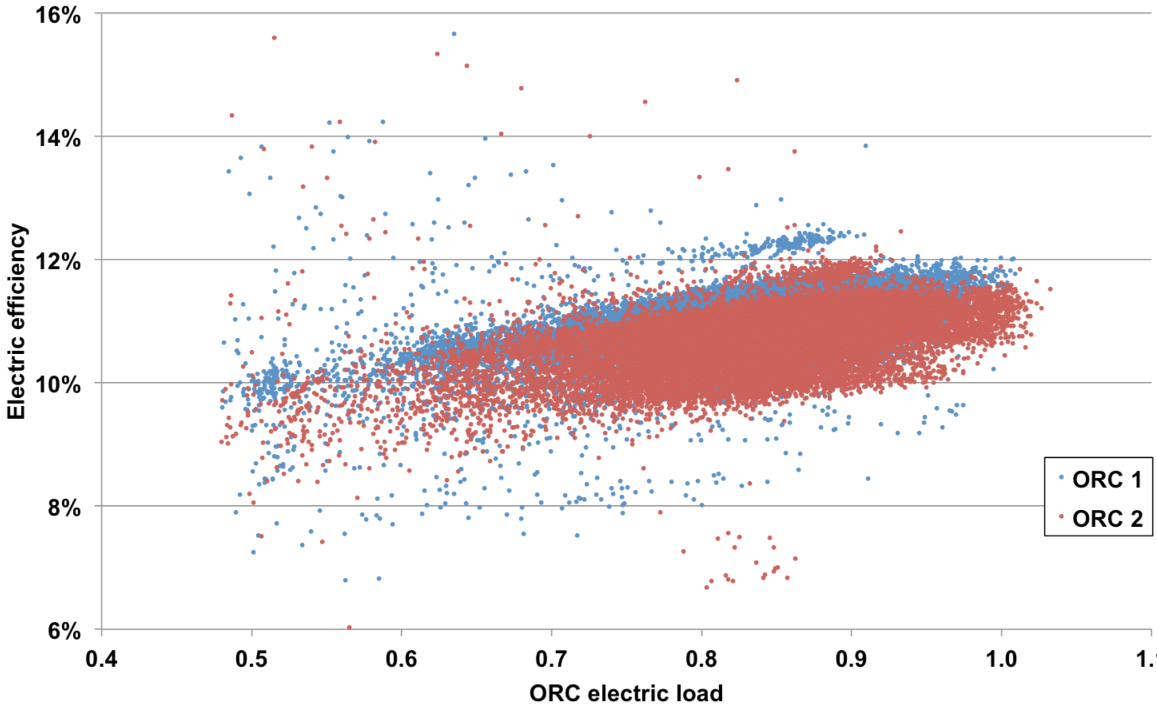


Figure 4.8 ORC gross electric efficiency over electric load

## 4.2 Case study 2 – medium size ORC unit

Due to a non-disclosure agreement with the plant owner, the location of the system and other sensible information will not be published in this work. Therefore, a less detailed description of the system will be provided.

### 4.2.1 Description of the case study

The ORC analysed in this case study is representative of the standard size developed for biomass applications. The gross output power is about 1 $MW_{el}$, with an electric efficiency of 20%. The fluid used in the ORC is the hexamethyldisiloxane (also called MM or HMDSO), an organosilicon compound with the formula $O[Si(CH_3)_3]_2$. It is a colourless liquid with a molar mass of 162.38 g/mol, a density of 0.764 g/cm$^3$. It is however a highly flammable liquid and vapour, which can cause serious eye irritation, it is normally stable and non-reactive with water.

The ORC cycle is a subcritical regenerated cycle, operating with a high pressure of about 9 bar and a low pressure of about 0.1 bar. The heat is supplied to the ORC unit by a silicon oil circuit, which uses Therminol 66 and operates between 290°C in the supply side and 145°C in the return side. This circuit, at a pressure of about 5 bar, supplies also an internal heat network: therefore in some cases the ORC unit is working at partial load. The condenser is connected to a water loop cooled by an evaporative tower of a nominal capacity of 5.4 $MW_{th}$ with operating temperatures in design conditions of 38°C / 26°C and a water flow rate of 92.6 kg/s.

The main ORC characteristics are reported in Table 4.3.

| Parameter | | **Value** |
|---|---|---|
| Organic Fluid | | MM |
| Gross output power | $MW_{el}$ | 1.11 |
| Net output power | $MW_{el}$ | 1.06 |
| Gross electric efficiency | - | 20% |
| Net electric efficiency | - | 19% |
| Total heat input (evaporator) | $MW_{th}$ | 5.54 |
| Evaporator inlet temperature | °C | 290 |
| Total heat output (condenser) | $MW_{th}$ | 4.40 |
| Condenser temperatures (in/out) | °C | 26 / 38 |

Table 4.3 ORC nominal characteristics

The main difference with respect to other ORC units is the installation in parallel with the heat network, because of the need of high temperature heat (which cannot be provided by the condenser of the ORC unit). This particular configuration is an interesting source for operation data in unusual conditions, as partial load is often avoided in biomass plants: ORC units are usually running at full load operation to maximize the electricity production and minimize O&M costs. This particular case study allows to analyse a wide range of electric load with actual data from a real unit in operation and not at laboratory scale.

### 4.2.2 Operation data analysis

The monitoring system of the ORC unit is recording a number of operation data, with a time step of 1 minute. The main data recorded by the monitoring system which are of interest for a thermodynamic analysis are the following:

- gross output power;
- captive power consumption for auxiliary systems;
- ORC fluid high and low pressure;
- ORC fluid temperatures at different points;
- thermal oil temperatures at inlet and outlet of the evaporator;
- water temperatures at inlet and outlet of the condenser.

A number of useful data are available from real measurements. However, for a complete definition of the operation of the system a simulation model is necessary in order to define the missing information (e.g. the mass flows of the different circuits). The available operation data have been analysed considering the hourly average values, in order to avoid transient conditions which cannot be correctly analysed through a stationary model.

The available data are recorded over more than one year of operation (from January 2013 to May 2014). However, the ORC unit has not been operated continuously. This choice is not related to the availability of the unit, but is rather caused by a more wide organization strategy of the whole energy production site. Therefore, the performance analysis has not considered annual parameters like energy production, operation hours, availability, etc.

### 4.2.3 Thermodynamic simulation of the system

In this case study the simulation of the system has the advantage of a deeper knowledge of the operation temperatures and pressures in all the points of the cycle. The same simulation model presented in the previous case has been used, but with less hypotheses for the thermodynamic calculation of the cycle operating points. The pressure losses and the energy losses in heat exchangers and pipes have been ignored.

The main simulation assumptions related to the efficiency of the components are listed in Table 4.2. The mechanical and electrical efficiencies of the turbine have been set through a recursive analysis of the electric gross efficiency of the ORC cycle, with respect to design conditions. The turbine isentropic efficiency doesn't need to be defined in the simulation, as the data available allow to define the real conditions of the fluid at the inlet and outlet of the turbine, without the need to consider the isentropic transformation.

| Parameter | | **Value** |
|---|---|---|
| Turbine mechanical efficiency | $\eta_{t,m}$ | 0.955 |
| Electric generator efficiency | $\eta_{t,e}$ | 0.95 |
| Pump isentropic efficiency | $\eta_{p,is}$ | 0.70 |
| Pump electric motor efficiency | $\eta_{p,e}$ | 0.98 |
| Pump mechanical efficiency | $\eta_{p,m}$ | 0.90 |

Table 4.4 Main simulation assumptions

The nominal operation conditions of the unit are shown in Figure 4.9. The orange lines represent the regenerator conditions.

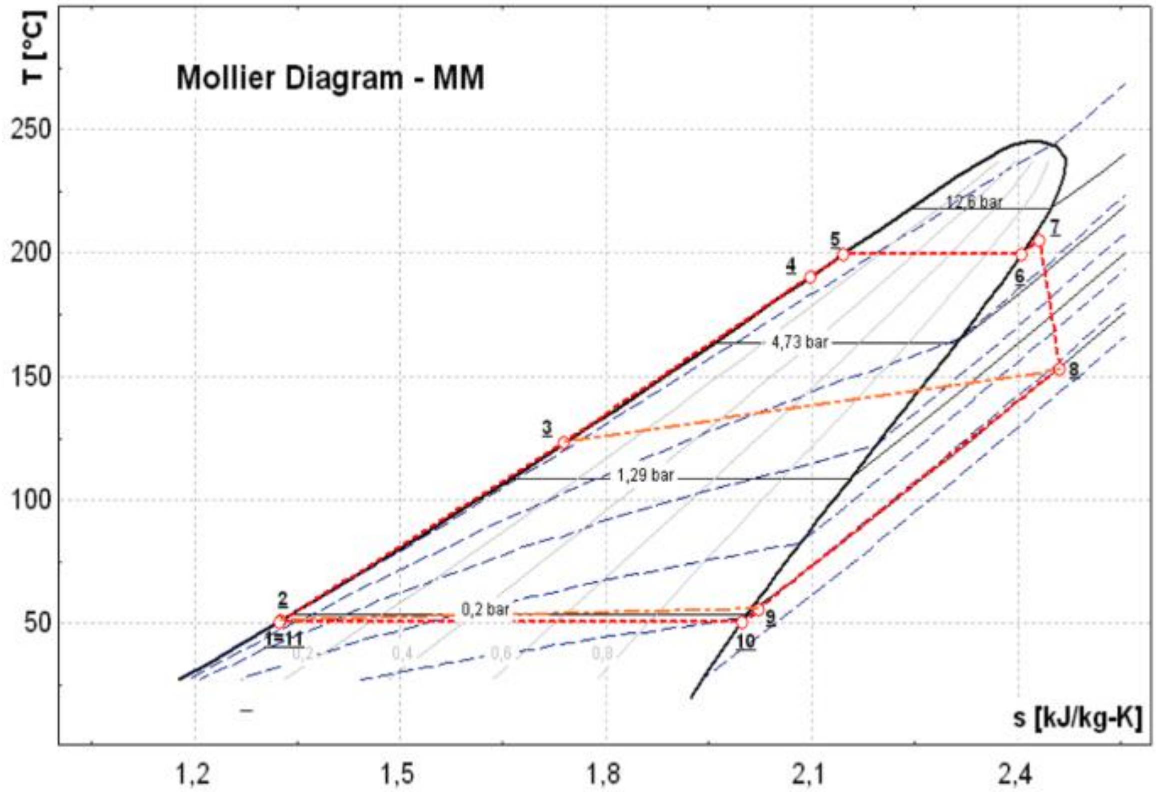


Figure 4.9 ORC simulated nominal conditions

Considering energy balances equations the mass flow rates of the MM, the thermal oil and the cooling water in the condenser have been calculated for each point of operation.

### 4.2.4 Operation data results

The available effective operation data of the unit covers about 3,000 hours over a time range of 12,000 hours. Figure 4.10 shows the gross output power throughout the monitoring activity. The unit mainly worked near nominal power (and in some cases also at slightly higher power), but the partial load operation has been frequent. While in summer all the available heat from thermal oil has been used for the ORC unit, in winter and middle seasons the heat is also required for space heating purposes. This aspect is particularly evident in the chart around the 8,000 h value. The unit has been operated down to 10% of the nominal power for several hours of operation.

Figure 4.11 reports the variation of the power consumption of the auxiliary systems with respect to the gross output power from the turbine. The larger share of auxiliary systems consumption is related to the pump, which causes the parabolic trend that can be noticed in the plot. For nominal output power, the total consumption of the auxiliary systems is in the range from 35 to 48 $kW_{el}$. These values are in accordance to the rated consumption of 46 $kW_{el}$ in design conditions, which is about 4% of the gross output power of the unit. The intercept with the y-axis, representing the amount of power required also at zero load, is about 4 $kW_{el}$.

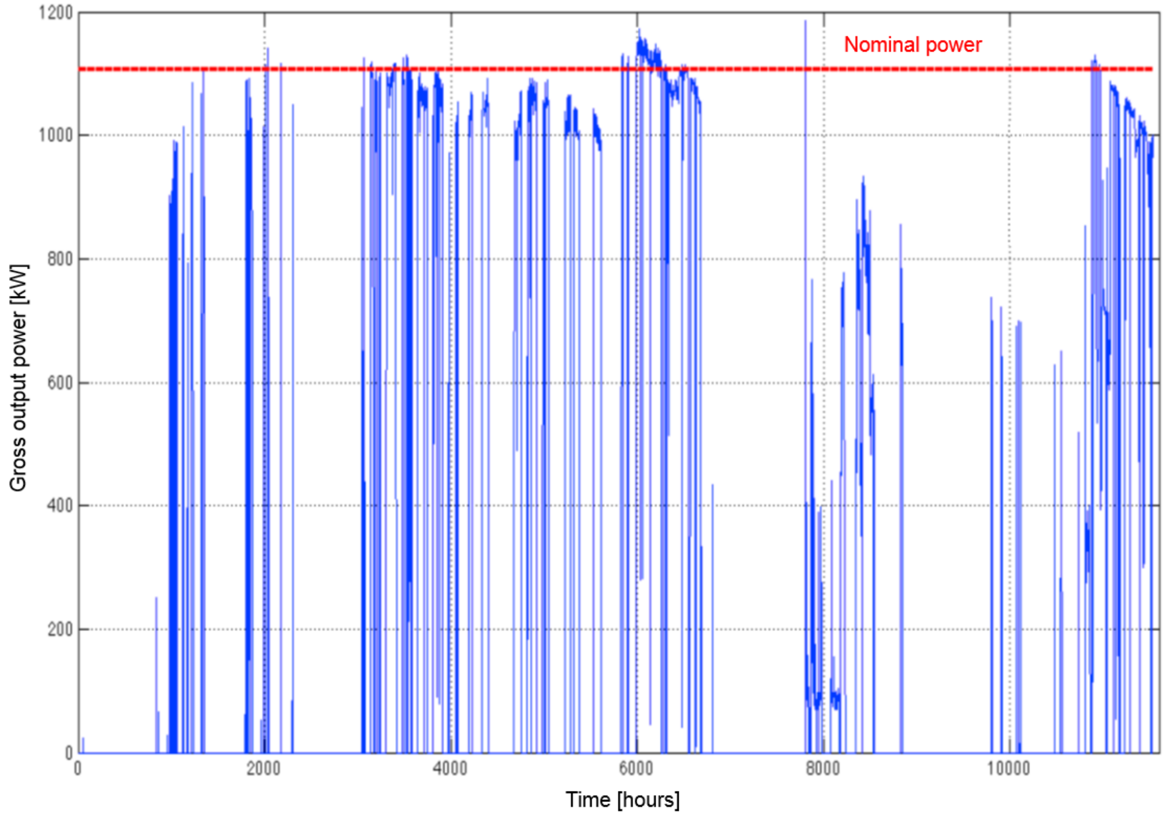


Figure 4.10 ORC gross output power during operation

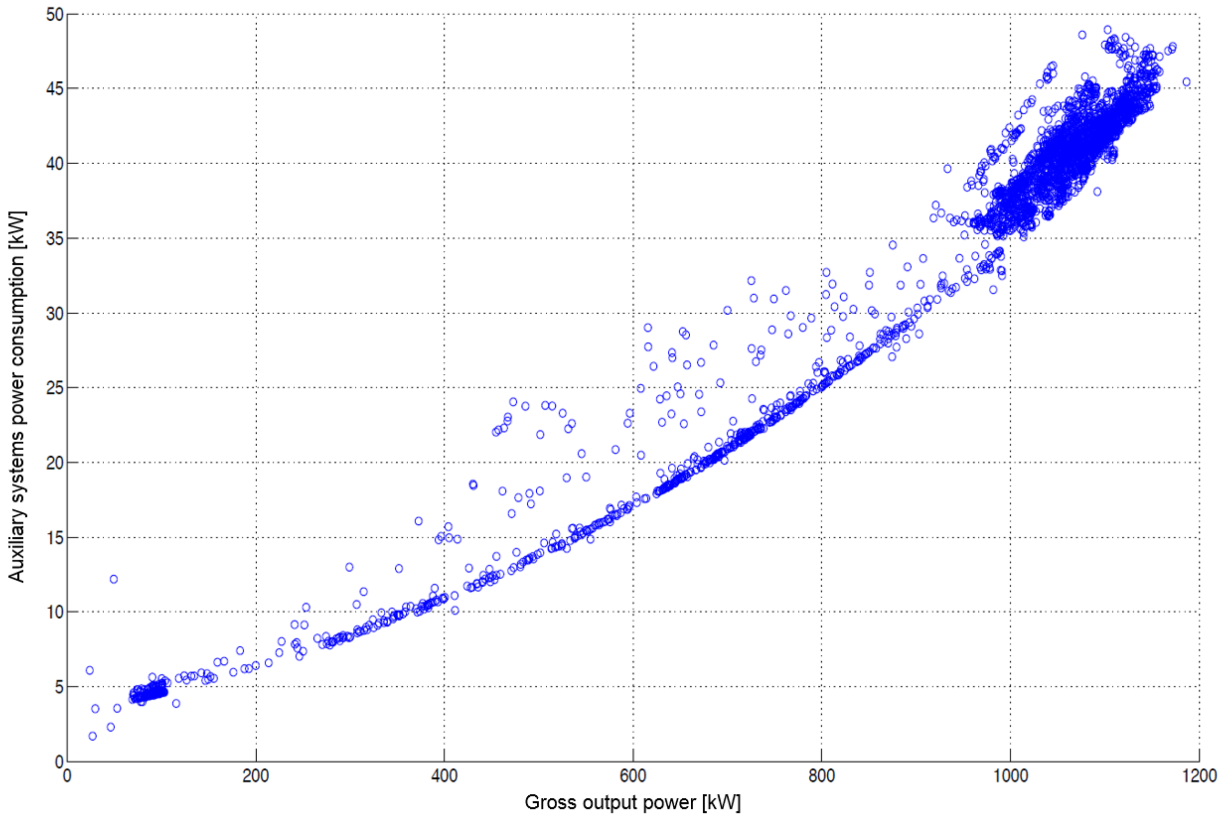


Figure 4.11 Auxiliary systems power consumption

The main result of the application of the simulation model is the possibility of calculating the electric efficiency of the cycle. Using the model described in the previous paragraph it has been possible to obtain the chart of Figure 4.12. The gross electric efficiency shows a good performance both at full load and at partial load, remaining near the nominal value down to about 30% of the power load (ratio between the output power and the nominal power).

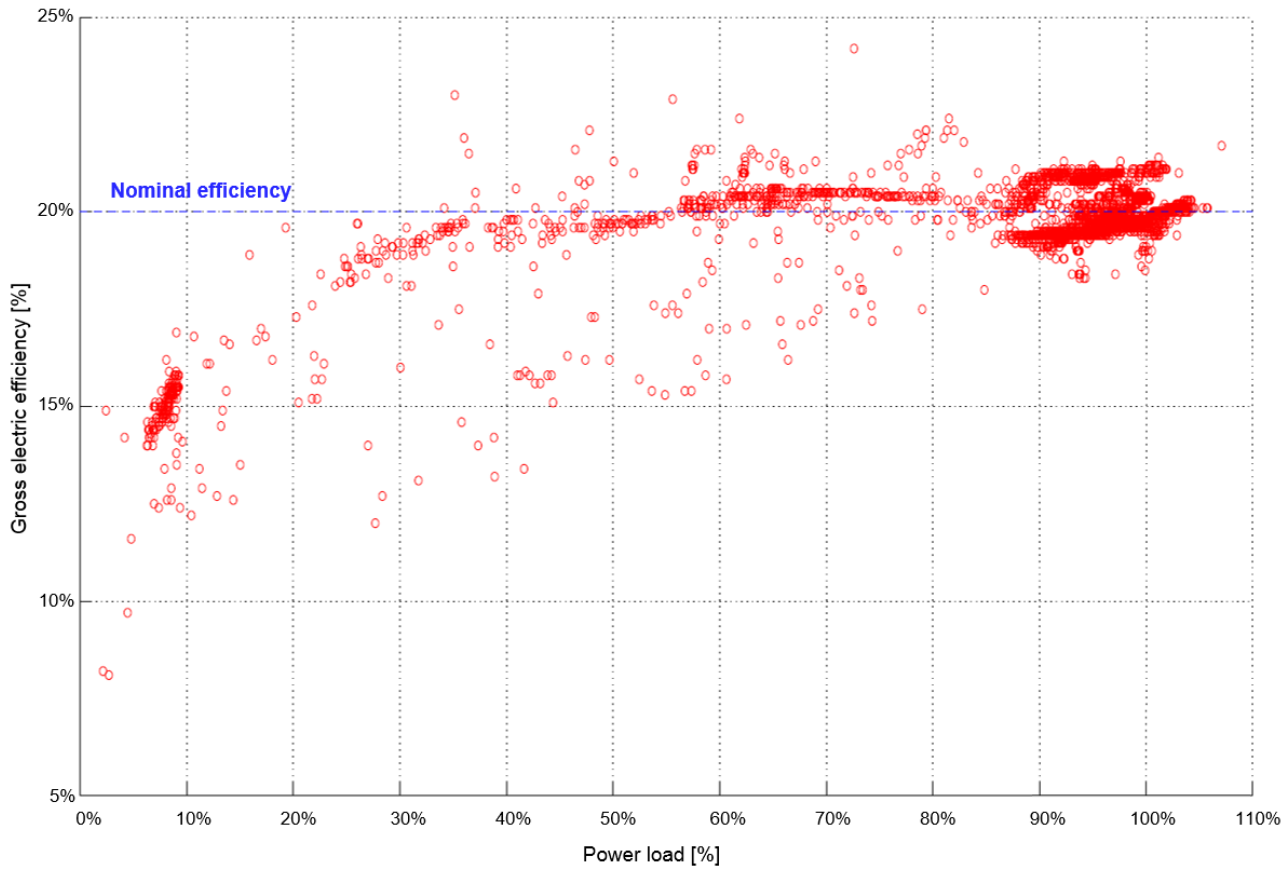


Figure 4.12 ORC gross electric efficiency

There is a significant amount of operation points with power load lower than 10%, with a gross efficiency of about 15%, which correspond to 75% of the efficiency at full load. This aspect is of interest as is not a common operation strategy in biomass systems. This analysis confirms that ORC units can provide a good performance also with variable loads. The net efficiency shows a similar behaviour, being the gap between gross and net efficiency almost always constant.

The scattering of the points is related to other parameters than power load, which can affect the electric efficiency (e.g. the condenser temperatures or the evaporator temperatures). A significant variability can be found in the range from 90% to 100% of the power load, where the majority of operation points lays, with about 20 ± 1% of gross efficiency. The two main clouds are related to the season of operation, which affects the condensing temperatures. With cold outdoor temperatures the condensing towers can provide a better cooling in the cycle, increasing the electric efficiency.

The availability of the measurement of real temperatures allowed to verify the actual effectiveness of the regenerator, which is usually a value set from hypotheses. The results of the calculation are reported in Figure 4.13. The dependence from power load seems very weak, being the value almost always near 71%. It appears to be slightly decreasing near 100% of power load, probably because of the saturation of the heat exchanger capacity.

Finally, the relation between the condensing pressure and the water temperature at the condenser outlet is shown in Figure 4.14. This behaviour is in line with the design conditions and the physical relations between condensing temperature and pressure.

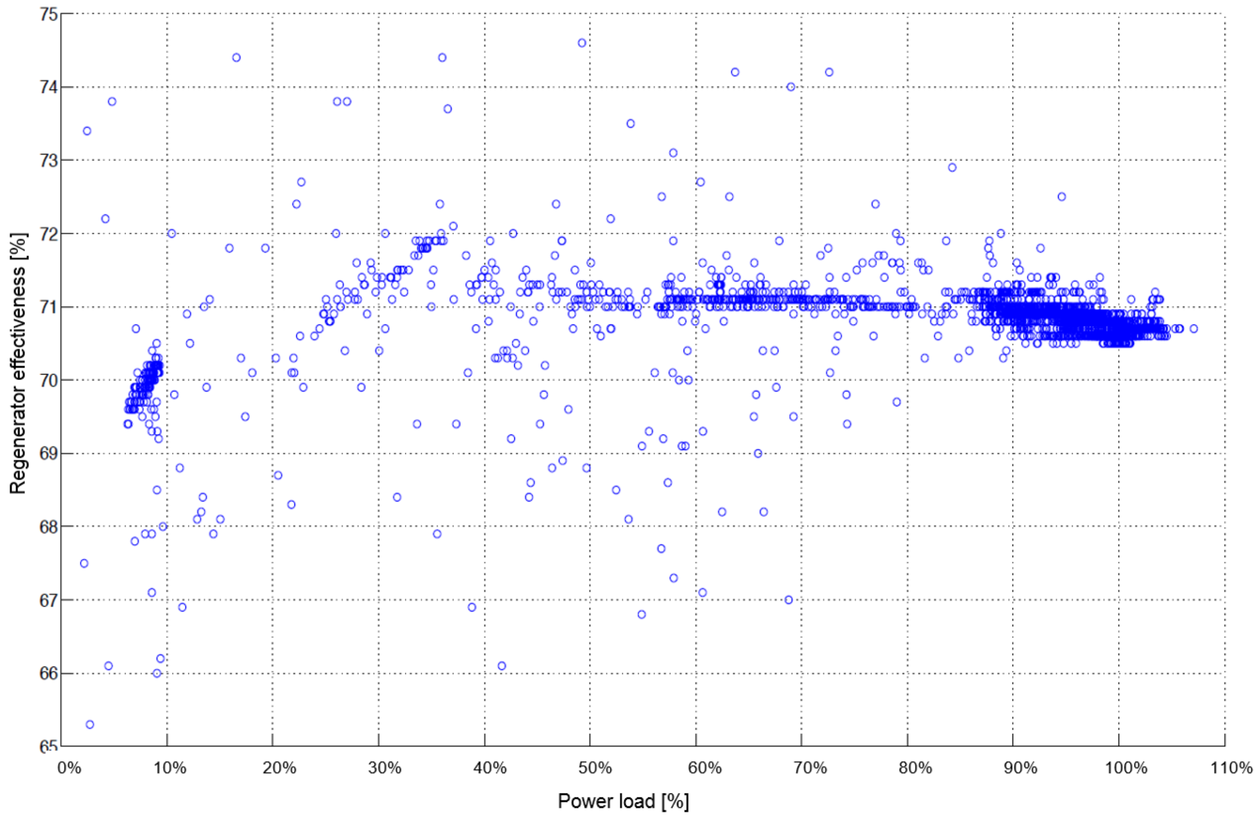


Figure 4.13 Regenerator effectiveness vs power load

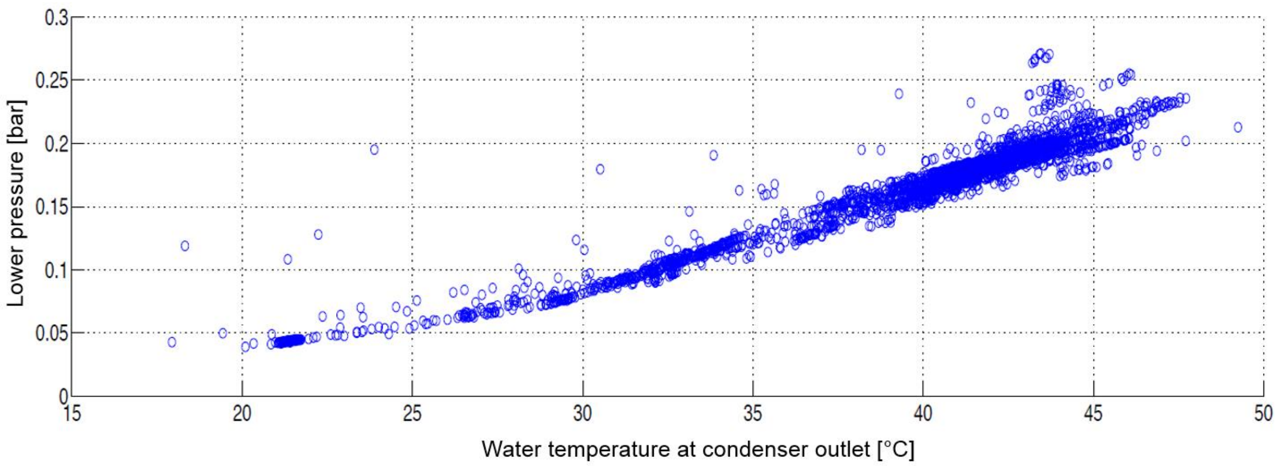


Figure 4.14 Relation between lower pressure and water temperature at condenser outlet

## 4.3 Common considerations and critical points

The two case studies described in 4.1 and 4.2 are representing two different applications of the same technology, with common aspects but also significant differences.

The main aspect to be considered is the difference level of technological maturity: while the first application represent an innovative unit for the industrial sector (for the size of the units and the possibility of recovering heat from very low quality wood biomass), the second case study is a mature technology with years of operation experiences and a number of installations all over the world. This significant difference needs to be considered when comparing the behaviour of the two energy systems.

However, the first system shows a good availability throughout the year, with 7,600 – 7,700 hours of operation per year. Considering the quality of the input biomass and the low maturity of this kind of ORC units this is an interesting performance. Some small differences have been found between the two ORC units, mainly on average output power and efficiency. In the second case study, the operation of the ORC unit is dependent on a wider operation strategy including other generation systems. Therefore, its annual availability is not a significant value, being limited by the heat source availability rather than maintenance time or unit failings.

Both systems show a low performance decrease at partial loads, the electric efficiency being quite constant at lower power loads, but with some significant differences. In the first case the electric efficiency in operation is lower than the design value, and a performance decrease at partial loads can be observed. The main cause is related to the water temperature in the evaporator, which decreases at partial loads. Specifically, the partial load of the system appears to be caused by the lower evaporation temperature, resulting in lower enthalpy available for the turbine. In the second case the electric efficiency is in line with the design values, and in some cases it is even higher. The partial load behaviour is quite constant, down to 30% of the power load. In this case the partial load operation is a choice related to the availability of heat source, which is supplied with a constant temperature but at different mass flow, as a part of the heat is requested by other parallel industrial processes.

However, the main criticality of these two case studies lies in the low efficiency in the conversion of heat into electricity, which reaches only 12% in the first case and 20% in the second case. This value represents the gross efficiency of the cycle, being further reduced by the heat production in the boiler. Both systems are cooled with outdoor air condensers, thus dissipating heat to the environment. In this configuration the overall energy efficiency appears really low, the main part of the energy being dissipated. This issue can be solved by an accurate planning of the units that allows coupling them to an existing heat demand in order to increase the overall efficiency of the system.

The ORC systems of medium size (output power of 1÷3 $MW_{el}$) supplied by wood biomass are usually operated in CHP, connecting the condenser to a district heating grid at 80-90°C. Prando et al. [38]

analysed the actual operation conditions of a biomass CHP system coupled to a small district heating network. The overall performance of the plant calculated over the year showed an electric efficiency of 10% and a thermal efficiency of 66%. The electric efficiency is slightly reduced by the need of increasing the working temperature of the condenser, but this issue is compensated by the large amount of heat produced, resulting in a good overall efficiency. It is clear that the heat recovery at the condenser is crucial in order to obtain an acceptable biomass fuel utilization. The use of wood biomass for simple power generation has too low efficiencies, and other energy conversions should be considered as an alternative.

# Chapter 5 Biomass systems simulation model – Integrated approach

The simulation of the ORC unit described in Chapter 3 is the core of the energy generation plant, but its performance is strictly related to the behaviour of other components. In particular, the performance of the biomass boiler that provides heat to the ORC unit is often the more critical part of the CHP unit. Other components that have important effects on the seasonal efficiency of the DH system are the backup and integration boilers, which can run either on biomass or on fossil fuels, and heat storage systems, which allow to avoid the continuous matching between demand and supply. Therefore an integrated approach is essential, as the DH network and the generation plant need to be considered as a combined system.

This chapter will describe the integrated simulation model of the system, which will be used for a planning case study presented in Chapter 6. Each additional component has particular features that can have consequences on the general system behaviour. Moreover, the operation of the system can be based on different strategies, which can lead in some cases to operation conditions that are far from the design assumptions.

Another fundamental aspect in DH systems analysis is the economic evaluation, both considering investment costs and running costs. In many cases the presence of incentives on renewable energy production can lead to different design and operation choices. The environmental impact of the operation of biomass systems is another important concern. However, a detailed assessment of environmental impacts cannot be performed as a general analysis but is related to each single plant and to the energy chain of the fuel that is used in the plant. For this reason, only an approximated analysis on global impacts (e.g. $CO_2$ emissions with respect to other solutions) will be provided in this study.

## 5.1 Integrated approach for system simulation

The performance simulation of a combined DH and generation plant system needs to take into account all the different components that are part of the system. A general system layout showing the main components is reported in Figure 5.1. The CHP unit is usually the core of the system, as it provides the larger amount of the annual heat required by the DH users. Other important units are the backup and integration boilers, the heat storage systems and the plant auxiliary systems (circulation pumps, fans, flue gas cleaning systems, biomass feeding systems, etc.). Moreover, the characteristics of the input biomass are a key parameter in the energy conversion process. For the same reason, also economic and environmental aspects need to be taken into consideration as well.

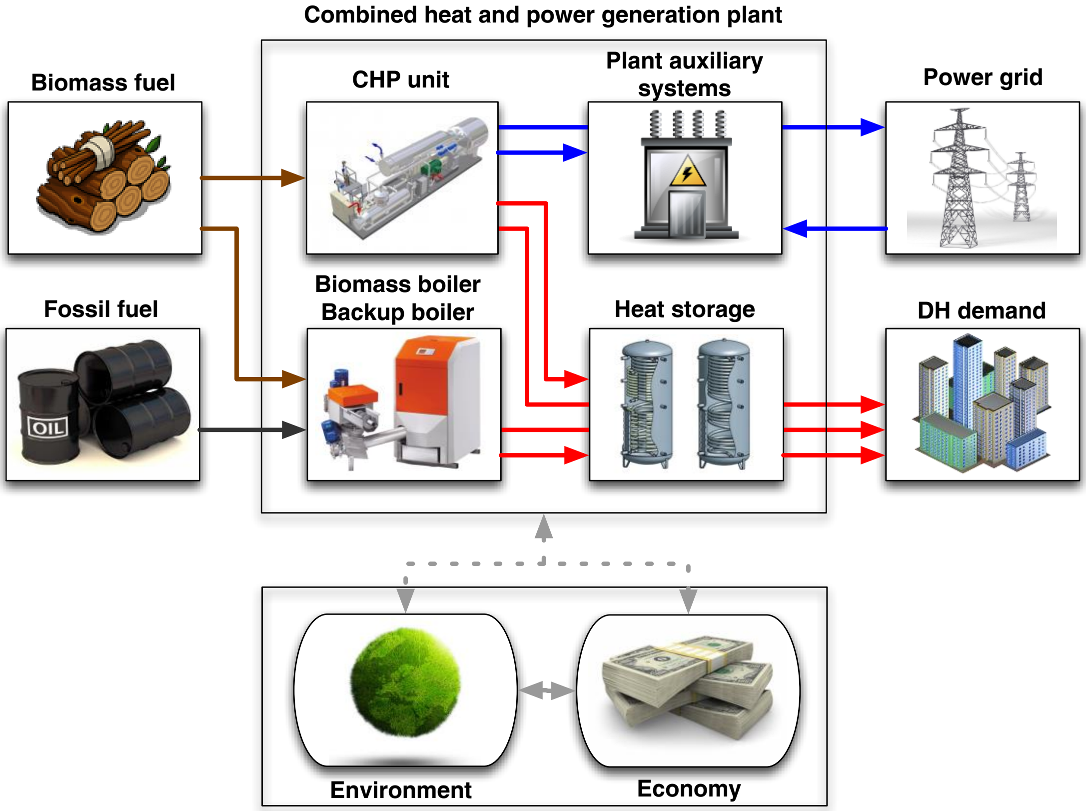


Figure 5.1 Conceptual representation of the simulation model.

This simulation has been performed in MATLAB programming language, due to its flexibility and capability of integrate different data sources. The simulation is focused on the energy generation plant, being the DH users demand independent from the generation plant behaviour. Therefore, the hourly energy need is considered as an input to the model, starting from the analysis described in paragraph 2.3.

The power grid is considered as a sink or source of electricity without limitations, i.e. depending on the hours the excess of power it is supplied to the grid and the shortage of power is provided by the grid. The connection to the grid has been set to medium voltage, as biomass systems are usually under 10 $MW_{el}$, which is the conventional limit for high voltage connections.

The ORC performance has been included in the MATLAB model using the simulation tool developed in EES, by mapping the unit performance depending on the working fluid and on the evaporation and condensation temperatures. Moreover, the operation at partial load has been considered using some correlations obtained from literature results (from [29] and [38]) and from the operation analyses described in Chapter 4. However, in most cases the ORC unit runs at nominal load for almost all the operation hours, providing the heat base load required by the network.

The components of the generation plant have in some cases strict relations between each other. For this reason an operational simulation requires a simultaneous simulation of each component for each time step. However, this model is limited to a stationary simulation of the system, considering that an hourly time step can be enough for avoiding the account of dynamic effects.

The consideration of dynamic performance would require an excessive detail for each component, with an increasing simulation time. Moreover, the dynamic operation of the system is usually limited to a negligible share of the annual operation.

## 5.2 Biomass boilers

The biomass boilers technologies have a wide variability, depending on the characteristics of the input biomass, the nominal thermal capacity and the type of heat carrier that is produced. The furnaces used in District Heating Systems have a nominal thermal capacity exceeding 100 $kW_{th}$, and they are generally equipped with mechanical or pneumatic fuel-feeding systems, allowing for an automatic operation performed through process control systems. Small and medium systems have generally furnaces with fixed bed combustion, with different grate furnace technologies available: fixed grates, moving grates, travelling grates, rotating grates and vibrating grates. The choice of the technology is mainly based on the fuel characteristics, as each of them has specific advantages and disadvantages.

The grate must be well designed and operated, in order to guarantee a homogeneous distribution of the biomass over the whole grate surface. The primary air need to be supplied equally over the different grate areas, as inhomogeneous air supply may cause slagging or increase the excess air needed for the combustion, decreasing the boiler efficiency. The movement of the biomass over the grate needs to be as smooth as possible, in order to maintain the bed of embers calm and homogeneous. This is necessary to avoid the release of fly ashes and unburned particles, and to avoid the formation of holes in the bed of embers. This requests are generally matched using continuously moving grates, bed height control systems and primary air fans with frequency control. The primary air is generally controlled separately for each section of the grate, to foster the separation of the different processes (drying, gasification and combustion). This function allows to control the furnace at partial loads down to 25% in smooth conditions [43]. The grates in biomass systems are usually water-cooled in order to increase the material durability and avoid slag formation, which is especially relevant for wood residues.

There are three main operation layouts in grate furnaces, depending on the flow direction of the biomass and the flue gases: counter-current flow, co-current flow and cross-flow. The counter-current combustion, with the flame in the opposite direction to the fuel, is generally chosen for the combustion of fuels with low heating values. Co-current flow layouts, with the flame and the fuel in the same direction, are more suitable for dry fuels with preheated primary air. Cross-flow systems are a combination of the previously mentioned layouts, and are applied in systems with vertical secondary combustion chambers [43].

The temperature control in the furnace is achieved by means of water-cooling and flue gas recirculation. The combustion systems designed for dry input biomass can be made with steel walls, whereas wet biomass systems need combustion chambers with insulation bricks. The bricks operate as heat accumulators and buffer combustion temperature and moisture content fluctuations, allowing a good burnout of the flue gases and a smooth combustion. The flue gas recirculation allows a better control than water-cooling, but it has the disadvantage of requiring an increased gas volume in the furnace and boiler sections.

Biomass boilers in DH systems can be used for simple network heat production or can be coupled to an ORC unit for CHP production. These two purposes are generally requiring different heat temperatures, and a single boiler cannot be used both for feeding the ORC unit and the DH network. ORC units usually require heat sources higher than 150°C (using superheated water, steam or thermal oil), whereas mostly DH networks operate with hot water (up to 90°C of supply temperature) or slightly superheated water (generally 120°C). The biomass boilers running on woodchips are usually subdivided in four main types depending on their output heat flow: hot water, superheated water, steam and thermal oil.

Hot water boilers are the most common and simple typology, and they are used for heat supply to the DH network or other heat users that request hot water. The use of superheated water can be determined by particular DH designs or industrial processes, or in some cases to supply small size ORC units. Superheated water boilers are usually installed for required heat temperatures lower than 160°C, while higher temperature needs (for CHP or industrial processes) are usually matched with steam boilers or thermal oil boilers. Steam boilers can require particular security measures depending on the country of installation, and for this reason in many applications thermal oil boiler are preferred, in spite of their higher investment costs.

### 5.2.1 Biomass boiler efficiency

The combustion of wood biomass shows common issues but also considerable differences with respect to the combustion of traditional fossil fuels. As a consequence, the definition of the boiler performance needs to deal with multiple aspects. Furthermore, the efficiency of biomass combustion cannot be measured continuously in real plants, due to the variability of the moisture content and the heating value of the input fuel. Laboratory performance tests usually provide a measurement of boiler efficiency at nominal load and at partial load (usually 30% of the nominal heat output). However, during its real operation the boiler may need to fit to different external conditions (e.g. biomass moisture content, water temperature, heat load, etc.).

The combustion efficiency is defined as the difference between the total energy input to the boiler and the energy content of the flue gases exiting the combustion chamber. It can be measured directly through the composition and temperature of the flue gases, along with the characteristics of the input fuel, and no further information on the boiler is needed.

The boiler efficiency is defined as the ratio between the useful heat output and the total energy input to the boiler. It is a most useful value as it takes into consideration also the radiation losses of the boiler and the unburned fuel in flue gases and ash. Due to the heat and fuel storage capacity of the boiler, a reliable measurement for the determination of instantaneous efficiency can only be performed in stationary conditions or as an average value over a certain operation period. The boiler efficiency can also be calculated in an indirect way by accounting the different heat losses, which is usually easier than performing the direct measurement.

The annual plant efficiency is calculated by dividing the useful heat by the energy input during a whole heating season. This value is generally of interest for an economic assessment of the system, rather than a comparison of different boilers. Many aspects affect the plant efficiency, including the system layout, the heat profile of the user, the climate conditions, etc. In particular the influence of part-load operation and the input biomass heterogeneity are major issues against high plant efficiencies. In particular, the part load operation of the boiler can be the result of a stationary part load (typically from 30% to 100%) or a standby mode with on/off operation of the fuel feeding.

Some experimental data on automatic biomass combustion plants have been collected in the framework of IEA Bioenergy Task 32 [44]. Combustion efficiency and boiler efficiency have been determined at stationary operation at 100%, 60%, 30% and 10% of the nominal load, using a 550 kW grate boiler test bench. The combustion efficiency resulted for 100% to 30% load in a range of 90.9 ±0.7% to 87.8 ±0.7%, the expanded uncertainty being quite small. For 10% load, the combustion efficiency was 86.1 ±2.0%. The boiler efficiency calculated by indirect determination method resulted for 100% to 30% in a range of 86.2 ±1.7% to 84.2 ±4.8%. For 10% of the nominal load, the boiler efficiency was 66.1 ±14.4%. The expanded uncertainty is still quite small for 100% load, but increases significantly with decreasing load due to the uncertainty of the thermal losses by radiation, convection and ash losses during stationary operation of the boiler. These data show a slight variation of the efficiency down to 30% of partial load, whereas the boiler efficiency at 10% is significantly lower, mainly because of the increased weight of the fraction of heat losses that has a constant trend (i.e. the radiation losses into the environment).

Some approximated models have been derived by Tillman [45] to describe the combustion. The following formula can be applied for the determination of combustion efficiency, in systems with a good separation between combustion chamber and heat transfer surface:

$$\eta_c = 96.84 - 0.28\, M - 0.064\, T_s - 0.065\, EA \tag{5.1}$$

where M [%] is the moisture in the fuel measured on a total weight basis, $T_s$ is the stack temperature and EA is the excess air in the combustion [46].

The approximation proposed by Tillman has been compared to the values proposed by some industrial boiler manufacturers in Figure 5.2. In the plot an excess air of 40% and a stack temperature of 120°C have been considered for convenience, whereas in real operation these two parameters are usually varying with moisture content. As noticeable in the chart, each boiler has different performances, depending on a number of parameters. The proposed approximation shows however a similar trend.

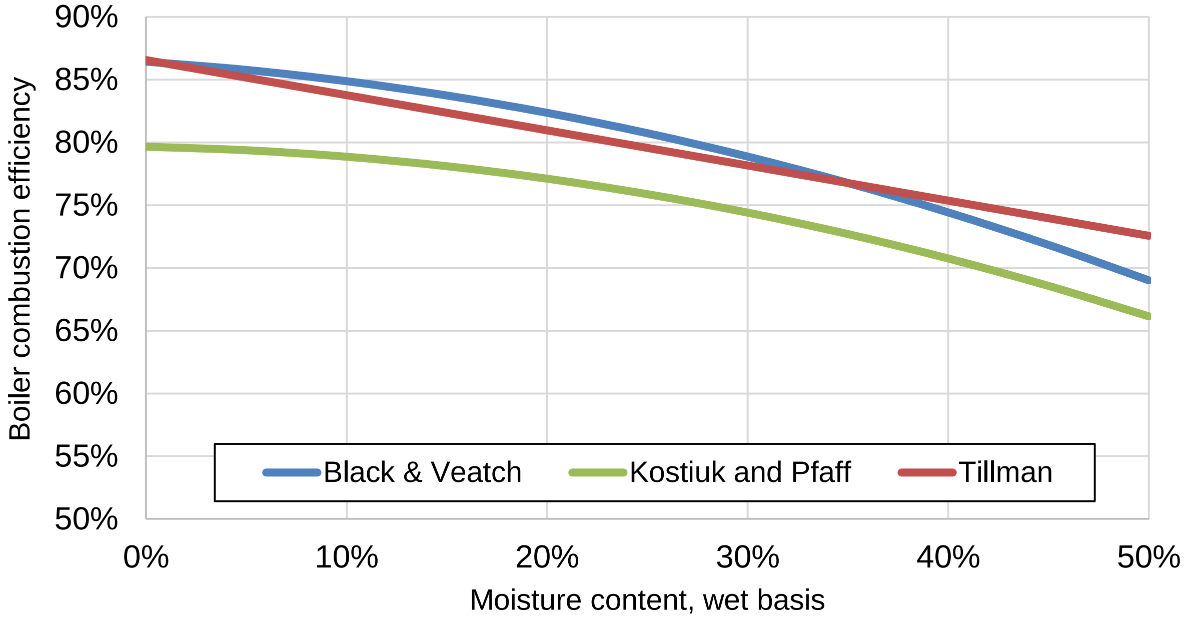


Figure 5.2 Biomass boilers combustion efficiency over moisture content.

Another important parameter for boiler efficiency is the stack temperature: the larger amount of the heat losses is usually due to the energy content of the flue gases in the stack. Depending on the type of biomass, there is usually a lower limit imposed by the possible corrosion in the stack due to the condensation of water vapour and acid gases dissolved in the water. As a result, the stack temperature and the excess air are usually maintained at relatively high values. However in some plants an economizer is used for an additional heat recovery from the flue gases. Some research has been performed on biomass condensing boilers [47], which can provide a considerable increase of the heat supplied to the users. However, the commercial applications are still limited, also because of the need of low temperature heat uses and the strong impacts of biomass moisture content and ambient conditions (e.g. the altitude).

## 5.3 Heat storage systems

A heat storage system (HSS) allows to create a delay between the heat consumption and the heat production, avoiding the need of an exact match of the supply and demand at any time. This advantage is significant when there is an important difference in the energy prices over the time, or where the possibility of splitting the demand over day and night lowers the nominal power of the generation systems needed to match the peak power. This is particularly important for DH systems, as in some cases the daily operation load has significant peaks of demand (see paragraph 2.2.3). The HSS can be an interesting solution for increasing the overall annual efficiency of the system, decreasing the primary energy consumption [48]. The HSS can also be used in renewable energy plants, where solar heat need to be collected when it is available, or where biomass generators have generally a higher efficiency if they are operated at constant load.

HSS can be for daily storage or for seasonal storage, depending on their application. Daily storage systems are usually water thanks of a size proportional to the energy requirements (usually up to 1,000 $m^3$ per single tank). Seasonal storages usually require a much larger amount of space, together with a better insulation. For this reason, often they are built in the ground, or they benefit from existing reservoirs. Seasonal storages are currently built only for solar heat systems, where the availability of surplus heat during the summer season can be stored for the winter season demand. Other technologies are being investigated in order to increase the storage efficiency (phase-changing materials, rock bed storage, etc.), but their application is still limited to demonstration or pilot plants.

The HSS installed in CHP systems are usually intended for short-term storage of hot water (daily storage or intra-daily storage), and they are built as large water thanks where water stratifies at different temperatures. The design of the HSS depends on the supply temperature and network pressure: below a water supply temperature of 100°C the HSS can be designed as an atmospheric tank, whereas for higher temperatures there is a need of a pressure vessel.

An example of the daily operation of a HSS in a DH system is shown in Figure 5.3. The HSS is loaded at night, and benefits for the possibility of being loaded by the CHP systems, and the unloading of the HSS covers the morning peak load. This application has the double advantage of increasing the CHP efficiency and lowering the morning peak demand to be supplied by the boilers. The daily profile can be highly variable, depending of the kind of users connected to the network, the generation units, the operational logics and the economic framework (variability of market prices, presence of incentives, fuel costs, etc.).

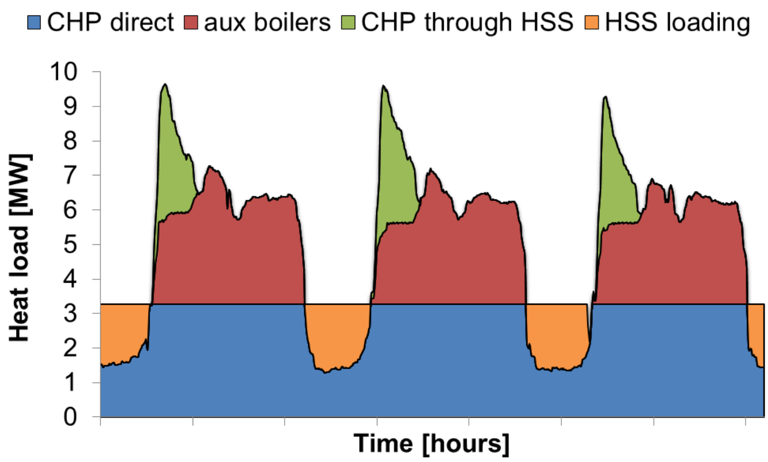


Figure 5.3 Typical daily operation of a Heat Storage System.

In the current simulation model the HSS has been considered for a day-night operation: the excess energy produced during the night by the CHP unit is supplied to the HSS, until it reaches the maximum amount that can be stored. The HSS is then completely unloaded during the morning peak, in order to limit the amount of energy required to the integration boilers. Depending on the size of the HSS, and on the amount of energy requested at night, the share of heat produced by CHP can be increased, leading to an improvement of the overall performance of the DH system. Considering such a short storage time and well-insulated water storage tanks, the heat losses into the environment account for a negligible share (common heat losses in HSS are about 1°C over 24 hours, according to [49]).

## 5.4 Other possible components

The possibility of integrating into the model other possible generation systems has been taken into account, although it has currently not been completely performed. The most interesting additional components that could be coupled to a biomass DH system are the solar panels, solar dryers and absorption chillers for trigeneration.

<u>Solar collectors</u>

The possibility of integrating solar systems for heat production is being investigated with an increasing interest ( [50], [51], [52] and [53]). The use of solar energy is currently considered as a marginal integration, even if some DH systems already have a considerable solar fraction. This solution has significant advantages for the systems that need to run also during the summer season, due to the need of supplying some types of users (e.g. domestic hot water, hospitals, etc.) or to the particular climate conditions (e.g. mountain villages). On the other hand, the solar system can be designed in order to store during the summer a part of the heat required by the users during the winter. This solution is being applied in some DH plants in Northern countries, where the installation of seasonal energy storage systems is already a commercial solution, while it is not yet applied in Southern Europe. A major limitation of solar heat is the need of low-temperature DH networks, in order to increase the solar fraction and to optimize both the production and the seasonal storage. While low-temperature networks are quite diffused in Denmark, Sweden and Germany, they are not a common solution in Southern Europe.

<u>Solar biomass dryer</u>

Another interesting application for the use of solar energy in a wood biomass system is the possibility of using it for drying the woodchips used in the plant. An example of the system layout is shown in Figure 5.4, where a comparison to the classic layout including solar collectors is proposed. In a solar dryer, the solar radiation is collected by air collectors, which are used to pre-heat the air that is needed for the drying of the biomass. The drying process has the advantage of increasing the energy content of the wood biomass supplied to the CHP unit. As a consequence, the benefits are shifted towards the heat and power production, resulting in a higher amount of final useful energy than in the case of solar collectors directly coupled to the DH network.

The comparison of the two solutions for exploiting solar radiation proposed in Figure 5.4 shows the annual energy that can be provided by each square meter of solar collector in the two configurations. This calculation shows that one square meter of solar collectors, which receives around 1.40 MWh/y of radiation (average for northern Italy), can supply 0.65 $MWh_{th}$ to the DH network. If the same radiation is used in a solar dryer to increase the quality of the input biomass, the additional amount of heat that can be supplied to the network rises to 0.87 $MWh_{th}$, and an additional amount of 0.12 $MWh_{el}$ can be produced by the CHP unit (already considering the electricity needed by the fans of the dryer).

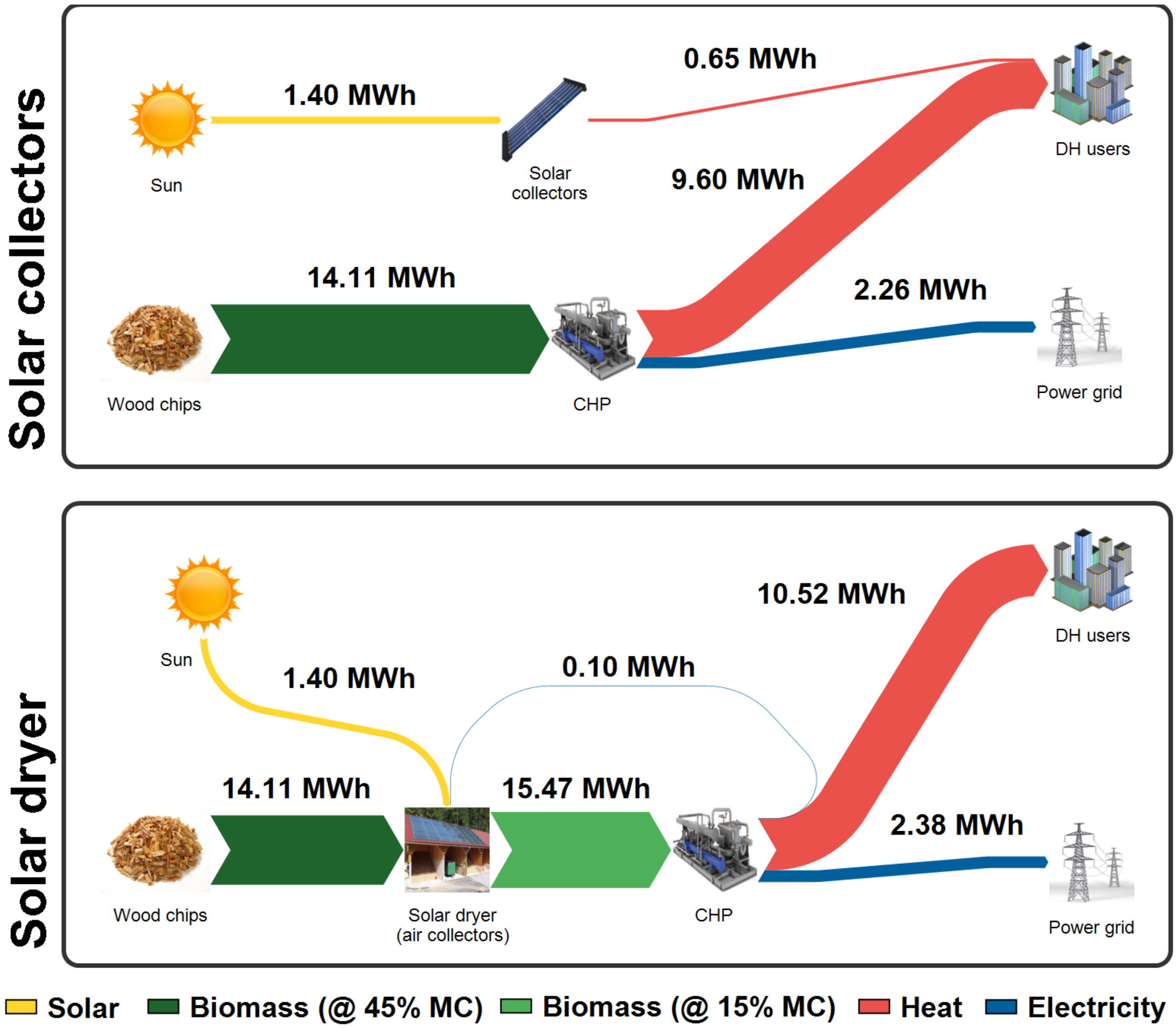


Figure 5.4 Comparison between solar dryer and solar collectors (values per $m^2$ of collector).

Absorption chillers for trigeneration

The use of district heating networks for the cooling production during summer season through absorption chillers is a solution that is being considered, especially when it allows the production of cooling through RES (e.g. biomass, geothermal). This component could be integrated into the simulation model, in order to evaluate the CHP performances also during the summer season, in the case of cooling users.

This solution is currently limited to the systems where a large availability of waste heat provides significant economic advantages in the production of cooling from absorption chillers. Its application to biomass systems has been deeply investigated in the literature, but there are currently few applications. The use of trigeneration can increase the annual utilization rate of biomass systems, and become an advantage where the CHP units are currently running all the year and dissipating heat during the summer.

## 5.5 Operational logics in the energy system

The operational logics of a DH system are among the most important drivers of the actual energy performance of the components.

In the DH systems where there is a co-presence of CHP units and boilers, usually the priority is given to the CHP, for economic and energetic reasons. The relation between the operation of CHP and integration boilers concerns both design choices and operation choices. Usually the return of investment of a CHP unit is based on high annual hours of operation, as the investment costs are significantly higher than for heat generators. In addition to economic reasons, the combined production of heat and power has generally a higher exergetic efficiency, and should therefore be preferred to separate production.

Although the energy balance of an ORC unit clearly shows that the main output is the recoverable heat, resulting in a high heat-to-power coefficient, often the electricity gains a higher importance due to its higher economic value (especially in the case of incentives). Generally speaking, a CHP unit is generally considered as power driven or heat driven, depending on its primary purpose. Considering DH systems, the CHP units should behave considering the heat as their primary aim.

For the same reason, the CHP units in DH systems should be operated during the heating season only, when the users request heat. However, in some cases the operator tries to extend the operation hours of the CHP unit, mostly for economic reasons.

The installation of heat storage systems is often necessary for the optimization of the use with a CHP unit. However, in some biomass boilers a constant operation can provide significant advantages in terms of conversion efficiency, reliability of the combustion process and emissions control. In these cases a HSS may be an interesting solution for avoiding the ON/OFF control of the biomass boiler or frequent load variations that can worsen its performance.

Considering daily charge and discharge of the HSS, the charging process is usually performed at night with a slowly heat supply from the generation units, whereas during the morning peak the discharge process is usually much faster, needing to provide a considerable amount of heat in a narrow time period. In some cases the discharge process could be slower, depending on the shape of the load profile and on the characteristics of the heat generators that are available.

In the presence of multiple heat sources, the operation priority is a key factor, and it is normally dependent on the characteristic of each generation unit, including:

- the startup and shutdown features, i.e. the dynamic behaviour during these phases: the time needed to reach nominal conditions during startup, the performance in this phase, the flexibility in varying the output load;
- the operation costs, including primarily the fuel costs but also the balance with the revenues from the energy produced. In some cases the fuel costs can be also variable depending on the hour of the day (e.g. heat pumps);
- the type of source, programmable or non-programmable, e.g. the solar radiation cannot be programmed, and therefore it should be given the highest priority when it is available;
- considering CHP units, the convenience of selling electricity to the grid could be variable depending on hourly or monthly prices;
- the compliance between the heat generator and the current network temperatures (e.g. for DH systems with variable temperatures throughout the year, some units can behave differently depending on the temperature at which the heat should be supplied).

Some general considerations can be followed, keeping in mind that each system could have particular situations leading to different operational logics. In general, the RES are given the highest priority, especially if they have no fuel costs or minimal running costs (e.g. solar, geothermal). CHP units are usually operated mostly at full load and for as much hours as possible: for this reason their size is usually significantly lower than the annual peak of consumption (they are usually sized for 30% ÷ 40% of the design heat peak).

## 5.6 Economic analysis

The economic analysis of the DH system is a fundamental aspect, as it is often the main driver in many design and operation choices. Some information are provided on the basis of the price ranges found in literature or public sources, although each case is particular and could benefit or suffer from different economic conditions. While the fuel, electricity and heat prices are reasonably reliable, it is not trivial to generalize an investment cost for such a system, as a number of different components need to be considered for an exact definition of the total investment cost. Moreover, the presence or absence of some components is often related to the specific design choices, and therefore the economic values that are used are an average value that can be improved when studying a specific case study.

As different details can affects the investment cost, the economic results that will be provided are intended to represent an average value, and they should be considered as a comparison index between different solutions rather than a real value for a particular system.

The main aspects that need to be considered for an economic analysis of a CHP system are summarized in Table 5.1.

| Investment costs | Operation costs and revenues |
|---|---|
| ORC unit | biomass cost |
| biomass furnace and boiler | O&M costs of the plant |
| heat storage system | ash and waste disposal |
| civil works for building | electricity revenues |
| biomass supply system | heat revenues |
| engineering and piping | |

Table 5.1 Investment and operation costs for CHP units

### 5.6.1 Investment costs and O&M

A detailed economic assessment of the investment cost of an energy generation plant is beyond the scope of this work, as there are a number of aspects that need to be considered on a case by case basis. However, some parametric analysis can be performed, in order to obtain an approximated value for the investment cost of the plant.

The investment costs of ORC units are reported in Table 5.2 (from [54], author's calculation from [55]). The investment cost includes the ORC unit, the biomass furnace, the thermal oil boiler, the fuel handling and supplying system, as well as the civil works, the piping and the engineering. Considering the installation of the unit in an existing DH system, the operation costs can be considered as a fixed value accounting for the work of an additional person, regardless of the size of the unit. The maintenance costs have been considered as 1% of the investment cost, in accordance with [55].

| ORC unit | **Investment cost [€]** | **Specific cost [€/kW$_{el}$]** | **O&M cost [€/y]** |
|---|---|---|---|
| 400 kW$_{el}$ | 3,794,000 | 9,485 | 68,840 |
| 600 kW$_{el}$ | 4,251,000 | 7,085 | 73,260 |
| 800 kW$_{el}$ | 4,708,000 | 5,885 | 77,680 |
| 1,000 kW$_{el}$ | 5,165,000 | 5,165 | 82,100 |
| 1,200 kW$_{el}$ | 5,662,000 | 4,685 | 86,520 |

Table 5.2 Investment and O&M costs for biomass-fired power plants with ORC units [54]

The investment cost of the heat storage system has been considered as a constant parametric value of 2,400 €/m$^3$, which is representative for the usual storage sizes that are used in DH applications (in accordance with [48]). No O&M cost has been associated with the HSS, as they are generally not requiring ordinary maintenance.

A further operation cost that need to be considered is related to the disposal of the ashes that are produced during the combustion. The disposal cost is generally in the range of 100÷250 €/t, depending on the location and the annual amount of ash. In some cases a differentiation could be established between the bottom ashes and the fly ashes, due to the different processes that are needed for disposal.

### 5.6.2 Biomass price

The cost of the biomass usually accounts for the larger part of the operation costs of a CHP system. The price of the wood chips can be highly variable, depending on the source (forest wood, biomass residuals, sawmills wastes, etc.), the moisture content, the chip size, the volume of the supply or the period of the year. The common woodchips supplies are fresh woodchips (up to M50) or partially dried woodchips (M30), whereas for some applications dried woodchips can be required (down to M20). There is no standard price for chipped wood, as without national regulations the price can vary in the different regions. Considering the Italian market, an example of woodchips price evolution is provided in Figure 5.5, where an historical trend of fir woodchips price is provided (for the Province of Bolzano).

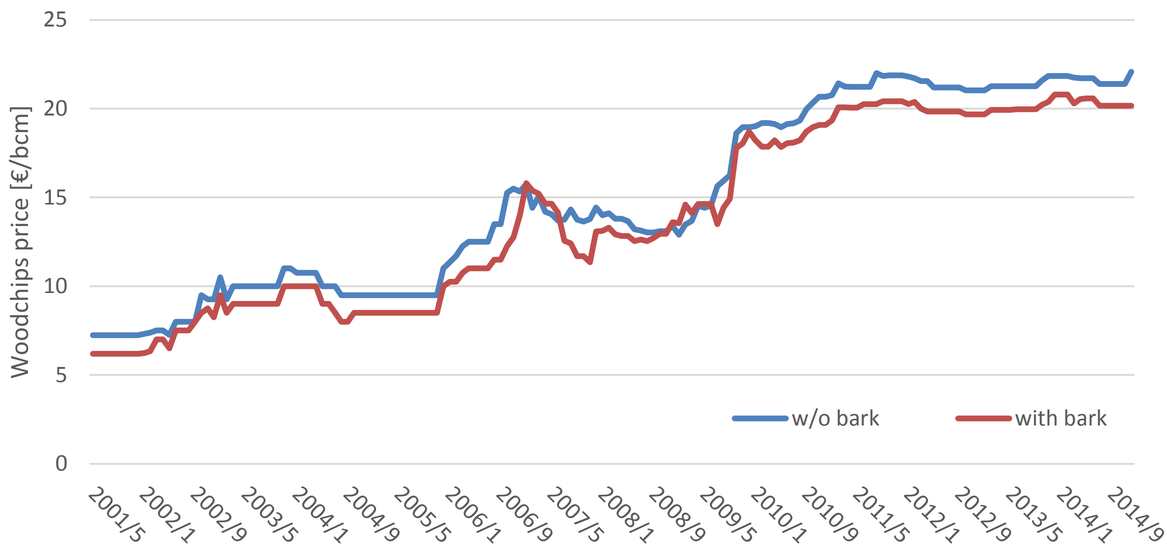


Figure 5.5 Woodchip reference price for the province of Bolzano, Italy (source: [56]).

### 5.6.3 Electricity price

The electricity price is a key parameter in the economic analysis of the system, as the price at which the electricity is sold to the grid can be highly variable depending on the location, the time, the kind of commercial agreements or the presence of incentives. The monthly price in European markets has been significantly fluctuating in last years, due to a number of factors: the medium term market prices are linked to fuel prices (natural gas, coal, oil), and the hourly prices can show different values over the day depending on the amounts of energy sold in the market. In last years the increasing share of unpredictable RES power plants (mostly wind and solar) has led to some market distortions:

in Germany the market price became negative for some hours of the day, while in Italy the peak-cost became in some cases lower than the off-peak cost, due to a high availability of photovoltaic energy during the day. The monthly energy baseload prices for some regional European markets are shown in Figure 5.6 and Figure 5.7 (data from [57]).

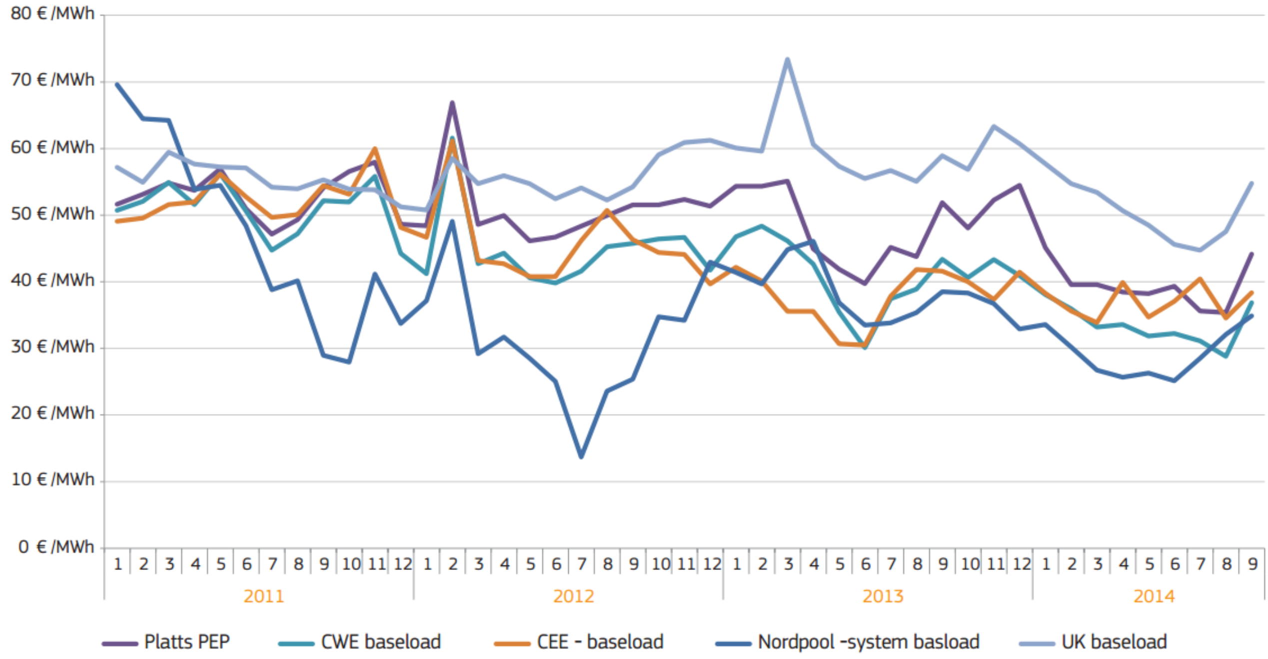


Figure 5.6 Monthly baseload prices in electricity markets (Northern and Central Europe) [57].

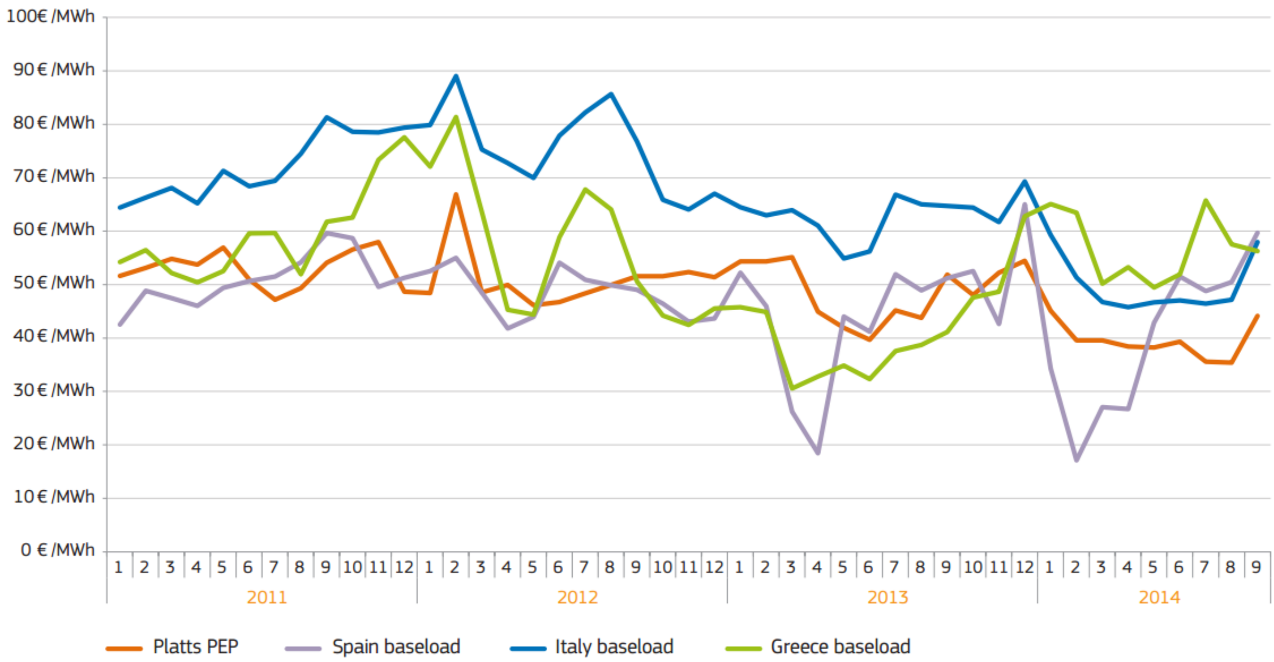


Figure 5.7 Monthly baseload prices in electricity markets (southern Europe) [57].

Considering the Italian electricity market, a longer trend (about ten years) can be seen in Figure 5.8. While the average prices are in the range 50 – 100 €/MWh, the maximum and the minimum prices show significant differences: while the maximum prices reached values higher than 300 €/MWh for some hours of operation, during the summer of 2013 the minimum price reached zero in some holydays hours.

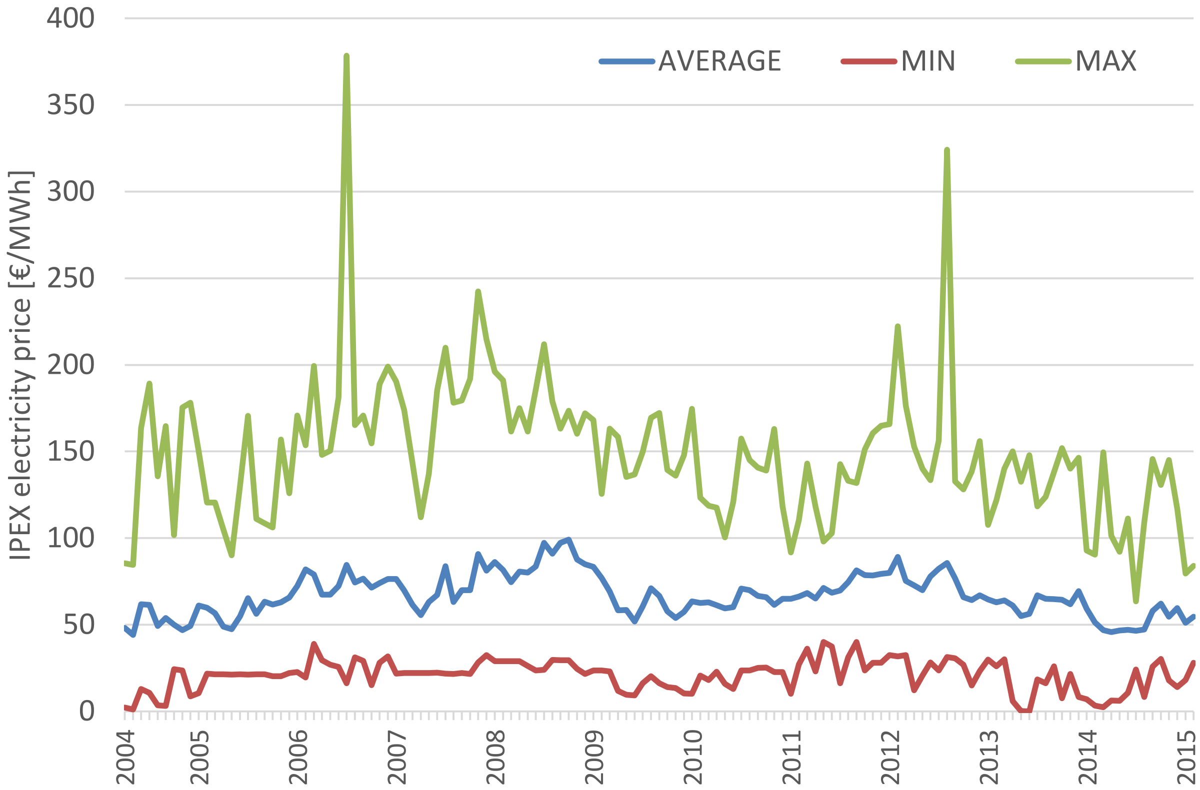


Figure 5.8 Monthly electricity prices in the Italian market (IPEX) [58].

Depending on the size of the plant, the selling price of electricity can be much linked to the Italian market (IPEX) price, or the owner can have particular agreements with an electricity wholesaler with constant prices over the day.

Moreover, considering RES power plants, the presence of incentives changes drastically the operational logics of these plants. Figure 5.9 shows the electricity incentives for electricity production from wood biomass, considering the plants with a nominal power output lower than 1 $MW_{el}$. While the old incentive (Italian law 23/07/2009 n.99) was composed by a single value of 280 €/MWh for any use of local biomass, the new incentive has some distinctions:

- power threshold of 300 $kW_{el}$, providing higher incentives for smaller systems;
- differences between biomass products and by-products;
- additional bonus of 40 €/MWh for systems operating in CHP;
- additional bonus of 30 €/MWh for systems that respect some pollutant emission limits.

Thanks to these values, the electricity sold to the grid can reach values up to 325 €/MWh, which is around six times higher than the average market value of electricity.

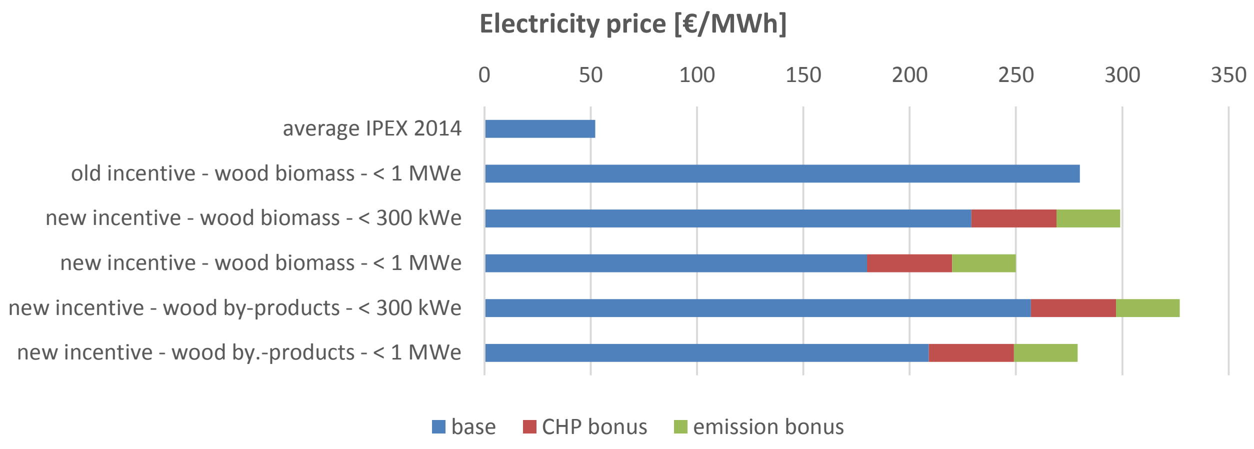


Figure 5.9 Italian incentives on electricity production.

The new incentives tried to lower the base value and to reward the smaller plants that are operating in CHP mode and with particular attention to emissions of pollutants. However, the incentives are still much higher than the average market price of electricity.

### 5.6.4 Heat price in DH networks

The heat price in DH networks is an important parameter to quantify the economic value of the heat produced in biomass CHP systems. The price of the heat is usually a result of different components, including the operating costs of the system, the type of generation units (CHP or simple heat generators), the type of fuel, the investment cost of the network. Often the heat price can be defined as a discount for the user with respect to the heat source that has been substituted (e.g. natural gas, fuel oil, etc.), usually between 10% and 20%. The price of other fuels, mainly natural gas, can also be used for the updates of the price, which are usually performed each year or each quarter.

A comparison between the average prices in some European nations is provided in Figure 5.10 (Data from [20]). These average prices can be the result of a widely differentiated set of DH tariffs, which can consist of a fixed price for each unit of heat consumed by the user, or they can include a fixed fee depending on the nominal power of the sub-station or the volume of the building. The tariff formulations are widely variable, and in Italy there is currently no standardisation of the DH heat prices. Due to this high variability, a comparison between different systems is not trivial. However, it can be stated that the current tariffs are generally in the range 70 ÷ 90 €/MWh, with some exceptions down to 60 €/MWh and up to 115 €/MWh [20].

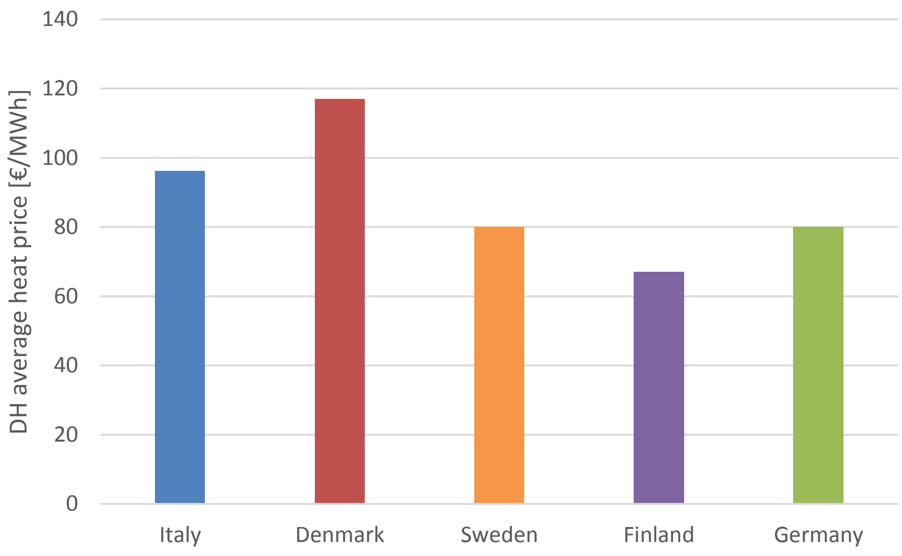


Figure 5.10 Average DH heat prices for some European countries [20].

All these aspects need to be considered for an economic balance of the CHP system. These considerations can be applied for a general evaluation of CHP systems, although in some cases there is a need of using tailored values due to specific site or component conditions. In particular, the values presented in this sections are representative of the average Italian conditions, but some values are subject to a considerable variability (especially biomass price and heat price).

An example of economic analysis of a DH system, integrated with the energy performance analysis, is presented in Chapter 6.

# Chapter 6 Simulation model application: integration of wood biomass CHP units in existing DH systems

The simulation tool has been used for an application on a test study, in order to investigate the installation of a wood biomass CHP unit in an existing district heating system.

Existing DH networks supply a considerable share of the final energy consumption for building heating in some European countries. The largest part of DH generation plants is currently supplied by fossil fuels, and there is a growing interest of increasing the share of renewable energy sources. The possibility to install a wood biomass generation unit in an existing DH system can be an effective solution to reach this objective.

In this chapter an existing medium-size DH system has been considered as test study, in order to evaluate the best sizing of an ORC unit and a heat storage system for the optimization of the conversion efficiency and the primary energy savings [54]. The economic analysis that has been performed allows to evaluate the current Italian incentives for electricity production from renewable energy sources. This financial support has been compared to the current electricity market, in order to assess the actual advantages considering the hourly operation of the system over the year.

## 6.1 Objectives

This chapter provides an example of application of the model that has been described in the previous sections. The model has been applied to optimize the size of a wood biomass CHP unit installed in a real existing DH system. The simulation has been performed using the real operation data, in order to describe the CHP behaviour in the actual conditions of operation during the heating season. This is an advantage of using existing DH systems, where historical operation data are available, as the new system layout can be optimized starting from the data of the actual consumptions and energy production.

The operation of the system has been analysed considering its energetic performance and the economic aspect. The conversion efficiency and the primary energy savings are evaluated for different sizes of the CHP unit and the HSS, given a particular DH network already in operation.

Various operation strategies have been compared, calculating the performance parameters in the cases of full year operation at full load (which is often the more diffused choice in Italy for the presence of incentives on electricity production) and heating season operation, with both power driven strategy and heat driven strategy.

The economic analysis has been performed considering the current Italian incentives for electricity production from renewable energy sources, compared to the power market price (IPEX) to evaluate the price of electricity without incentives. Moreover, some sensitivity analyses have been performed in order to take into account the possible variations of the future prices, as the market price has been subject to significant variations in last years (see paragraph 5.6.3).

This application has been performed with three main objectives:

- to assess the performances that could be reached in installing wood biomass CHP units in existing DH networks for increase the share of RES in final consumptions, in accordance with the EU 2020 targets;
- to perform an analysis on the current Italian incentives, e.g. to verify if they are defined in accordance to a maximization of the energy conversion efficiency in the systems;
- to compare different operation strategies, from an energetic and economic point of view, in order to assess the behaviour of heating season operation with respect to the widely applied full year operation.

## 6.2 Methodology and case study definition

As described above, the goal of this analysis is to perform an evaluation of the advantages that could be reached by installing an ORC unit and a HSS in an existing medium-sized DH system, considering both energetic and economic aspects. An existing case study will be considered for this simulation, using real data for the DH network heat demand for a better consistency with the actual operation conditions.

The DH system considered as case study is located in Leini, a little town of about 15,000 inhabitants in the outskirts of Torino. About 500,000 cubic meters of buildings (residential, commercial and public administrations) are connected to a 12-km DH network supplied by two biomass boilers (of 5 $MW_{th}$ each) and a natural gas backup boiler (3.5 $MW_{th}$). The annual thermal energy supplied by the system is about 17 GWh, with a consumption of more than 9,500 tons of chipped wood. The final energy consumed by the users is about 14 GWh, and the average network losses are around 19% (2013 data) [16].

In the current configuration the system produces only heat (no CHP), without any HSS coupled to the boilers. The boilers are experiencing an average annual efficiency quite low, due to the frequent partial and variable load operation. In the simulation presented in this section, one of the boilers is substituted with a thermal oil boiler supplying a CHP unit and a heat storage system that can be used to balance the heat demand variations.

This case study has been chosen to represent the average conditions for a medium DH system located in northern Italy. Its climate conditions, with 2,722 degree days, are in line with the systems located in the plains, while mountain systems have usually colder climate conditions. Considering the specific heat consumptions, this case study is much similar to Torino DH system (see Figure 2.10), which has a similar climate but is about 100 times larger. This comparison shows that the DH demand is generally not affected by the size, but rather by the climate conditions and the type of the buildings.

The amount of daily energy produced from each boiler in the current system configuration has been registered for five heating seasons, from 2007/2008 to 2011/2012. The monitoring system can measure the heat production with a more precise time step, but currently these data are not stored in the database and therefore they are not available as historical trend. For this reason, the Torino DH dataset has been used to estimate the daily variation in the heat load of the users. This assumption is justified by the geographic proximity of the systems, resulting in similar climate conditions, and by the similar users' typology. The heat demand has been scaled proportionally to the building volume supplied by the DH network and the resulting model has been applied to the case study for the year 2010 for this analysis. This year has been chosen because of a complete availability of operation and weather data, without any missing data point. The simulation has been performed with an hourly time step.

The main factor in the choice of a CHP unit is usually the nominal power output, affecting both technological and economic constraints. As a general rule, the CHP unit is usually designed in order to cover less than half of the thermal peak load, in order to operate at its rated output for a sufficient number of hours over the year. In the Leini DH the maximum heat required by the network reached 9.8 $MW_{th}$ in 2010, while the total nominal power of the wood-fired boilers is equal to 10 $MW_{th}$ (an additional backup boiler running on natural gas is available, but has seldom been used).

Considering the behaviour of the heat load, the CHP unit may have a useful thermal power lower than 5 $MW_{th}$. As a result, the power range of the ORC for the parametric analysis has been set from 400 $kW_{el}$ to 1,200 $kW_{el}$, with a gross electrical efficiency of 19.0% and a thermal efficiency of 77.9% (considering the ratio between the output energy and the heat supplied to the ORC unit), and a heat to power ratio equal to 4.1. The nominal efficiencies can be considered constant in this range of power, in accordance with the data provided in Figure 3.4. The nominal efficiency of the thermal oil boiler coupled with the ORC unit has been set to 85% (considering the LHV of the fuel). In the full load operation, the CHP unit is assumed to work continuously at nominal power conditions, coupled with a HSS in order to recover part of the surplus heat produced during off-peak hours.

The HSS performance has been considered on the basis of a supply temperature of 90°C in the DH network and a return temperature of 60°C. The HSS simulation has been performed considering daily load/unload cycles, without accounting for eventual infra-day cycles.

The remaining biomass-fired boiler currently in operation is assumed to operate as auxiliary boiler, in order to supply the excess heat demand. The actual efficiency of the boilers at partial load is currently under investigation, and therefore an estimated value of 80% has been assumed as an average annual efficiency. This value is lower than the nominal efficiency of the new thermal oil boiler in order to take into account the partial load operation and the age of the boilers. The DH grid losses have been calculated for the year 2010, and they are equal to 15.4%.

Overall CHP system efficiency and primary energy savings have been selected as main indicators for the system performances. The indicators are calculated considering all the different conditions throughout the year that have been analysed by the simulation tool. The overall CHP system efficiency is the ratio between the sum of electricity and useful heat (supplied directly or through HSS to the network) produced by the CHP unit and the total biomass consumption of the unit. The efficiency has been calculated for the whole period of operation, considering all the different conditions that occurred during the year.

Primary energy savings have been evaluated calculating the PES index (Primary Energy Saving), as defined in the Directive 2004/8/EC (and transposed in Italy with the *DM 4 agosto 2011*) as follows:

$$PES = \left(1 - \frac{1}{\frac{CHP\ H\eta}{Ref\ H\eta} + \frac{CHP\ E\eta}{Ref\ E\eta}}\right) x\ 100\% \qquad (6.1)$$

Where:

| | |
|---|---|
| PES | is the Primary Energy Saving index; |
| CHP Hη | is the heat efficiency of the cogeneration production defined as annual useful heat output ($Q_u$) divided by the fuel input used to produce the sum of useful heat output and electricity from cogeneration ($F_{in}$); |
| Ref Hη | is the efficiency reference value for separate heat production; |
| CHP Eη | is the electrical efficiency of the cogeneration production defined as annual electricity from cogeneration (E) divided by the fuel input used to produce the sum of useful heat output and electricity from cogeneration ($F_{in}$); |
| Ref Eη | is the efficiency reference value for separate electricity production. |

The annual useful heat output ($Q_u$), the annual electricity produced (E) and the fuel consumptions ($F_{in}$) are calculated by the simulation tool.

The efficiency reference values listed above are defined in the annexes of the DM 4 agosto 2011; for wood-fired systems Ref Eη is 0.33 and Ref Hη is 0.86. Some corrections are applied to Ref Eη as a function of geographical position (and consequent average ambient temperature). The resulting value for Ref Eη in this case study is equal to 0.33369.

In Italy a CHP system smaller than 1 $MW_{el}$ needs to reach a positive PES to be considered as CAR (High Efficiency Cogeneration), whereas larger systems need to reach at least a PES of 0.1. These thresholds are assumed to set the eligibility of a system to receive incentives related to high efficiency cogeneration (defined as "CHP bonus").

## 6.3 Economic framework

A basic economic evaluation, performed through the calculation of the simple payback time, has been performed considering some parameters derived from the specific case study (e.g. biomass characteristics, component size) and the Italian incentive framework for RES.

The value of investment cost for the ORC has been already described in paragraph 5.6.1 (see Table 5.2, [55]). This value includes the ORC unit, the wood-fired thermal oil boiler, the pipe connections, the building and the design and installation costs. The cost of the HSS has been estimated equal to 2,400 €/$m^3$. No other investment costs have been taken into account since the DH system is already in operation. When considering new DH systems, the cost of the network is usually a considerable share of the total investment.

The operational costs are mainly related to biomass consumption and in a minor part to maintenance costs (including ash disposal, auxiliary consumptions, etc.). The lower calorific value for chipped wood has been assumed to 3 kWh/kg (considering chipped wood with a moisture content of 35%), according to the current biomass supply conditions in Leini, and the base price for biomass equal to 75 €/t (corresponding to 25 €/MWh). The maintenance costs for the ORC unit have been expressed with respect to Table 5.2 (the annual cost of a person plus 1% of the investment cost, [55]).

The profits of the system are related to the incomes from the heat supplied to the users, the electricity produced and the available incentives. The current price of the heat sold to final users depends on many parameters, and is often defined by different tariff formulations. A base price of 90 €/MWh has been considered, according to the current conditions of the case study. This price is in line with the average DH price in Italian systems (see paragraph 5.6.4).

From January 2013 the electricity produced from renewable sources in Italy is promoted with a feed-in tariff described in *DM 6 luglio 2012*. The base price offered for biomass-fired systems (considering chipped wood produced from forests) is equal to 180 €/$MWh_{el}$ for nominal electric power between 300 $kW_{el}$ and 1 $MW_{el}$, and 133 €/$MWh_{el}$ for power larger than 1 $MW_{el}$. Some additional bonuses can be added to this base price, as defined in the same Decree:

- "CHP bonus" of 40 €/$MWh_{el}$ for units operating with "high efficiency CHP" (i.e. PES>0 for $P_{el}$<1 $MW_{el}$ and PES>0.1 for $P_{el}$>1 $MW_{el}$);
- "emission bonus" of 30 €/$MWh_{el}$ for systems respecting prescribed pollutant emission limits for $NO_X$, CO, $SO_2$,TOC and dust.

A sensitivity analysis has been performed considering the variations of heat price and electricity price, comparing the results with the current incentive framework. The aim of this analysis is to provide a methodology to assess the effect that the incentive tariff can have in lowering and shifting the optimal layout of the system.

## 6.4 Main results of the simulation

The behaviour of the system has been simulated with an hourly time step, considering the share of heat supplied by each generation unit. Figure 6.1 show an example of the hourly operation during one year of a CHP unit of 450 $kW_{el}$ coupled to a HSS of 185 $m^3$ (corresponding to 100 $m^3/MW_{th}$), which is operated at full load during the heating season. In this case the main share of the heat from CHP is provided directly to the network, but a significant share is supplied through the HSS, allowing for overcoming the matching between demand and supply. The CHP provides a total of 55% of the annual heat demand of the DH network, while the remainder is provided by the biomass boilers. The amount of energy that is dissipated during the mid-seasons is lower than 2% (considering a full-load constant operation of the ORC).

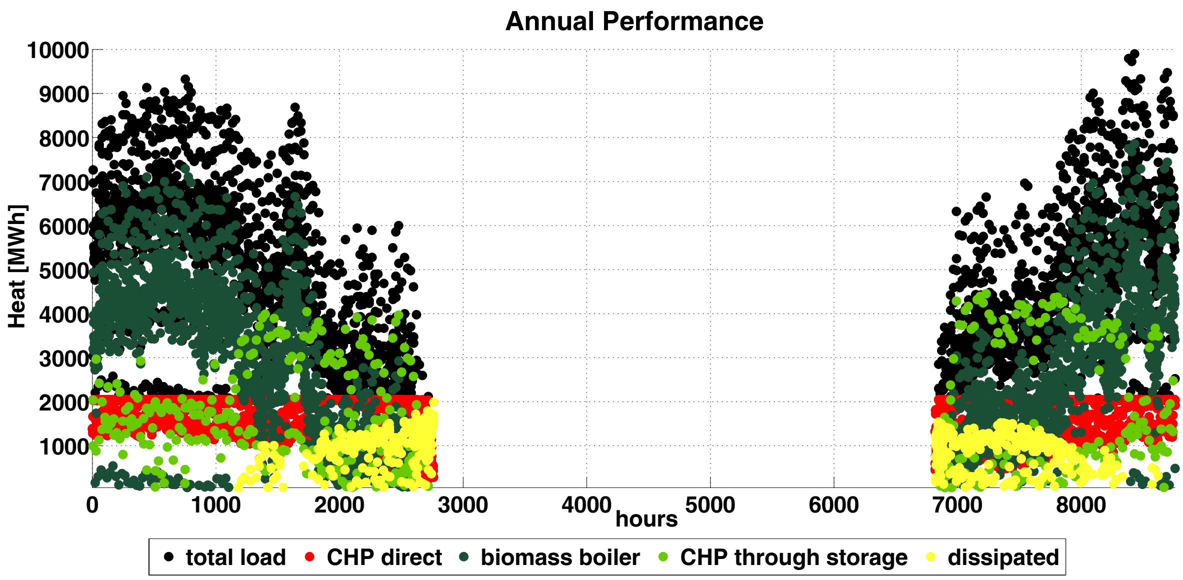


Figure 6.1 Hourly load over one year of operation (CHP size 450 $kW_{el}$).

The same simulation has been performed with variable CHP and HSS sizes, in order to assess the relevance of both parameters in the performance of the system. Figure 6.2 provides an indication of the behaviour of the overall CHP system efficiency, while in Figure 6.3 the effect on PES index is reported. The two plots show a similar trend, as in the current system these two indexes are strictly correlated.

The optimum system configuration for this case study, considering only the energetic performance, requires a small CHP system (under 600 $kW_{el}$) and a HSS larger than 100 $m^3/MW_{th}$. However, the installation of a HSS larger than 150 $m^3/MW_{th}$ appears to have negligible effect on the efficiency, especially for smaller CHP systems.

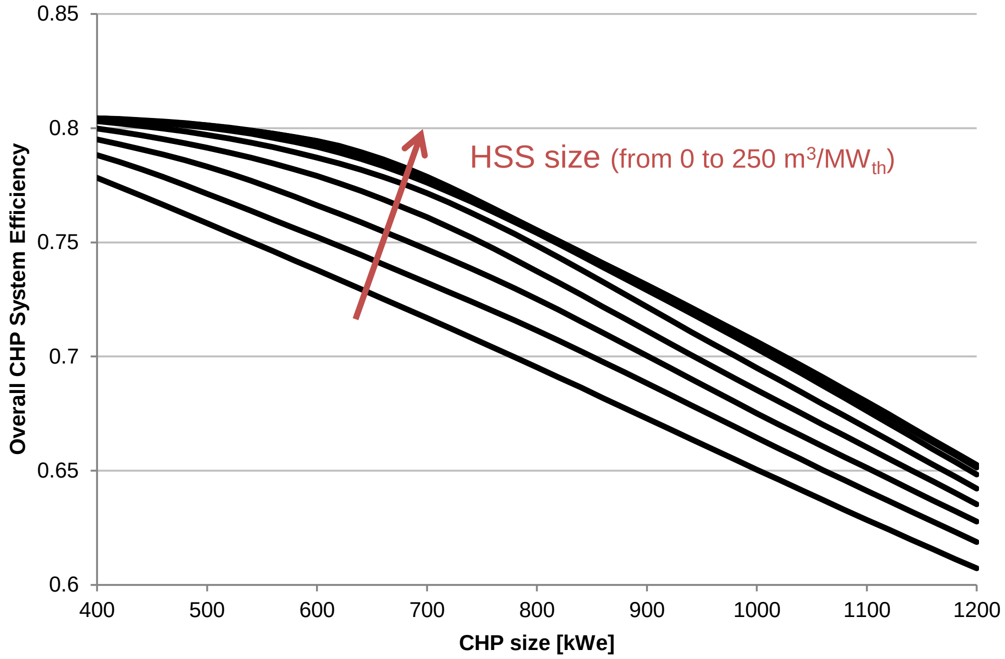


Figure 6.2 Relation between the overall CHP system efficiency and the CHP and HSS size.

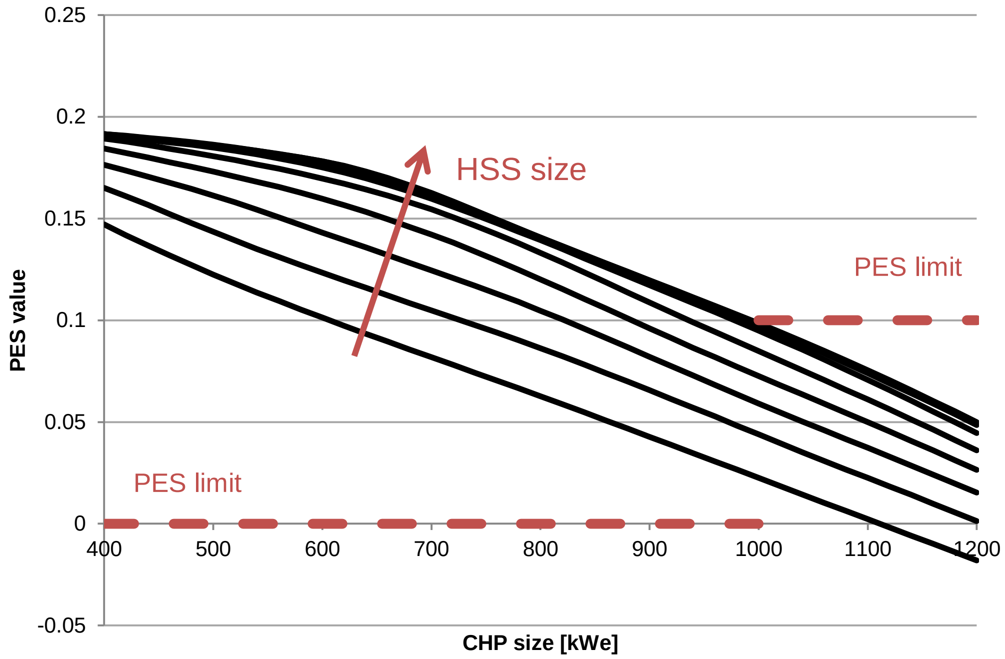


Figure 6.3 Calculation of the PES w.r.t. the CHP and HSS size.

The larger systems have generally a lower efficiency compared to the smaller ones because of the share of heat that needs to be dissipated during the middle seasons, when the heat required by the users is lower. This share could be lowered by part load operation of the ORC, but this solution is not usually considered in real systems. Moreover, it has to be recalled that the heat dissipation requires additional energy consumptions related to the operation of the cooling towers, which has not be considered in this study.

The case study under investigation has been analysed also from the economic point of view, considering the simple payback time as the main output parameter for some comparisons and considerations. Two cases have been considered, in order to compare the current market conditions (first case) and the effect of the incentives (second case). The results are in Figure 6.4.

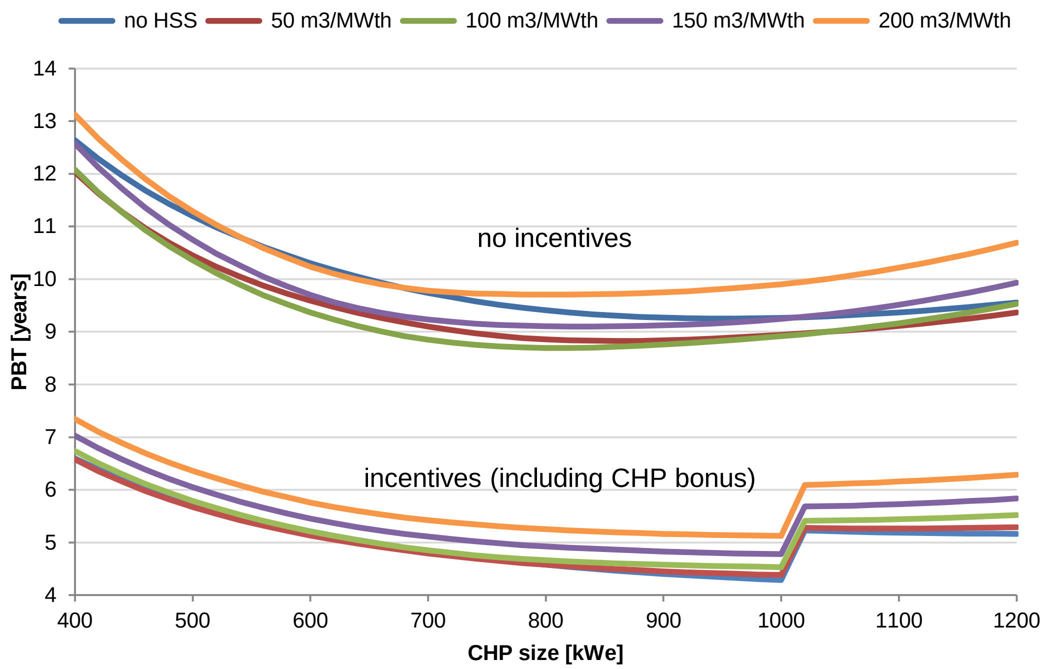


Figure 6.4 Relation between simple payback time and CHP size, for different HSS sizes.

The first case examined refers to the current Italian market, without considering the incentive tariff for electricity production from renewable sources. The average national price for electricity has been considered equal to 75 €/MWh$_{el}$.

The results of the calculation are showed in the upper curves of Figure 6.4. The minimum payback time (8.7 years) occurs for an 820 kW$_{el}$ ORC unit and a HSS of 100 m$^3$/MW$_{th}$, corresponding to about 335 m$^3$. However, the PBT has slight variations in a wide range of parameters, resulting in significant higher values only for small ORC units coupled to large HSS. The economical optimum in this case differs from the energetic optimum, but in both cases the variations are low and therefore an acceptable solution can be found.

The same analysis has been carried out for the current Italian incentive framework, as described previously. The lower curves reported in Figure 6.4 show some significant differences with respect to the upper ones. The presence on the incentive on electricity production lowers the PBT range, which is lower than 7.5 years for all the cases under examination. The minimum value of the PBT occurs at 1 MW$_{el}$, in correspondence of the discontinuity of the incentive.

A secondary effect of the incentive tariff is evident from the modification of the differences between the curves: as the electricity becomes much more profitable than heat, the investment for a HSS is not economically justified. Thus, the optimum values of PBT are associated with systems without HSS or with a very small one, in contrast with the energy performance analysis showed in Figure 6.2 and Figure 6.3. In this case it is not possible to find an optimal solution that combines both the energetic and the economic point of view.

Some sensitivity analyses have been performed with respect to electricity price, heat price and biomass price. The base prices are 75 €/MWh for electricity, 90 €/MWh for heat and 25 €/MWh for wood biomass. In all the cases a HSS of 100 $m^3/MW_{th}$ has been considered.

Figure 6.5 shows the variation of the payback time over the electricity price. The average annual IPEX prices are marked on the plot, considering the range of average prices from year 2005 to 2014, as well as the values of the base incentive. The lines show that the smallest ORC system is significantly different from the others, due to a higher specific investment cost. It has to be reminded that the previous incentives on electricity production from local biomass reached 280 €/$MWh_{el}$, which is a value outside the current plot.

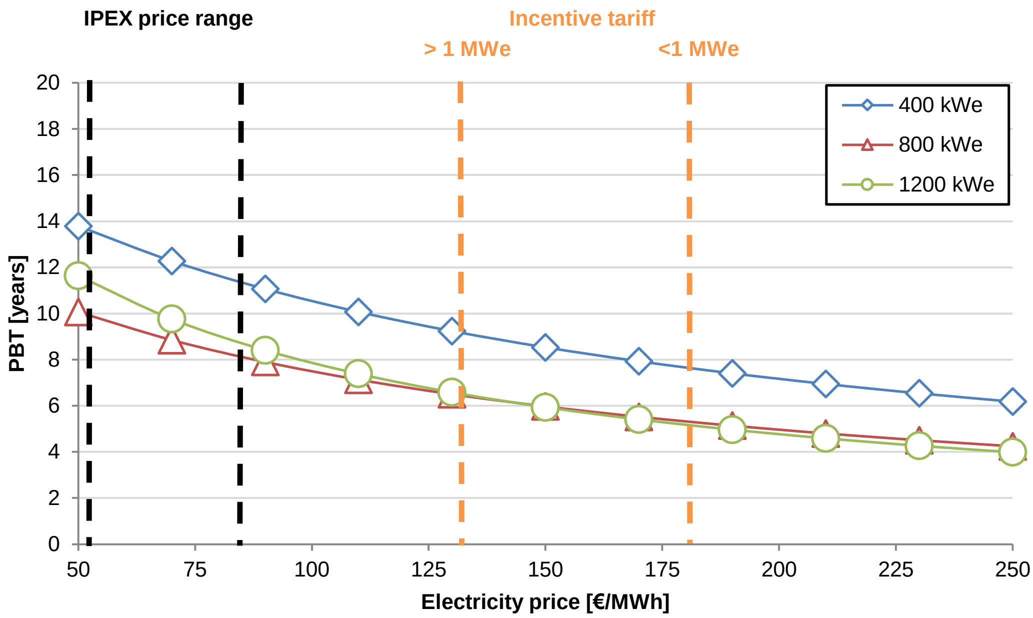


Figure 6.5 Sensitivity of payback time over electricity price.

The variation of the heat price (Figure 6.6) has a greater effect on the PBT, as the quantity of heat supplied to the users is higher than the electricity produced. However, there are currently no incentives on the heat production for the power range under examination, therefore the range of variation of heat price remains lower than the previous case. It has to be observed that the same reduction in PBT can be achieved by a lower increase of the heat price with respect to the current incentive on electricity price. However, a more complex analysis is needed in order to compare the effort required at National level to define such an incentive.

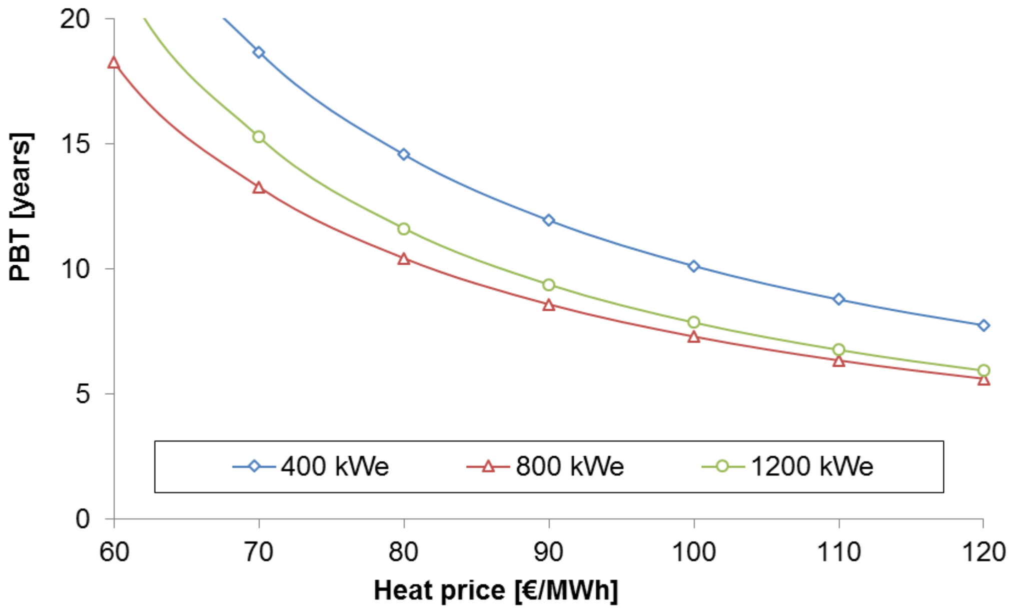


Figure 6.6 Sensitivity of payback time over heat price.

The third sensitivity analysis refers to the biomass price (Figure 6.7), which can vary depending on the material, the origin, the transport costs, etc. If the biomass is the waste from some process (e.g. pruning residues) its value may be equal to zero, but the fuel quality in these cases is often very poor, and consequently the actual conversion efficiency should be assessed.

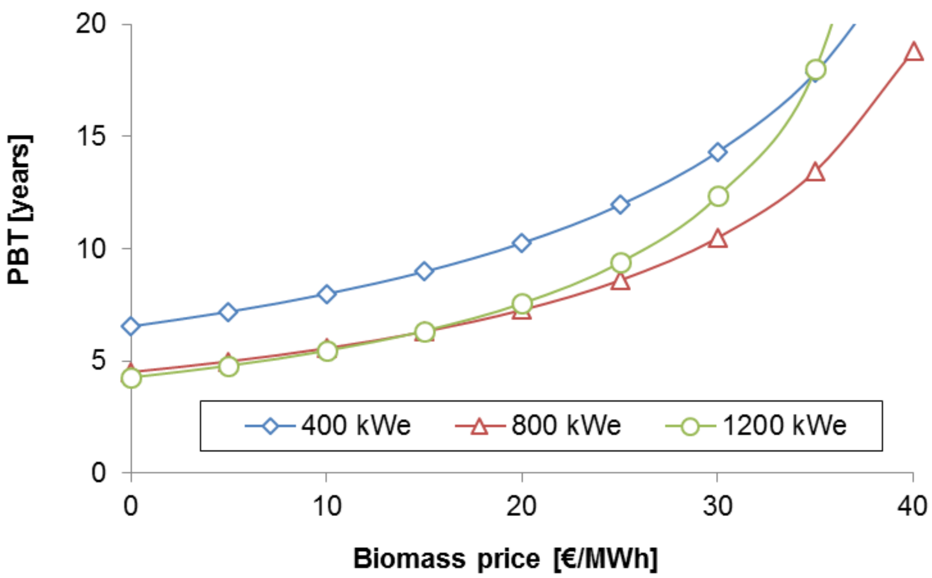


Figure 6.7 Sensitivity of payback time over biomass price.

Considering the annual economic balance, the main revenue is related to the heat sales, ranging from 53% to 58% depending on the case. On the electricity side, the incentive value is almost twice the electricity price for units smaller than 1 $MW_{el}$, and slightly higher for larger plants, but the amount of energy produced remains lower than heat, due to the characteristics of the ORC units. Looking at the operation costs, 80% to 90% of the cost is due to the biomass, while O&M costs account for the remaining part. The share of HSS in the total investment cost varies from 0% to 34%, and in the optimum configuration without incentives it is equal to 14%.

The system configuration showing the best payback time without incentives (ORC unit of 820 $kW_{el}$ and HSS size of 335 $m^3$) is used as reference also for an operational strategy analysis of the system. In all the cases considered above, the system is operated only during the heating season, where there is a heat demand from the DH network. However, due to the incentive feed-in tariff on electricity production, it is not infrequent that some systems are operated throughout the year to maximize the revenues from electricity production.

The comparison of these two different operational strategies of the system (Figure 6.8) underlines that the full year operation is profitable only in the presence of incentives for electricity production from renewable sources: without incentives, the payback time remains much lower in heating season operation. Moreover, the greater the incentive, the greater the advantage of this operation mode. It has to be noticed that for this specific case study the CHP bonus, granted only to the share of the electricity produced in "high efficiency CHP", does not affect this trend in a significant way.

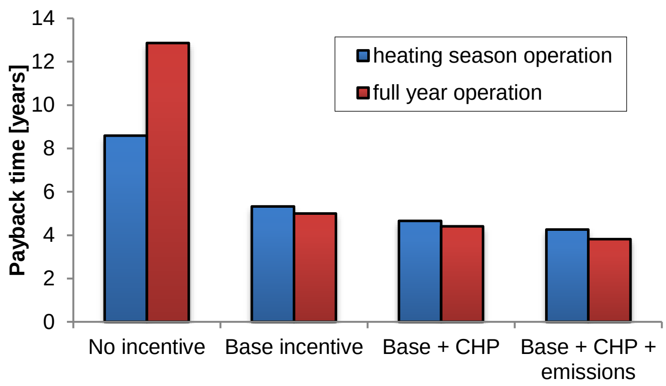


Figure 6.8 Simple payback time for heating season operation and full year operation.

These results are valid in the frame of the current Italian feed-in tariff. Nevertheless, the same methodology can be extended to any incentive framework. The new incentives (*DM 6 luglio 2012*) that has been considered in this application have tried to limit the excessive reward on electricity production by lowering the feed-in tariff with respect to the previous incentive framework (*Tariffa Omnicomprensiva*). While the government has decreased the incentive tariff, at the same time the average IPEX value has seen a significant drop (about 25% from 2011 to 2014), and therefore the tariff remains significantly higher than the average market conditions. The simulation results clearly show that this reduction is still not sufficient, and other measures need to be performed in order to promote an efficient use of the CHP units in biomass systems.

Other kind of incentives (e.g. quota-based incentives) could lead to different results, reaching an economic optimum with a higher energy performance. However, these alternatives are depending on many different parameters, and could be the object of further analyses.

# Conclusions

This PhD Thesis provides a description of an integrated approach that has been used to analyse the operation of wood biomass CHP units in District Heating systems. A simulation model has been developed, considering the behaviour of each component of the system in order to provide a comprehensive analysis. The comparison with real operation data has been used to verify some hypotheses and results.

The approach of the analysis has been chosen in order to be applicable for three different purposes: (1) system feasibility assessment, (2) operation analysis of real systems and (3) support in local energy planning. The use of the same tool has the advantage of providing comparable results for these different applications.

Considering the demand side, the analysis of different DH systems provided interesting relations between systems with much different size but similar characteristics. On the other hand, it has been noticed that a number of parameters influence the heat demand. Therefore an approximated model that can be used to precisely describe any DH network cannot be developed. Each potential DH system needs to be carefully analysed in order to estimate its operation conditions, as the system behaviour throughout the year has larger impacts than the nominal design conditions have.

The possibility of analysing the real operation of ORC units with a narrow time step has been a crucial advantage for understanding some of the criticalities of the biomass plants. Again, a well-designed system needs to foresee the expected operation conditions that could be faced over the years. Sometimes a larger flexibility should be preferred to a higher nominal efficiency, as the testing conditions are often not line with the actual operation of the CHP units. These aspects become particularly significant for biomass systems, where the characteristics of the input fuel have a larger variability than in the case of traditional fossil fuels.

The system approach that has been chosen for the simulation model allows to take into account the effects of the behaviour of each component of the system. Even if the CHP unit is usually providing the largest amount of heat over the year, its operation is strictly related to other components such as heat storage systems and integration boilers. The operational logics, including the generator priorities, have a fundamental importance in the actual system performance, and an integrated simulation of all components is essential in order to account for their effect.

The simulation tool can be used for the evaluation of energy policies, e.g. by estimating the effect of the incentives for the promotion of RES and energy efficiency. The incentives on energy production from RES are currently a diffused mechanism aiming to promote the European goals set for 2020 and 2030. An example of this evaluation has been proposed in this work, analysing the current Italian incentives for electricity production from RES, which are among the highest in Europe. The focus has been set on a specific subject: the integration of a wood biomass ORC unit coupled to a heat storage system in an existing DH network. The results show that the current incentives are not promoting the highest conversion efficiencies, rather focusing too much on electricity production and neglecting the benefits of the heat that can be produced in cogeneration. Moreover, these incentives often promote a full-year operation of the ORC unit, resulting in a huge amount of operating hours with a total heat dissipation, thus causing overall conversion efficiencies lower than 15%.

Wood biomass systems are still facing some margins of performance increase, especially for medium or small size systems. A critical aspect is the difference that can be noticed between the design conditions and the actual operation, where the main issues are usually related to the biomass quality (i.e. moisture content and chips size) and to the frequent variable load operation. A continuous characterization of the input biomass is often impossible, therefore a significant attention must be paid during the fuel supply phase. The optimization of the efficiency is also sometimes hindered by the low cost of the biomass, which can lead to focus the interest merely on the output power rather than on the conversion efficiency.

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