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Advanced Waste-To-Energy Cycles

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Introduction

The socio-economic growth of the most industrialized countries has involved a progressive increase of waste production. The increase in environmental and healthy concerns, combined with the possibility to exploit these ordinary life by-products as a valuable energy resource, has led to explore alternative methods for waste final disposal. In this context, the energy conversion of Municipal Solid Waste (MSW) in Waste-To-Energy (WTE) power plant is one of the principal means of an integrated waste management; its potential is increasing throughout Europe, both in terms of plants number and capacity, furthered by legislative directives.

The dominant technology for energy recovery from MSW is direct combustion over a moving grate with the generation of superheated steam feeding a steam turbine in a Hirn cycle. The amount of energy recovered from the MSW combustion can vary significantly with the characteristics of MSW fed into the boiler (composition, mass flow rate and lower heating value), the combustion technology, the configuration and features of the recovery boiler (adiabatic or integrated) and the characteristics of the thermodynamic cycle. Due to the heterogeneous nature of waste, some differences with respect to a conventional fossil fuel power plant have to be considered in the chemical-to-electrical energy conversion process. The thermodynamic efficiency of a WTE power plant is constrained, mainly, by the following aspects: i) the maximum temperature of the steam cycle which is limited by the well-known corrosion problems, mainly affecting the high temperature section; ii) as a consequence of steam superheated temperature constrains, low evaporative pressure are necessary in order to avoid high fraction of liquid at the steam turbine outlet; iii) the typical modest power output and steam mass flow rate of a WTE power plant imply low steam turbine isentropic efficiency. As a consequence, the thermodynamic efficiency of WTE power plants typically ranging in the interval 25% ÷ 30%.

The new Waste Framework Directive 2008/98/EC promotes production of energy from waste introducing an energy efficiency criteria (the so-called “R1 formula”) to evaluate plant recovery status. Although the energy efficiency criteria can be regarded just as a starting point for all the Member States, it constitutes a reference to create quality standards for waste recovery. The energy recovery status provides incentives for future investments in WTE plants located close to energy customers. The aim of the Directive is to drive WTE facilities to maximize energy recovery and utilisation of waste heat, in order to substitute energy produced with conventional fossil fuels fired power plants.

This calls for novel approaches and possibilities to maximize the conversion of MSW into energy. In particular, the idea of an integrated configuration made up of a WTE and a Gas Turbine (GT) originates, driven by the desire to eliminate or, at least, mitigate limitations affecting the WTE conversion process bounding the thermodynamic efficiency of the cycle.

The aim of this Ph. D thesis is to investigate, from a thermodynamic point of view, the integrated WTE-GT system sharing the steam cycle, sharing the flue gas paths or combining both ways. The carried out analysis investigates and defines the logic governing plants match in terms of steam production and steam turbine power output as function of the thermal powers introduced.

The framework of this research activity integrates into a collaborative project involving *Alma Mater Studiorum - Università di Bologna - DIEM* and the multi-utility company *Gruppo Hera- Divisione Ingegneria Grandi Impianti*. Hera is one of the leading Italian multi-utility companies operating, in particular, in the design and managing of WTE facilities. Thus, the HERA-UNIBO joint research project is aimed at assessing the possibility to integrate HERA's WTE power plants with GT.

Structure of the manuscript

The thesis is divided into four main parts.

Part I presents an overview of the scenarios of interest and motivates the work by outlining the fundamental key aspects concerning the waste to energy conversion process. In particular, Chapter 1 presents an overview on municipal solid waste production and disposal for both, the European and the Italian contest. Chapter 2 introduces and describes the basic concepts related to Waste-To-Energy conversion process, highlighting the most relevant aspects binding the thermodynamic efficiency of a WTE power plant.

Part II. In Chapter 3, a thermodynamic analysis is carried out in order to quantify the influence of the main steam cycle parameters and plant configuration on WTE efficiency. The aim of this preliminary analysis is to understand and compare possibilities and benefits of a thermodynamic cycle upgrade for a WTE power plant.

Part III explains and discusses the Hybrid Combined Cycle (HCC) concept. In particular, two basic types of hybrid dual-fuel combined cycle arrangements are detailed in Chapter 4: steam/water side integrated HCC and windbox repowering.
With reference to WTE-GT steam/water side integration, the logic governing plants match in terms of steam production, as function of the thermal powers introduced, are investigated and explained. This thermodynamic analysis, carried out in Chapter 5, assesses and defines, for a given layout and operative conditions, the optimum WTE-GT plant match in terms of system input thermal powers, to maximize steam generation, power output and to minimize discharged outlet temperatures.
Moreover, several proposed WTE-GT steam/water side integrated layouts, with reference to one and two pressure level heat recovery boiler configuration, are presented and detailed in Chapters 5 and Chapters 6, respectively.
Instead, Chapter 7 focuses on hot and cold windbox integrated layouts where the gas turbine exhaust, with or without pre-cooling, is supplied to WTE boiler and used as preheated combustion air.

Part IV, Chapter 8, concludes the manuscript discussing open research issues to evaluate the efficiency of a WTE-GT integrated configuration. The difficulty in defining a performance index capable of quantify the efficiency of the integrated system, compared to separate generation, lies in assign the extra power generated as a consequence of systems integration. Several performance indexes, specifically developed to take into account a system receiving different sources as input and producing useful energy output, are defined and discussed.

Keywords:

Waste-To-Energy, Municipal Solid Waste, Gas Turbine, Thermodynamic Analysis, Thermodynamic Cycles, Parametric Analysis.

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Nomenclature

Abbreviation

BC	Bottomer Cycle
C	Compressor
CC	Combined Cycle
CHP	Combined Heat and Power
EU	European Union
GDP	Global Domestic Product
GT	Gas Turbine
HCC	Hybrid Combined Cycle
HRSG	Heat Recovery Steam Generator
HP	High Pressure
ICE	Internal Combustion Engine
LHV	Lower Heating Value
LP	Low Pressure
MS	Multi Source
MSW	Municipal Solid Waste
SI	Synergy Index
ST	Steam Turbine
T	Turbine
TC	Topper Cycle
WFD	Waste Framework Directive
WTE	Waste-To-Energy

Symbols

c	specific heat [kJ/kgK]
F	power input [MW]
g	acceleration of gravity [m/s^2]
h	specific enthalpy [kJ/kg]
H	total enthalpy [kJ/kg]
m	mass flow rate [kg/s]
p	pressure [bar]
Q	thermal power [MW]
S	entropy [kJ/kgK]
T	temperature [°C]
u	velocity [m/s]
U	useful energy output [MWh]
z	altitude [m]

Greek Symbols

ε	effectiveness
η	efficiency

Subscripts and Superscripts

I	first law
II	second law
boil	waste-to-energy boiler
ECO	ECOnomizer
el	electric
ev	evaporation
exh	exhausted
i	generic i-th input
lat	latent heat
max	maximum
NU	Non Useful heat
O	Outlet
ref	reference
rev	reversible process
s	steam
ss	single source
sc	sub-cooling
SH	SuperHeater
W	Waste

1. Waste Overview

In the recent years, the increased focus on energy resources and environment has changed waste perception. The growing scale of economy activities, standards of living and population have lead to a sharp increase in the quantity of waste generated. Waste poses a highly complex and heterogeneous environmental problem since each human activity, inevitably, results in generation of waste due to imperfect utilization of energy and resources.

There are several definitions of what exactly constitutes waste and many classifications which attempt to categorize waste flows. According to the 2006/12/EC Waste Framework Directive (WFD) [1] and amended by the new 2008/98/EC WFD [2], waste is defined as "*any substance or object the holder discards, intends to discard or is required to discard*". Waste can be categorized with respect to the source that generate it; according to this, Municipal Solid Waste (MSW) defines what is generated by households and contains the so called “rest waste” (what is left over after recycling process) as well as organic waste, glass, paper and other recyclable materials. Waste can be treated in several ways.

Directive 2008/98/EC also sets the basic concepts and definitions related to waste management and lays down waste management principles introducing the "waste hierarchy": waste prevention is still considered as the main goal followed by re-use and recycle; nevertheless, a growing an interest in Waste-To-Energy (WTE) conversion process must be pointed out.

The possibility to generate energy, in the form of heat or electricity, is becoming an attracting way to deal with waste. Various WTE technologies exist today, such as waste incineration, anaerobic digestion, gasification, etc; despite that, mass burn incineration is still the most common. The prominence of Waste-To-Energy differs widely from country to country across Europe. Denmark is one of the countries where waste incineration with energy recovery constitutes the largest part of the waste treatment options. Energy recovered from waste accounted in 2007 for around 20% of the Danish heat production and 4% of the electricity production [3].

In order to provide a general overview on municipal solid waste, the following paragraphs focus on MSW generation and disposal for both, the European and the Italian contest. With the aim to provide also a legislative focus on waste matter, a brief description on European Directives, with particular interest to the latest 2008 WFD, is carried out in the last paragraph.

1.1. *Overview on waste production and disposal for European countries*

The socio-economic growth of the most industrialized countries has involved a progressive increase of the municipal solid waste production. The MSW average production, in the EU-27, was over half a ton per person in 2009, hence issues related to the disposal of MSW turn out to be very important. Figure 1 shows the amount of MSW produced per person for the EU-27 countries [4]; it can be seen a very wide range of values: Italy, with a MSW production of about 540 kg/p.p./year, lies just above the EU-27 average value[1].

Among all the MSW processing technologies, landfill was the cheapest and simplest way to deal with the final treatment of waste. Nowadays, the increase in environmental and healthy concerns combined with the possibility to exploit these ordinary life by-products, as a valuable energy resource, has led to explore alternative methods for waste final disposal. Figure 2 highlights the EU-27 percentage allocation of MSW final treatments. Landfill is still the predominant treatment option for most of EU countries, only few exceptions (like the Netherlands, Denmark and Sweden) have a high level of alternatives for final treatments disposal. Incineration, as a waste management technique, varies greatly from country to country; percentage of waste incinerated ranges from zero (e.g. East countries) to about 50% (e.g. Denmark) with an average value, for 2009, equal to 20%. Germany, Austria, the Netherlands, Sweden, Denmark and Belgium dispose less than 10% of their municipal waste in landfill, on the contrary, nine states landifilling more than 80% of their total MSW production. Figure 3 shows, for the European Countries, the waste incineration, for 2009, in terms of kg/p.p year. In 2009, approximately 51.2 million tons of municipal waste were sent to incineration plants in the EU-27, 98% of which from EU-15 Member States.

Costs of waste disposal methods is variable depending on the technology adopted and on the country. As a rule of thumb, typically, incineration costs are twice the costs of landfill. A recent study [5] quantifies incineration costs between 100 €/ton and 250 €/ton of waste, whereas costs for landfill have a range down to 20 €/ton of waste.

[1] It is important to point out that as definitions and waste categories differ from one country to another, some of the values given may not be directly comparable.

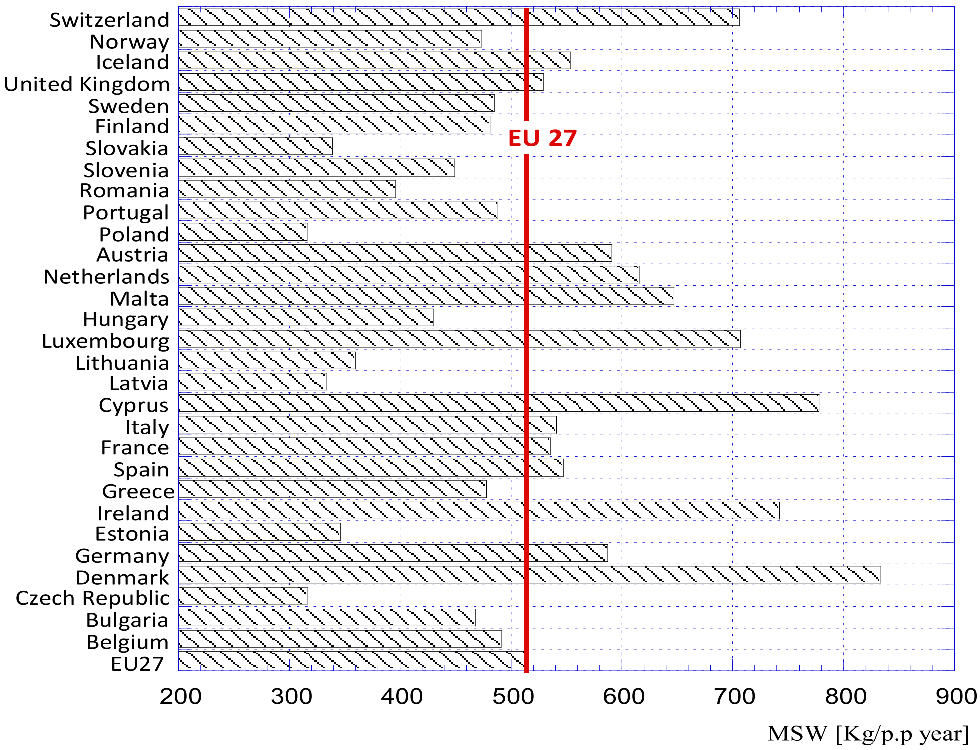

Figure 1 : MSW produced per person per year in the EU27 countries (2009) [4].

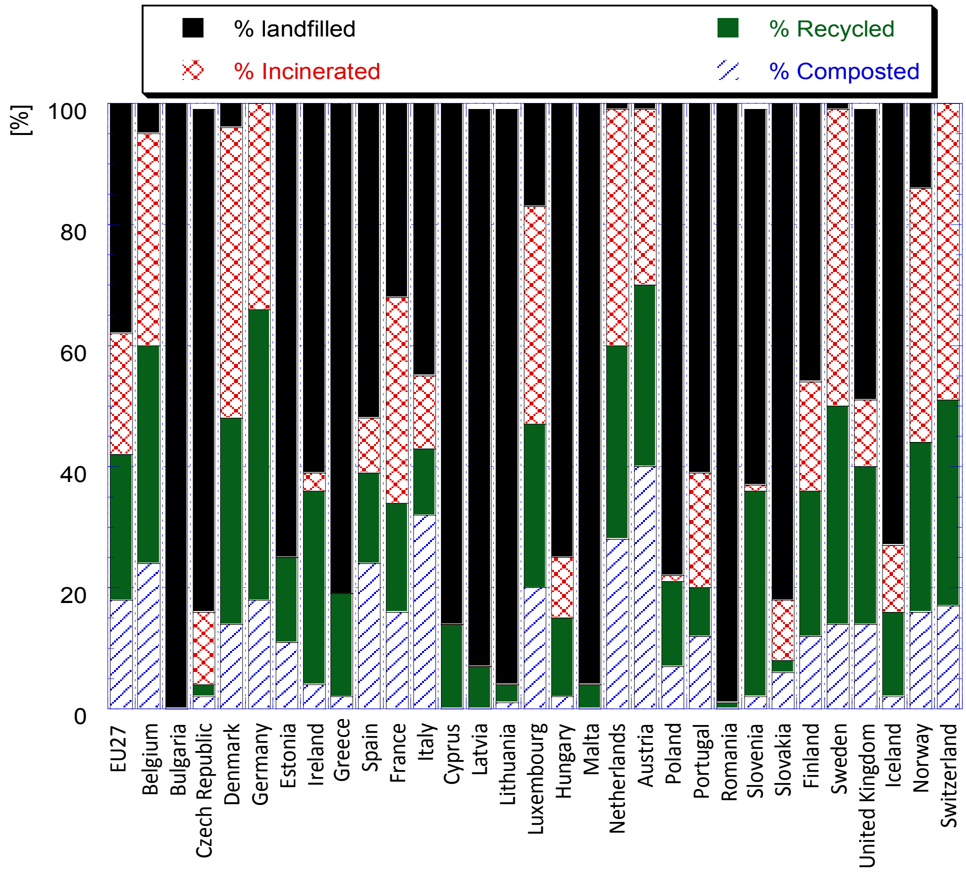

Figure 2 : Percentages of MSW final treatments in the EU 27 countries (2009) [4].

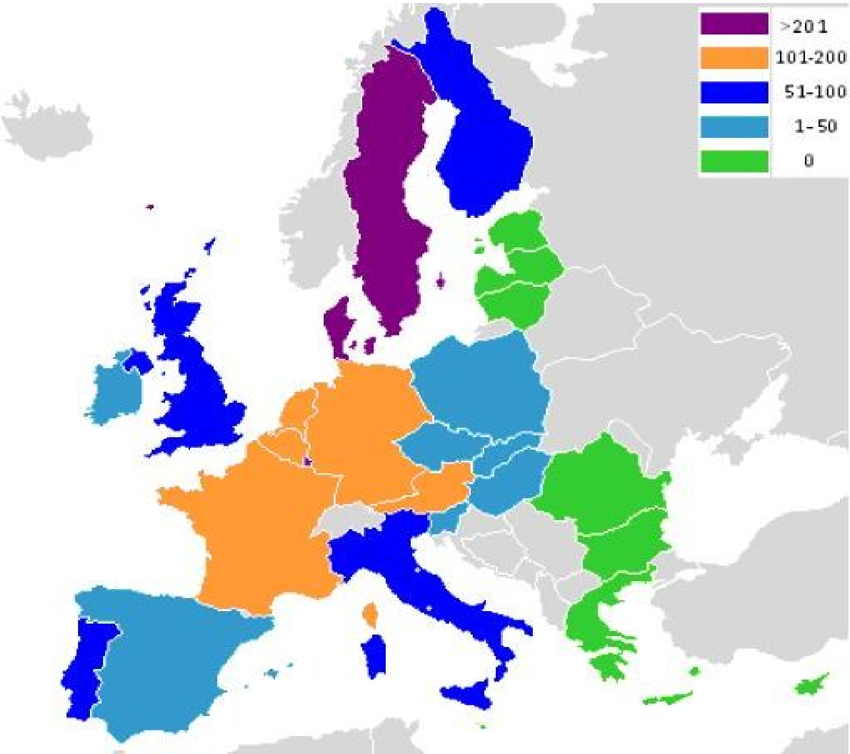

Figure 3 : kg of MSW incinerated per person in European countries, in 2009 [4].

At the end of 2005 (latest European available data), 388 incineration plants were operating in EU-15 countries with an average capacity equal to about 500 ton/day. The waste treatment capacity of each installation varies greatly across Europe. Table 1 reports ISWA's census data [6] including the number and the capacity (both, total and average) of the incineration plants for each European country. A recent study by CEWEP [7] forecasts a significant increasing trend for incineration with energy recovery in Europe: MSW treatment capacity of European plants will grow from 64 million tons in 2006 to over 100 million tons in 2020 (+59%), with a marked increase in the recovery of energy, both in the form of heat and electricity.

Table 1 : Number and capacity of MSW incinerators plant in the EU-15 (2005) [6].

Country	Number of incinerators	Total capacity [t/day]	Average capacity [t/day]
Austria	9	2184	243
Belgium	18	8808	489
Denmark	34	13848	407
Finland	1	192	192
France	127	45816	361
Germany	68	58680	863
Greece	-	-	-
Ireland	-	-	-
Italy	52	17088	329
Luxembourg	1	310	310
Netherlands	13	16080	1237
Portugal	3	4920	1640
Spain	10	9264	421
Sweden	30	5880	588
United Kingdom	22	12312	410
Tot EU15	**388**	**195˙382**	**504**

1.2. Overview on municipal solid waste production in Italy

Data concerning waste production in the Italian regional contest are reported in Table 2, from 2005 to 2009, and in Figure 4, from 1998 to 2009 [8] and [9]. Central regions, in 2009, have the highest per capita production of waste, equal to about 604 kg per person per year, while the value is lower for the South, which records 493 kg/p.p. year. Emilia Romagna (with 666 kg/p.p. y) has the highest waste production per habitant followed by Toscana (with 663 kg/ p.p. y).

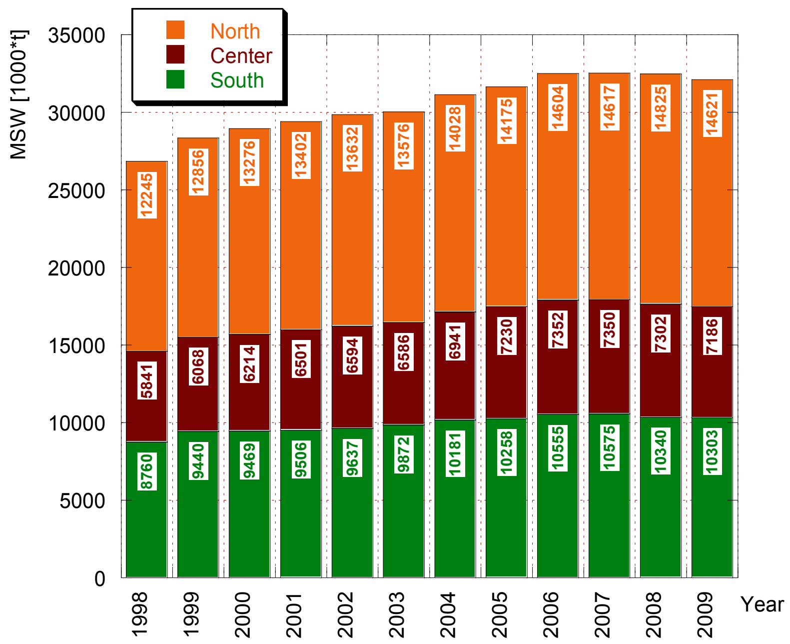

Figure 4 : Amount of MSW produced in Italy from 1998 to 2009 [8], [9].

The evolution of MSW production is linked to several factors, one of the most influencing is the Gross Domestic Product (GDP). The influence of economic conditions on the production of waste is presented in Figure 5 where GDP yearly percentage increaseis shown. Comparing Figure 4 and Figure 5, the decrease in waste generation between 2008 and 2009 is explained considering a decrease in GDP equal to -3.7 %.

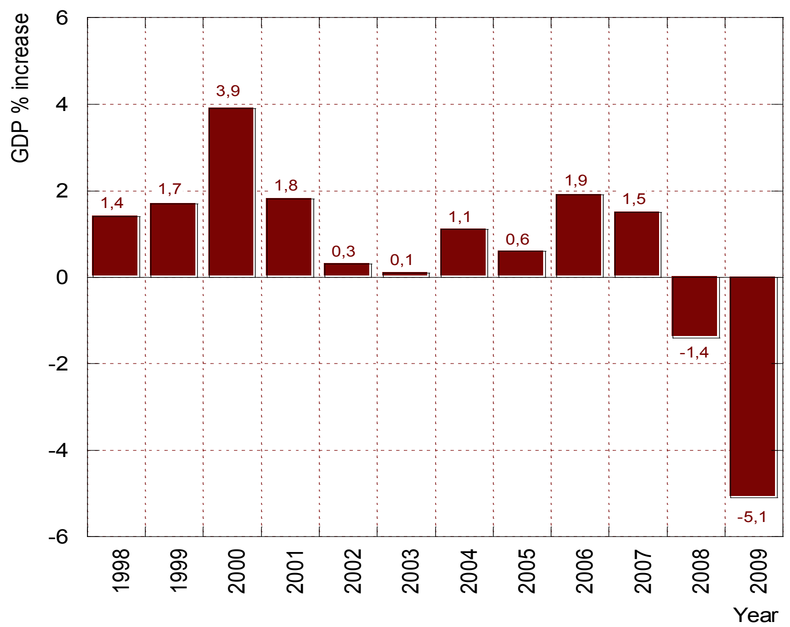

Figure 5 : GDP yearly percentage increase in Italy from 1998 to 2009 [10].

Table 2 – Amount of MSW produced per person in Italy from 2005 to 2009

Region	2005	2006	2007	2008	2009
	[kg/p.p*y]				
Piemonte	513	523	516	508	505
Valle d'Aosta	594	599	601	608	621
Lombardia	503	518	512	515	501
Trentino Alto Adige	485	495	486	496	501
Veneto	480	498	491	494	483
Friuli Venezia Giulia	498	494	506	497	479
Liguria	601	609	610	612	605
Emilia Romagna	666	677	673	680	666
North regions	**531**	**544**	**539**	**541**	**530**
Toscana	697	704	694	686	663
Umbria	641	647	639	613	590
Marche	573	565	564	551	537
Lazio	617	611	604	594	587
Center regions	**639**	**637**	**630**	**619**	**604**
Abruzzo	532	534	527	524	514
Molise	415	405	404	420	426
Campania	485	495	491	468	467
Puglia	486	517	527	523	527
Basilicata	385	401	414	386	382
Calabria	467	470	470	459	470
Sicilia	520	542	536	526	516
Sardegna	529	519	519	507	501
South regions	**494**	**509**	**508**	**496**	**493**
Italy	**539**	**550**	**546**	**541**	**532**

1.3. Overview on municipal solid waste disposal in Italy

Waste management means all activities related to the entire cycle of waste, from their production to their final destination (collection, transport, recovery and disposal). The municipal waste management takes into account composting facilities, anaerobic digestion, mechanical biological treatment, incineration and landfill. In 2009, landfills receives around 45% (Figure 6) of the total waste managed. Even in Italy, landfill is still the most common form of waste management, but not dominant, in fact, accounting the other strategies (recovery, treatment and disposal) the total percentage is higher (55%) than that of landfill [8].

Comparing regional data on waste production and disposal in 2009, Figure 6 highlights an inhomogeneous situation for Italy. Lombardia records the lower use of landfill as final waste disposal, representing 7% of its total waste production. In particular, northern regions, with the exception of Liguria, shows the lower use of landfill with respect to their total production. An exception, in the southern regions, is represented by Sardenia where regional legislation is driving towards recycle and reuse of waste.

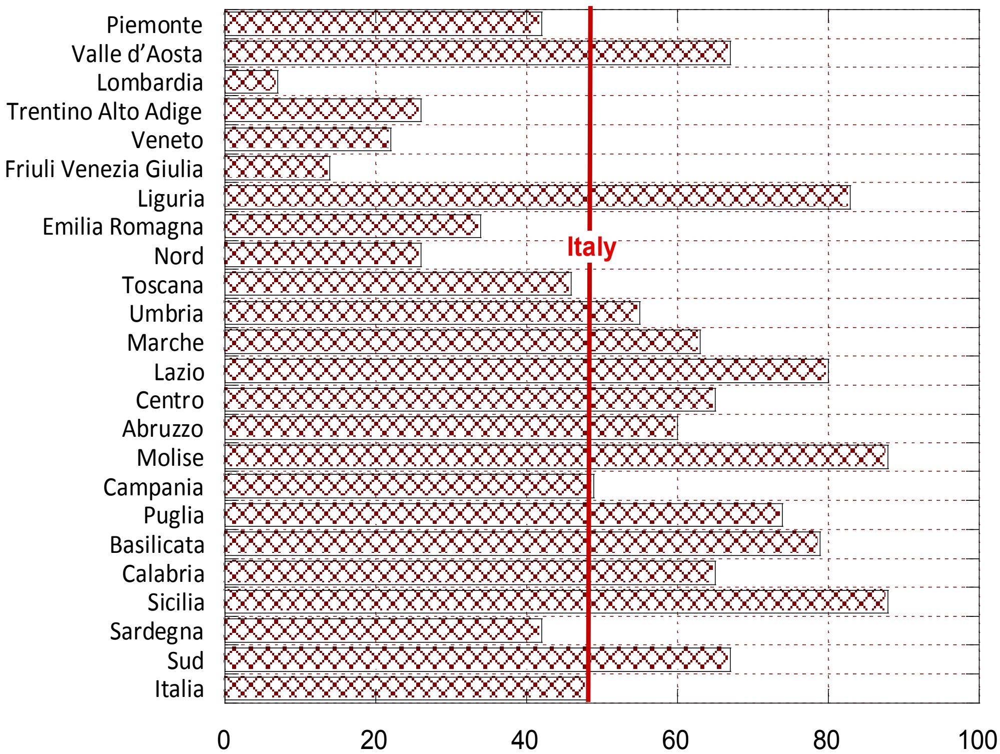

Figure 6 : Percentage of landifill disposal on the total waste production for the Italian regional contest, in 2009 [8].

1.4. Overview on waste legislation

The revised 2008/98/EC Directive sets the basic concepts and definitions related to waste management and lays down waste management principles such as the "waste hierarchy" [2]. The European Union's approach to waste management is based on three fundamental principles: waste prevention, recycling and reuse and improving final disposal and monitoring. Based on the European Union's approach to waste management, the best and most economical way of dealing with waste is to minimize its production; if waste cannot be prevented, as many of the materials as possible should be recovered, preferably by recycling. Where possible, waste that can not be recycled or reused should be safely incinerated, recovering the energy released with waste combustion, leaving landfill as the last option for waste disposal. Figure 1 schematically shows, through an inverted pyramid, the waste management's hierarchy suggested by European Commission's directives**.**

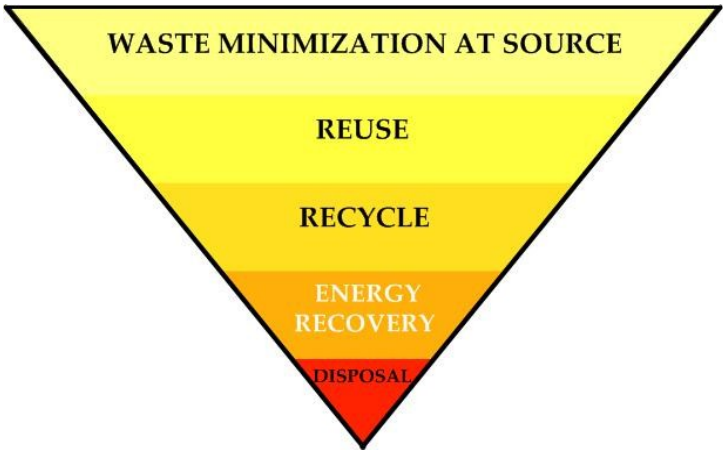

Figure 7 : Waste management's hierarchy as suggested by European Commission's directives.

Although thermal processing of waste is not the only option of waste disposal, in comparison with other processes (e.g. commonly used landfilling) it has a number of advantages, among which: speed of processing with respect to landfill disposal, possibility to process even extremely hazardous waste (e.g. hospital waste, etc.), possibility to adjust clean gas as well as solid residues outlet and possibility to effectively utilize heat released by the waste combustion, commonly known as Waste-To-Energy (WTE) [11].

WTE is referred to a thermal processing of waste including energy utilization. It means not only combustion of various types of waste (incineration) leading to a substantial reduction of their volume but besides that, WTE systems can provide clean and reliable energy in the form of heat as well as electric power (or both in the form of combined heat and power, CHP). The importance of Waste-To-Energy differs widely from country to country in Europe: differences are mainly due to countries environmental and energy policies combined with economic aspects. In some countries, waste are regarded as a renewable source, so energy [12] from waste is evaluated in the same way as biomass energy, which has strong support within the EU.

The waste incineration sector has been subject to legislative requirements regional, national and European level for many years.

The following guidelines for waste incineration plants have marked the growth of attention by the European Commission on production and waste management:

- 89/369/EEC on new waste incinerators [13];
- 89/429/EEC on existing MSW incinerators [14];
- 94/67/EC on the incineration of hazardous waste [15];
- 2000/76/EC on pollutant emissions on incineration of waste [16];
  2006/12/EC on waste disposal [17];
- 2008/98/EC on waste management and WTE efficiency definition [2].

Directives [13], [14] and [15] has been replaced, as from 28 December 2005, with Directive 2000/76/EC [16].

The aim of 2000 Directive [16], is to limit and avoid negative effects of incineration and waste co-incineration on the environment, focusing in particular on emissions into the atmosphere, in soil and water, but also to reduce the risks for human health. The objective is achieved by requiring stringent operational conditions and establishing rigorous emission limit values.

It defines "incineration plant" a fixed unit, or technical equipment, or equipment dedicated to thermal treatment of wastes with or without recovery of heat produced by combustion. The definition covers the site and the entire incineration plant, all incineration lines, waste reception and storage, facilities for the pretreatment, waste supply systems, facilities for the treatment of exhaust gas, facilities for on-site treatment or storage of waste residues, stack devices and systems for controlling, recording and monitoring incineration operations.

It defines "co-incineration plant" any stationary or mobile plant whose main purpose is the production of energy or material products and which uses waste, as a regular or additional fuel, or where waste is subjected to heat treatment for disposal. Similarly, the definition includes the incineration plant site and the set of all components constituting the plant.

Incineration and co-incineration plants are designed, constructed, equipped and operated to prevent emissions to cause significant pollution of the soil. The exhaust gases are also discharged in a controlled manner using a fireplace whose height affects the air quality and is likely to safeguard human health and the environment.

The 2006 legislation [1], redefines concepts such as the notions of waste, recovery and disposal and establishes permit requirements and registration for the institution or company that performs the management of waste. Subsequent Directive stated again the definition of waste, recovery and disposal and reinforced measures to prevent waste, by introducing a new approach that considers the entire lifecycle of products and materials, reducing the environmental impacts of production and waste management.

It is considered "waste" any substance or object which the holder discards or intends or is required to discard. If waste has one or more hazardous characteristics, that is explosive, oxidizing, highly flammable, irritant, harmful, toxic or carcinogenic is defined as "hazardous waste". While "organic waste" means waste biodegradable garden and park waste, food products from food processing plants, restaurants or private households. Organic waste is collected separately at the end of composting.

It defines "waste producer" anyone whose activities produce waste or make pre-processing, mixing or other operations that change the nature or composition of the waste. "Waste holder" is the manufacturer or the person who has legal hold; the "dealer" is the firm that acts as the buyer to buy and sell waste; "broker" is the company that deals with the recovery or disposal of waste on behalf of others, without take physical possession of the waste.

So the "waste management" includes the collection, transport, recovery and disposal of waste, the supervision of such operations and after-care of disposal sites.

The waste management plans shall contain at least the following elements:

a) the type, quantity and source of waste generated within the territory, the waste to be shipped to or from the country and an evaluation of future waste streams;

b) waste collection systems and major disposal and recovery installations, including any special arrangements for waste oils, hazardous waste or waste streams covered by specific community legislation;

c) an assessment of the need for new collection systems, the closure of existing waste facilities, additional infrastructure facilities;

d) sufficient information on the location criteria for site identification and the ability of future disposal or major recovery installations, if necessary;

e) general policies for waste management, including technologies and methods of planned waste management, or policies for waste posing specific management problems.

1.4.1. Overview on waste framework directive 2008/98/EC

The new Directive 2008/98/EC in addition to sets the basic concepts and definitions related to waste management, such as definitions of waste, recycling and recovery, it promotes the production of energy from waste with the so-called "R1" formula [2]:

$$R1 = \frac{Q_{prod} - (E_f + I_{imp})}{f_B * (E_f + E_w)} \qquad (1)$$

where:

- Q_{prod} means annual energy produced as heat or electricity (GJ/year). It is calculated with energy in the form of electricity being multiply by 2.6 and heat produced for commercial use multiplied by 1.1. Multiplicative factors represent, respectively, average efficiency for the electrical and thermal generation. For the electric generation the value of 0.385 has been chosen according to the average efficiency of coal fired power plants, while 0.91, for thermal generation, has been assumed according to average efficiency of boilers.

Thus, Q_{prod} can be expressed as follows:

$$Q_{prod} = 2.6 * Q_{prod,el} + 1.1 * Q_{prod,th} = \frac{Q_{prod,el}}{0.385} + \frac{Q_{prod,th}}{0.91} \qquad (2)$$

So, assuming an average of 38.5% for the electrical conversion efficiency and 91% for external heat generation, Q_{prod} can be regarded as an "equivalent primary energy": the energy input, with conventional fossil fuels, to generate the same amount of electricity and heat as the WTE power plant. It is important to understand that equivalence values are not exact coefficients or conversion factors. They provide an estimation of the energy that is required to produce the same amount of energy output externally.

- E_f means the annual energy input to the system form fuels contributing to the production of steam (GJ/year) (e.g. natural gas used, if necessary, as an additional fuel to increase post combustion temperature).
- E_w means annual energy contained in the treated waste, calculated using the net calorific value of the waste (GJ/year).
- I_{imp} means annual energy imported, excluding E_f and E_w (GJ/year).

- f_B is a factor accounting for energy losses due to bottom ash and radiation (assumed equal to 0.97).

Although the energy efficiency criteria for the recovery status of a WTE plant can be regarded just as a starting point for all the Member States, it constitute a reference to create quality standards for waste recovery. According to the achieved R1 value, WTE power plants can be classified as energy recovery operations if their energy efficiency (R1) is equal or above:

- 0.60 for installations in operation and permitted, in accordance with applicable Community legislation, before 1 January 2009,
- 0.65 for installations permitted after 31 December 2008,

The Directive allows municipal waste incinerators to be classified as “recovery operations”, provided they contribute to the generation of energy with high efficiency. The aim of the Directive is to promote the use of waste for energy generation in efficient municipal waste incinerators and to encourage innovation in waste incineration. In this context, it is important to note that “recovery” means any operation the principal result of which is waste serving a useful purpose by replacing other materials.

A number of criteria to compare the effectiveness of energy recovery and utilization in incineration plants have been proposed (as shown in Table 3) thus, the so called “R1” formula has been derived from previous and similar definition of energy efficiency for a WTE power plant. Their common feature can be seen in an effort to describe the relation between energy outputs (produced or exported energy), on one side, and energy demand, on the other. Such evaluation of energy streams in the unit represents one of the most important steps in the assessment of the efficiency of energy utilization. The main energy streams for a typical WTE power plant are shown in Figure 8.

According to The Confederation of European Waste-to-Energy Plants (CEWEP) [19] and [7], the value of the weighted average of the criterion called “plant efficiency factor”, P_{el}, as defined in Table 3, was 4.08 (the minimum was 0.04 and the maximum value was 21.08). According to CEWEP’s definition, if P_{el} is greater than 1, the plant can produce (and export) more energy than is needed and thus, part of the recovered energy can be utilized. Thus, plants with values of the criterion greater than 1 are rated as ‘Waste-To-Energy’ units.

Table 3 : Comparison of criteria for assessment of incineration plants.

Proposed by	Criterion name	Definition
The Confederation of European Waste-to-Energy Plants (CEWEP).	Plant efficiency factor	$P_{el} = \dfrac{Q_{prod} - (E_f + I_{imp})}{(E_f + I_{imp} + I_{circ})} \geq 1$
	Energy utilization rate	$\eta = \dfrac{Q_{prod} - (E_f + I_{imp})}{f_B * (E_w + E_f)} \geq 0.5$
Reference Document on Best Available Techniques for Waste Incineration (BREF).	Plant efficiency	$P_{el} = \dfrac{Q_{exp} - (E_f + I_{imp})}{(E_f + I_{imp} + I_{circ})}$

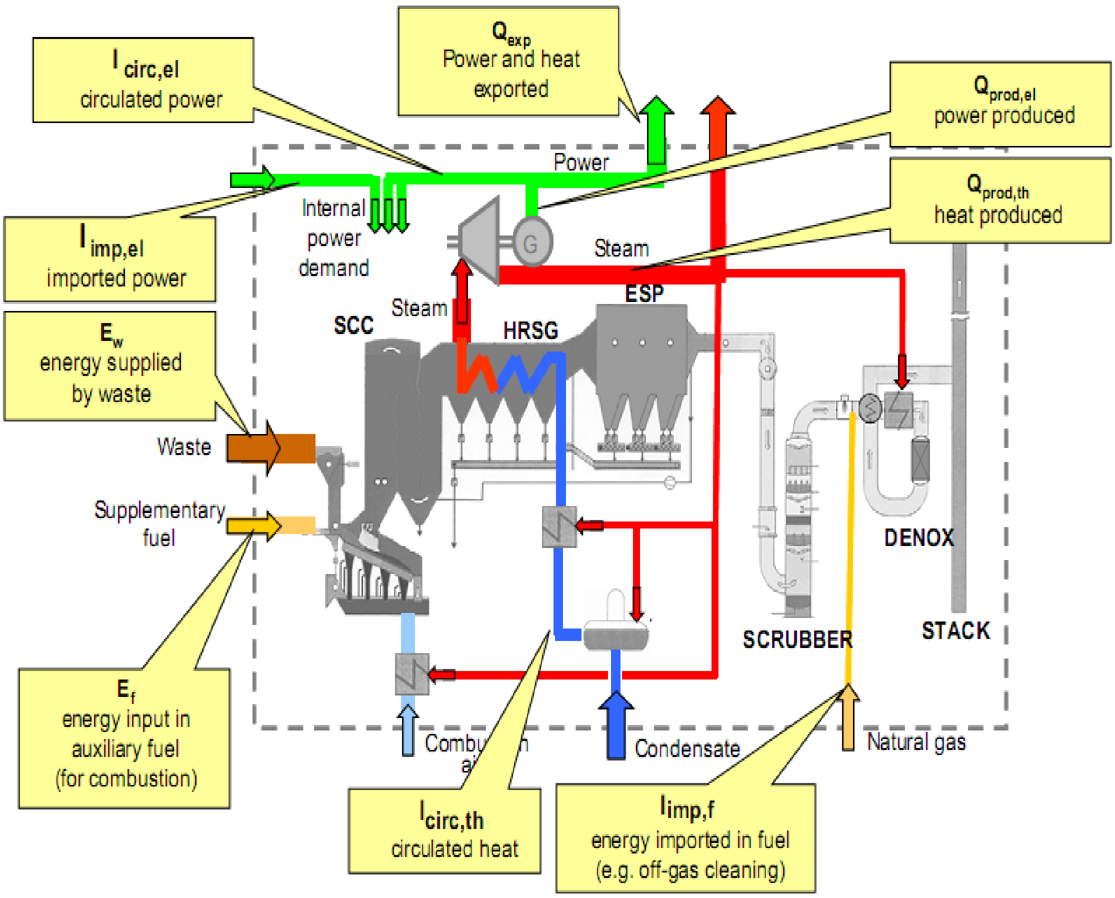

Figure 8 : Main energy streams in a municipal solid waste incinerator [20].

The energy recovery status provides incentives for future investments in WTE plants located close to energy customers. The aim of 2008 Directive is to drive WTE facilities to maximise energy recovery and utilisation of waste heat, in order to substitute conventional energy production plants fired by fossil fuels.
Thus, new possibility to maximize the conversion of municipal solid waste into energy are gaining growing interest. According to a recent CEWEP study [7], taking the reference year 2006, the total amount of WTE plants in Europe (EU27+ Switzerland + Norway) was 420. Of the total, 252 (60%) in Europe are below the R1 threshold or did not participate. As a results of this study, only 40% of total plants are proven to reach R1 status. One of the most efficient waste management system in Europe is the one of Denmark where, in the last 10 years, all large and medium sized WTE facilities have been converted to CHP production for district heating, making the heat from WTE the cheapest source of heating in Denmark [3].
In this context, the energy conversion of MSW through incineration in WTE power plant is one of the principal means of an integrated management; its potential is increasing throughout Europe, both in terms of plants number and capacity, furthered by legislative directives.

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2. Waste-To-Energy

The first waste incineration plant, known as "Destructor" was built in Nottingham, UK, in 1874. Waste incineration became established in many European countries at the end of 19th century, as a device to minimize waste's volume and to make it hygienic. Since then, as environmental awareness has grown, Waste-To-Energy technology has continuously developed and improved, increasing the importance of efficient energy generation [1]. Waste-To-Energy plants, in many European countries, become an important tools to produce heat and electricity replacing the energy produced by conventional power plants, burning fossil fuels. A study conducted by CEWEP, in 2008, on Waste-To-Energy potential across Europe highlights that 28 billion kWh of electricity and 69 billion kWh of heat could be generated if 69 million tonnes of household and similar waste, remaining after prevention, reuse and recycling, were treated in operative WTE power plants. Then between 7 to 38 million tonnes of fossil fuels (gas, oil, hard coal and lignite) could be substituted annually, emitting 19 – 37 million tonnes of CO_2. Replacing these fossil fuels, Waste-to-Energy plants could supply annually about 13 million inhabitants with electricity and 12 million inhabitants with heat [2].

The dominant technology for energy recovery from MSW is direct combustion over a moving grate with the generation of superheated steam feeding a steam turbine in a Hirn cycle. The amount of energy recovered from the MSW combustion can vary significantly with the characteristics of MSW fed into the boiler (composition, mass flow rate and lower heating value, LHV), the combustion technology, the configuration and features of the recovery boiler (adiabatic or integrated) and the characteristics of the thermodynamic steam cycle. Due to the heterogeneous nature of waste, some differences with respect to a conventional power plant burning fossil fuels have to be considered in the chemical-to-electrical energy conversion process. The incineration sector has undergone a rapid technological development over the last 10 to 15 years. Much of this change has been driven by legislation specific to the industry and this has, in particular, reduced emissions to air from individual installations; the sector is now developing techniques which limit costs, whilst maintaining or improving environmental performance thus, WTE power plants are growing and improving all around the world.

2.1. WTE overview in Europe

Information concerning the total amount of MSW thermally treated with incineration (expressed in millions of tonnes per year), the number of WTE power plants and the annual amount of electricity and heat (expressed in millions of MWh) generated by WTE, for some of the European country, are reported in Table 1. Data have been collected and elaborated starting from Country Report on Waste Management of each CEWEP member [1] and [3].

A 2009 CEWEP report [4] highlights the relevance and the importance of the Waste-To-Energy power plants to contribute to environment safety and protection and, at the same time, to ensure the necessary energy supply and primary fuels diversification. Figure 1 shows growth of Renewable Energy from WTE for the EU-27, in TeraWatt-hour, including a forecast of energy potential in 2020.

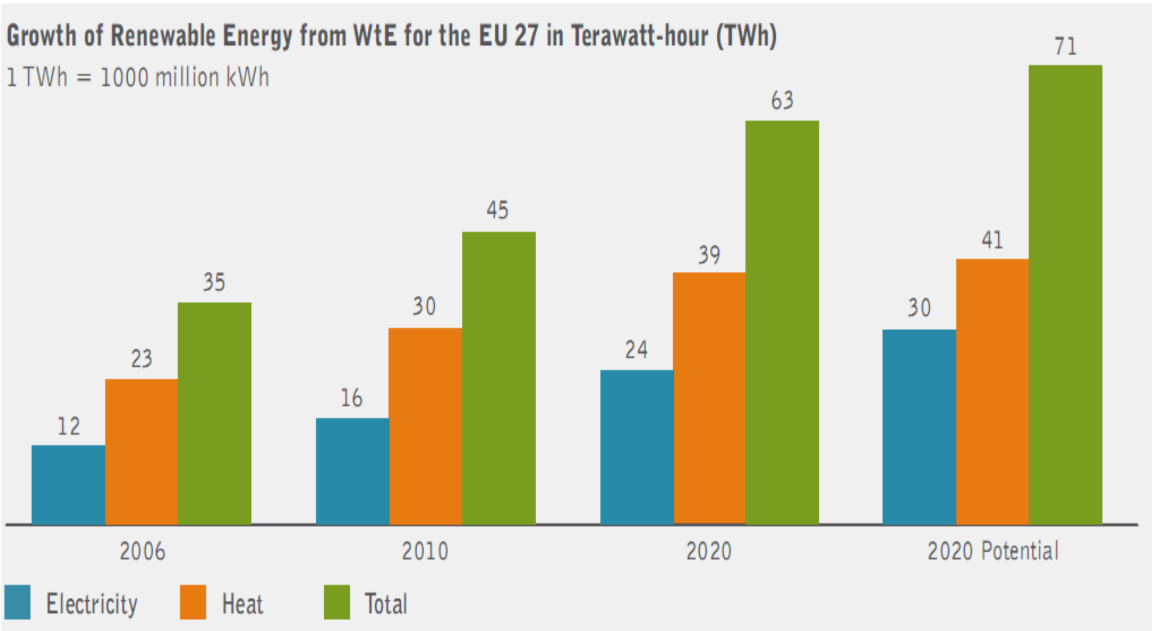


Figure 1 : Percentage growth of renewable energy from WTE for EU27 in TeraWatt-hour [4].

Table 1 : Data on WTE, from annual Country reports, 2010 [1] and [3].

Country	**Waste thermally treated [Millions of tones]**	**Number of incineration plants [-]**	**Generated Electricity [Millions MWh/y]**	**Generated Heat [Millions MWh/y]**	**Exported Electricity [Millions MWh/y]**	**Exported Heat[1] [Millions MWh/y]**	**Electricity produced with waste [%]**
Belgium	2.800	16	1.400	1.240	1.110	0.510	1.69
Denmark	3.590	29	1.866	7.034	1.586	6.331	5.04
Finland	0.179	2	0.017	-	0.003	0.335	0.02
France	13.000	132	3.489	6.573	2.767	6.155	0.61
Germany	19.066	69	7.666	-	5.724	14.160	1.29
Italy	4.600	49	3.100	0.900	-	-	1.06
Norway	1.091	20	0.105	2.806	0.105	1.873	0.07
Netherlands	6.000	11	2.907	-	2.326	-	2.76
Portugal	1.100	3	0.584	-	0.479	-	1.30
Hungary	0.101	1	0.173	0.757	0.143	0.144	0.46
Sweden	4.500	29	1.650	12.300	-	-	1.15
Switzerland	3.611	29	1.833	3.241	-	-	2.84

[1] Included “heating”, “cooling” and”steam”.

2.2. Basics of a WTE power plant

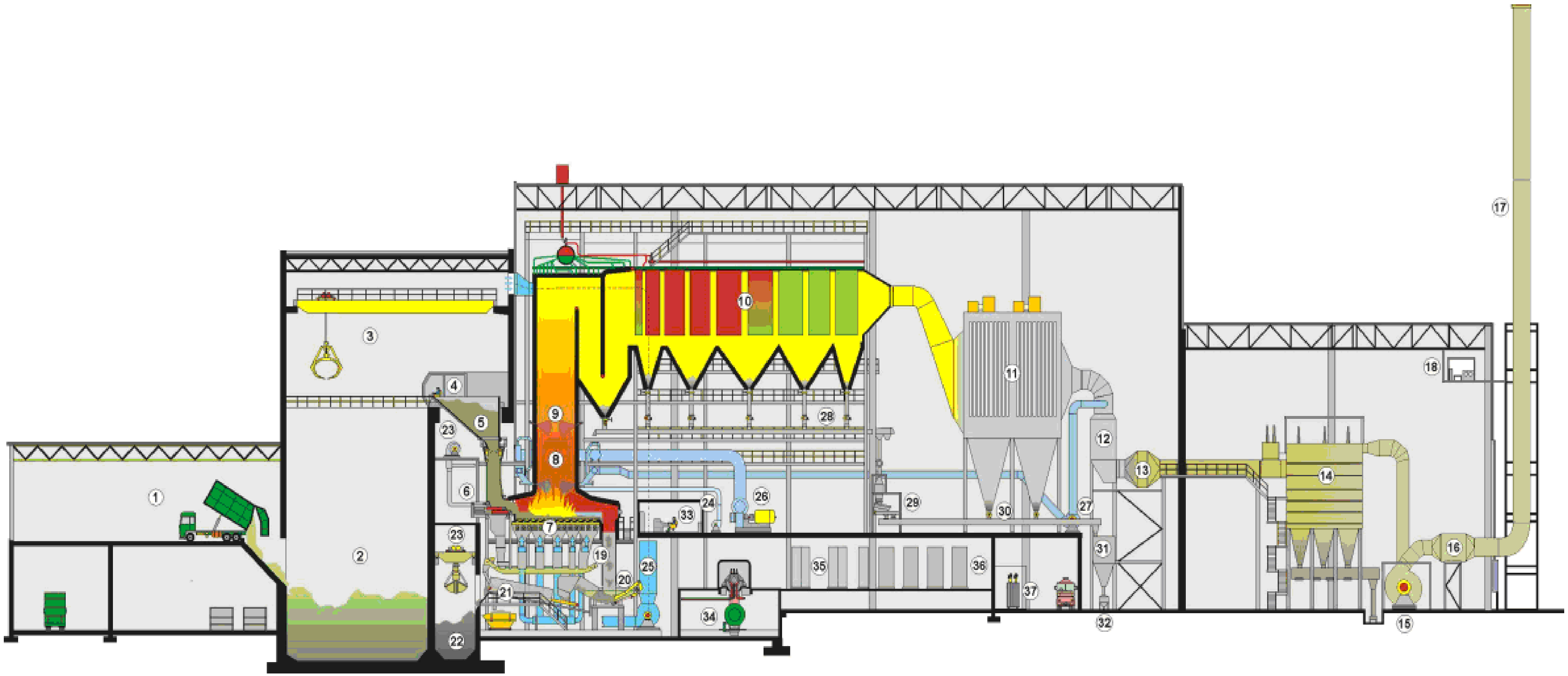
From a general point of view, a WTE power plant is schematically shown in Figure 2 and may include the following operations and sections:

• incoming waste reception;
• storage of waste and raw materials;
• pre-treatment of waste (where required, on-site or off-site);
• loading of waste into the process;
• thermal treatment of the waste;
• energy recovery (e.g. boiler) and conversion;
• flue-gas cleaning;
• flue-gas cleaning residue management (from flue-gas treatment);
• flue-gas discharge;
• emissions monitoring and control;
• waste water control and treatment (e.g. from site drainage, flue-gas treatment, storage);
• ash/bottom ash management and treatment (arising from the combustion stage);
• solid residue discharge/disposal.

Table 2 schematically shows the purpose of the main components of a WTE power plant.

Table 2 : purpose of the main components of a waste incineration plant [5].

OBJECTIVE	RESPONSIBILITY OF
– Destruction of organic substances – Evaporation of water – Evaporation of volatile heavy metals and inorganic salts – Production of potentially exploitable slag – Volume reduction of residues	Furnace
– Recovery of useful energy	Energy recovery system
– Removal and concentration of volatile heavy metals and inorganic matter into solid residues – Minimizing emission to all media	Flue gas cleaning



Technical data

Number of lines:	3
Wast capacity per line:	20 t/h
Thermal capacity per line:	61.5 MW
Steam output per line:	75 t/h
Steam pressure:	52 bar
Steam temperature:	440 °C

1 Tipping hall
2 Waste bunker
3 Waste crane
4 Crane operator cabin
5 Feed hopper
6 Feeder
7 MARTIN horizontal grate
8 Furnace
9 NO_x reduction (SNCR)
10 Steam boiler
11 Electrostatic precipitator
12 Economizer
13 Heat exchanger
14 Reactor with fabric filter
15 ID fan
16 Silencer
17 Stack
18 Measuring station
19 Grate siftings conveyor
20 Ram-type discharger
21 Bottom ash transport
22 Bottom ash bunker
23 Bottom ash crane
24 Cooling air system
25 Underfire air system with air preheater
26 Overfire air system with air preheater
27 Flue gas recirculation
28 Fly ash transport (boiler)
29 Crusher / screening plant
30 Fly ash transport (ESP)
31 Fly ash silo
32 Pneumatic fly ash transport
33 Control room
34 Turbine / generator
35 LV switch gear
36 MV switch gear
37 Transformer

Figure 2 : Typical new generation WTE power plant (AMSA, Milano)[6].

2.2.1. *Waste delivery and storage*

The waste delivery area is where the delivery trucks, trains or containers arrive in order to dump the waste into the bunker, usually after visual control and weighing. Figure 3 shows a typical municipal solid waste bunker device. Enclosure of the delivery area can be one effective means to avoid odour, noise and emission problems from the waste. The bunker is usually a waterproof, concrete bed. The waste is piled and mixed in the bunker using cranes equipped with grapples. The mixing of wastes helps to achieve a balanced heat value, size, structure, composition, etc. of the material dumped into the incinerator filling hoppers. The cab has its own ventilation system, independent from the bunker.

In order to avoid excessive dust development and gas formation (e.g. methane) from fermenting processes, as well as the accumulation of odour and dust emissions, the primary incineration air for the furnace plants is often extracted from the bunker area. Depending on the calorific value of the waste as well as the layout and the concept of the plant, preference is most often given to supplying the bunker air to either the primary or secondary air.

The bunker usually has a storage capacity of several days (commonly 3 - 5 days) of plan operational throughput, thus its depth can reach a few dozen meters. This is very dependent on local factors and on the specific nature of the waste.

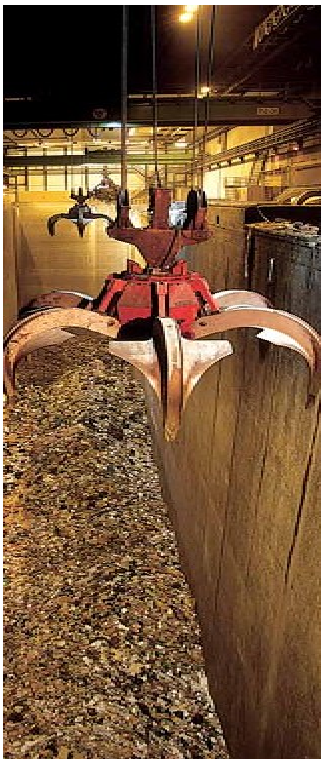


Figure 3 : A picture of a typical municipal solid waste bunker device.

Feeding means dosing the right quantity of fuel to the grate for steady combustion and energy production. Proper feeding is continuous and adjusted to the grate transport capacity to ensure an even fuel layer across the grate. This consistent feeding ensures minimal environmental impact because it promotes ideal and controllable combustion. The grate is fed at a variable rate adjusted to the energy production by means of a hydraulic pusher results in a steady and continuous feeding of the grate.

2.2.2. Basics of the combustion process

Basically, waste incineration is the oxidation of the combustible materials contained in the waste. Waste is generally a highly heterogeneous material, consisting essentially of organic substances, minerals, metals and water. During incineration, flue-gases are created containing the majority of the available fuel energy as heat. Burning of organic fuel substances occurs ones they have reached the necessary ignition temperature and came into contact with oxygen. The combustion process takes place in the gas phase, in fractions of seconds, and simultaneously releases energy where the calorific value of the waste and oxygen supply is sufficient. This can lead to a thermal chain reaction and self-supporting combustion, i.e. there is no need for the addition of other fuels.
The main stages of an incineration process, schematically shown in Figure 4, are [5]:

1) drying and degassing – here, volatile content is evolved (e.g. hydrocarbons and water) at temperatures generally between 100 and 300 °C. The drying and degassing process do not require any oxidising agent and are only dependent on the supplied heat;

2) pyrolysis and gasification - pyrolysis is the further decomposition of organic substances in the absence of an oxidising agent at approximately 250 °C – 700 °C. Gasification of the carbonaceous residues is the reaction of the residues with water vapour and CO_2 at temperatures, typically between 500 °C and 1000 °C. Thus, solid organic matter is transferred to the gaseous phase. In addition to the temperature, water, steam and oxygen support this reaction;

3) oxidation - the combustible gases created in the previous stages are oxidised, depending on the selected incineration method, at flue gas temperatures generally between 800 °C and 1450 °C.

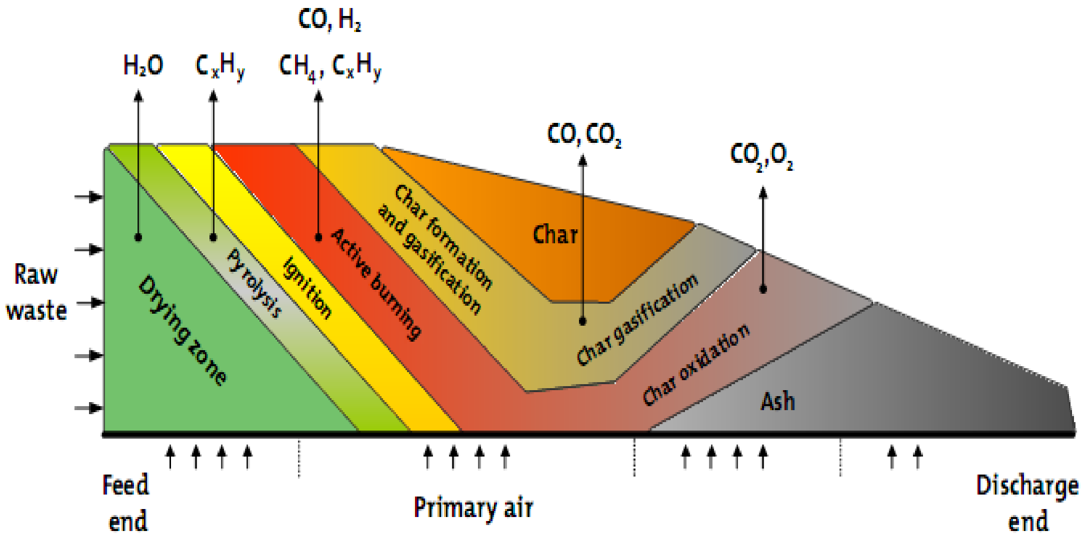


Figure 4 : schematic of process on the grate of a combustion chamber burning waste [7].

Thermal waste treatment is one of the most complex combustion processes. The process is mainly controlled by mass and heat transfer. In the illustration of Figure 4, a large part of the grate length has a deficit of oxygen (fuel rich condition) resulting in formation of combustible gases. The energy of the waste is released partly in the fuel layer and partly in the furnace room as combustible gases. Although the thermal treatment process of waste on the grate receives a certain amount of excess combustion air, gasification will take place in certain areas. Consequently, controlling the injection of primary air enable distribution of the individual reaction zones to obtain optimal combustion.

Staged combustion can be provided in several ways. One method for staging the combustion is a stepwise addition of combustion air to prevent complete combustion at the first stage. The changed in stechiometric rate will result in an increased flow of unburned gases into the furnace room. The energy released from the combustion process will move from the fuel bed to the furnace room, and a large part of the furnace room will be operating under slightly fuel-rich conditions.

The individual stages generally overlap, meaning that spatial and temporal separation of these stages during waste incineration may only be possible to a limited extent. Indeed the processes partly occur in parallel and influence each other. Nevertheless it is possible, using in-furnace technical measures, to influence these processes so as to reduce polluting emissions. Such measures include furnace design, air distribution and control engineering. In fully oxidative incineration the main constituents of the flue-gas are: water vapour, nitrogen, carbon dioxide and oxygen. Depending on the composition of the material incinerated and on the operating conditions, smaller amounts of CO, HCl, HF, HBr, HI, NO_X SO_2, VOCs and heavy metal compounds (among others) are formed or remain. Depending on the combustion temperatures during the main stages of incineration, volatile heavy metals and inorganic compounds (e.g. salts) are totally or partly evaporated. These substances are transferred from the input waste to both the flue-gas and the fly ash it contains. A mineral residue fly ash (dust) and heavier solid ash (bottom ash) are created. In municipal waste incinerators, bottom ash is approximately 10% by volume and approximately 20% to 30% by weight of the solid waste input. Fly ash quantities are much lower, generally only few per cent of input. The proportions of solid residue vary greatly according to the waste type and detailed process design.

For effective oxidative combustion, a sufficient oxygen supply is essential. The air ratio of the supplied incineration air to the chemically required (or stechiometric) incineration air, usually ranges from 1.2 to 2.5, depending on whether the waste composition and furnace system.

The incineration air accomplished the following objectives:

• provides the oxidant;
• cooling;
• avoidance of slag formation in the furnace;
• mixing of flue-gas.

Air is added at various places in the combustion chamber. It is usually described as primary and secondary, although tertiary air, and re-circulated flue-gases are also used.

The primary air is generally taken from the waste bunker. Primary air is blown by fans into the areas below the grate, where its distribution can be closely controlled using multiple wind boxes and distribution valves. The air can be preheated if the value of the waste degenerates to such a degree that it becomes necessary to pre-dry the waste. The primary air will be forced through the grate layer into the fuel bed. It cools the grate bar and carries oxygen into the incineration bed.

Secondary air is blown into the incineration chamber at high speeds via, for example, injection lances or from internal structures. This is carried out to assure complete incineration and it is also responsible for the intensive mixing of flue-gases, and prevention of the free passage of unburned gas streams.

Preheating the combustion air is particularly beneficial for assisting the combustion of high moisture content wastes. The pre-warmed air supply dries the waste, thus facilitating its ignition. The supply heat can be taken from the combustion of the waste by means of heat exchange systems. Preheating of primary combustion air can have a positive influence on overall energy efficiency in case of electricity production.

Figure 5 shows a typical 'capacity diagram' for a waste incineration plant where the operating area is highlighted as a function of waste mass flow rate, for different waste Lower Heating Value (LHV). The couple of variables, waste mass flow rate and LHV, indentify on vertical axis, the thermal input in the system. The diagram highlights also different area of operation where air preheating is required or auxiliary fuel is necessary.

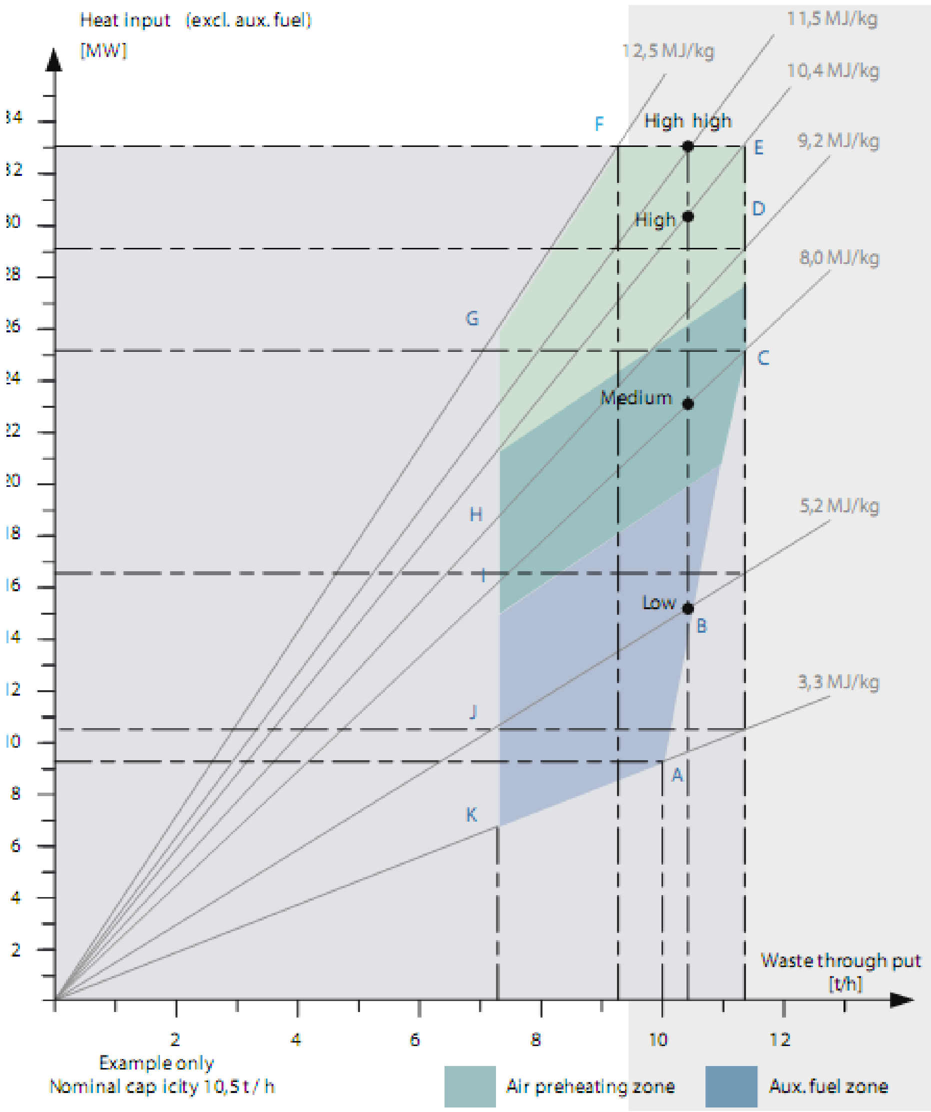


Figure 5 : A typical capacity diagram for an waste incineration system [7].

In Figure 6 a similar capacity diagram is reported showing the design and off-design operating points; the field of operation for a WTE power plants is defined by the area between these points of operation.

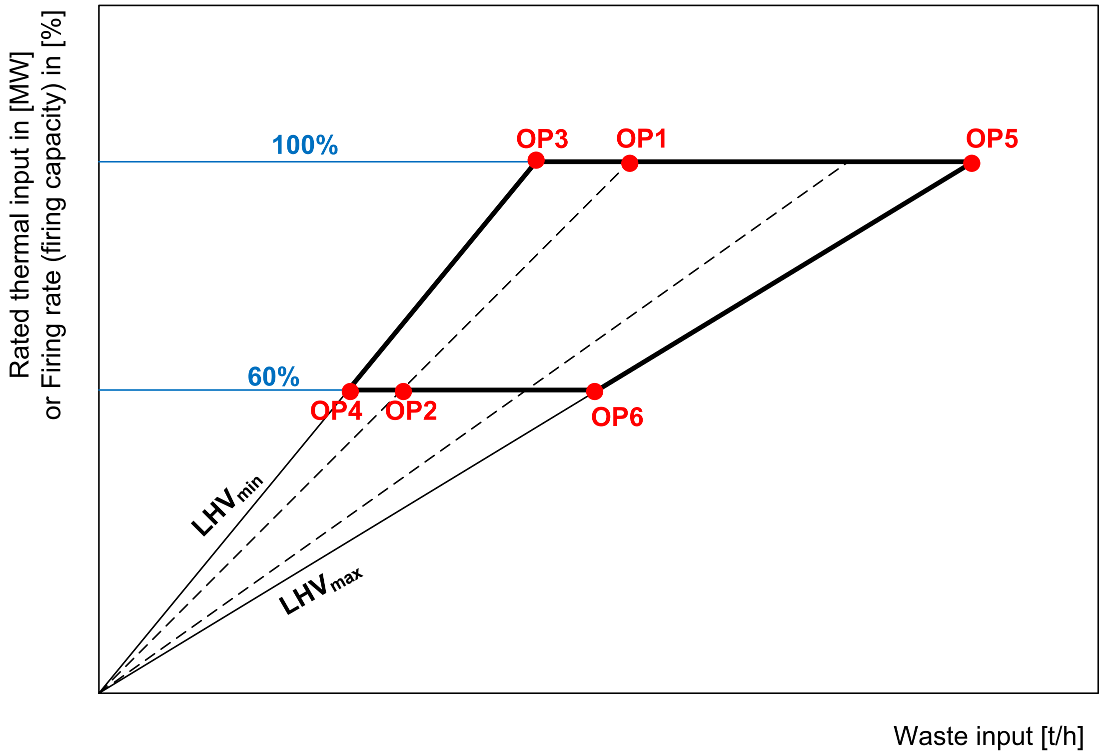


Figure 6 :Typical furnace capacity diagram of a waste incineration plant showing the design and off-design operating point.

Where:

- **OP1:** Design Point; Maximum firing rate (100%), Design waste input (average/typical LHV, average/typical waste throughput);
- **OP2:** Design Waste Input (average/typical LHV, average/typical waste throughput) at minimum boiler load (e.g. 60% capacity);
- **OP3:** Maximum boiler load with highest LHV (means little waste throughput);
- **OP4:** Minimum boiler load with highest LHV (means minimum waste throughput);
- **OP5:** Maximum boiler load with minimum LHV (means maximum waste throughput);
- **OP6:** Minimum boiler load with minimum LHV.

2.2.3. Combustion grates

A combustion grate is a transport device that moves the burning fuel from the inlet through the furnace to the bottom ash outlet. During transportation waste is mixed, and combustion air is added. Volatile material is released to the furnace and fixed carbon is burned on the grate. The grate is an integrated part of the furnace, where the fuel is converted into energy.

Grate incinerators are widely applied for the incineration of mixed municipal wastes. In Europe approximately 90% of installations treating MSW use grates [5].

Grate incinerators usually have the following components:

- waste feeder
- incineration grate
- bottom ash discharger
- incineration air duct system
- incineration chamber
- auxiliary burners.

When introduced onto the grate, as described above, the waste is first dried, then partly pyrolised under formation of combustible as well as incombustible gases. The combustible gases burn above the grate. The remaining waste is subsequently burned out on the grate to a total organic carbon (TOC) content of less than 3% before it falls into the – normally wet – bottom ash system. Primary combustion air is supplied from underneath through small openings in the grate. The air supply is determined by two considerations: firstly, enough air must be supplied to cool the grate (air-cooled grate), and secondly, enough air must be supplied to sustain the (primary) combustion. Thus, by partly cooling the grate by water (water-cooled grate), it is possible to adjust the primary air supplied to exactly the flow needed for the primary combustion process, only. Most grates are cooled, most often with air. In some cases a liquid cooling medium (usually water) is passed through the inside of the grate. The flow of the cooling medium is from colder zones to progressively hotter ones in order to maximise the heat transfer. The heat absorbed by the cooling medium may be transferred for use in the process or for external supply.

Water cooling is most often applied where the calorific value of the waste is higher (e.g.>12 - 15 MJ/kg for MSW). The design of the water cooled system is slightly more complex than air cooled systems.

The addition of water cooling may allow grate metal temperature and local combustion temperature to be controlled with greater independence from the primary air supply.

This may then allow temperature and air (oxygen) supplied to be optimised to suit specific on-grate combustion requirements and thereby improve combustion performance. A better control of grate temperature can allow incineration of higher calorific value waste without the normally increased operational and maintenance problems.

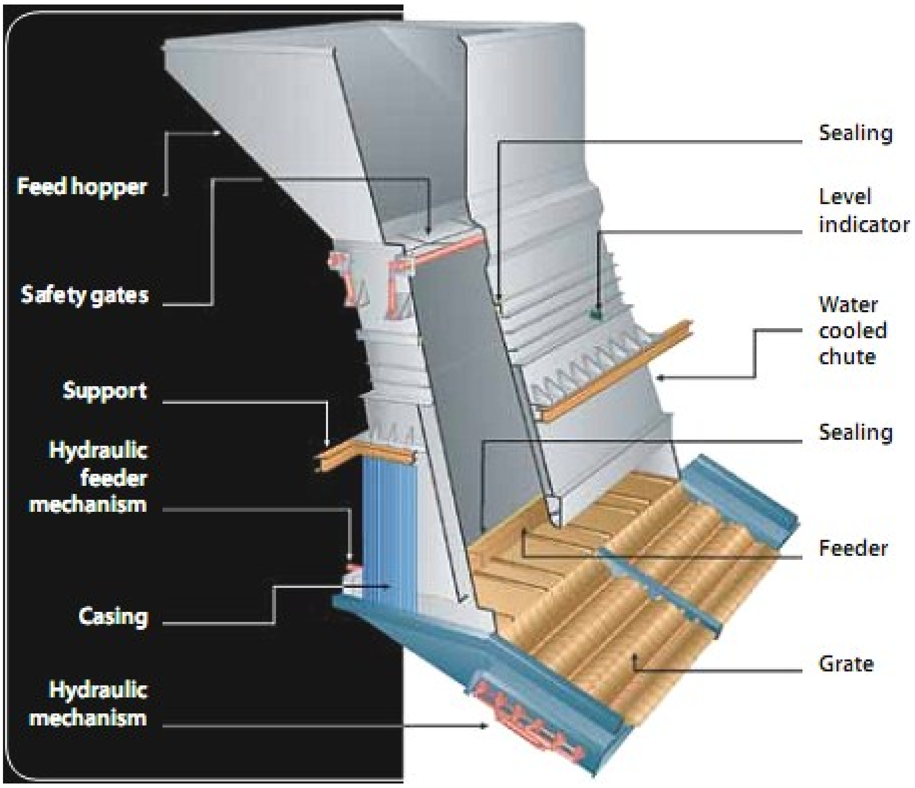


Figure 7 : A typical waste feeder and grate device [7].

Normally, the residence time of waste on the grates is not more than 60 minutes.

In general, one can differentiate between continuous (roller and chain grates) and discontinuous feeder principles (push grates). Figure 8 shows different types of grates while in Figure 9 a picture of a push forward grate is presented. Different grate systems can be distinguished by the way the waste is conveyed through the different zones in the combustion chamber. Each has to fulfil requirements regarding primary air feeding, conveying velocity as well as mixing of the waste. Many modern, facilities for municipal wastes, use reciprocating grates.

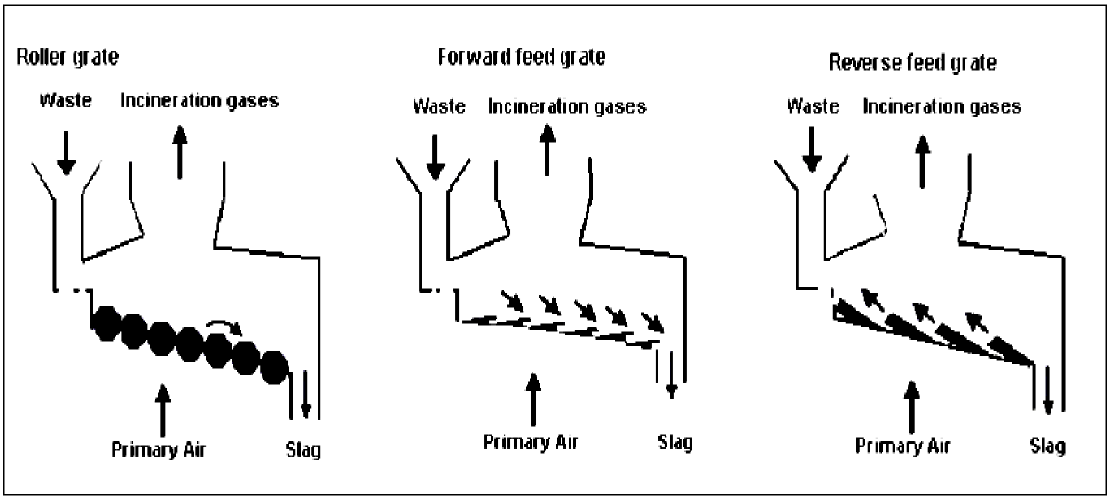


Figure 8 : Different grate types [5].

Figure 9 : A typical push forward grate.

Numerous variations of this type of grate exist, some with alternating fixed and moving sections, others with combinations of several moving sections to each fixed section. There are essentially two main reciprocating grate variations:

– Reverse reciprocating grate: the grate bars oscillate back and forth in the reverse direction to the flow of the waste and is comprised of fixed and moving grate steps.

– Push forward grate: the grate bars form a series of many steps that oscillate horizontally and push the waste in the direction of the ash discharge.

2.2.4. Incineration chamber and boiler

Combustion takes place above the grate in the incineration chamber (see Figure 10). As a whole, the incineration chamber typically consists of a grate situated at the bottom, cooled and non-cooled walls on the furnace sides, and a ceiling or boiler surface heater at the top. As municipal waste generally has a high volatile content, the volatile gases are driven off and only a small part of the actual incineration takes place on or near the grate. The following requirements influence the design of the incineration chamber:

- form and size of the incineration grate - the size of the grate determines the size of the cross-section of the incineration chamber;
- vortex and homogeneity of flue gas flow - complete mixing of the flue-gases is essential for good flue-gas incineration;
- sufficient residence time for the flue-gases in the hot furnace;
- sufficient reaction time at high temperatures must be assured for complete incineration;
- partial cooling of the flue gases in order to avoid fusion of hot fly ash at the boiler, the flue gas temperature must not exceed an upper limit at the incineration chamber exit.

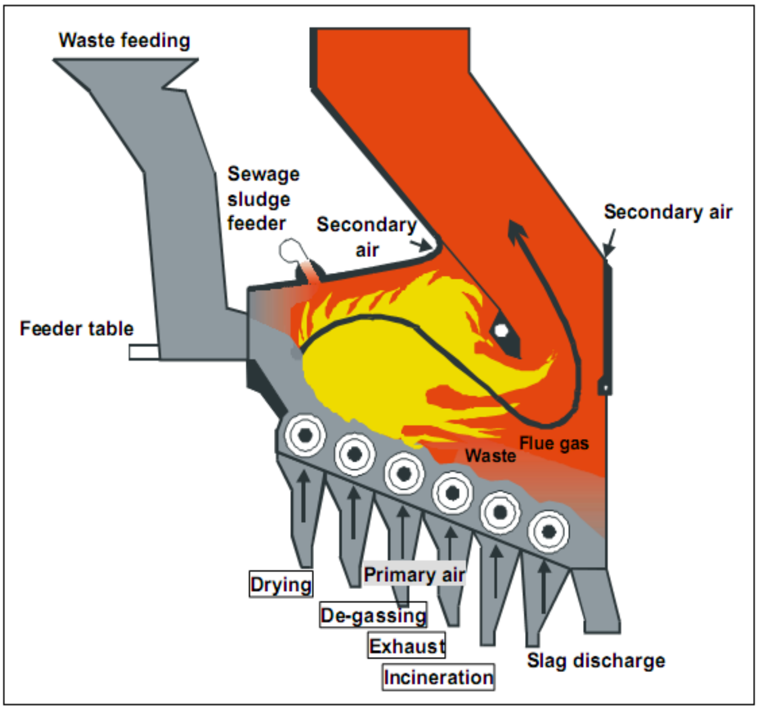


Figure 10 : Example of an incineration chamber [5].

2.2.5. Post combustion chamber

Modern standards require that the flue gas be exposed to a temperature of minimum 850°C for a minimum residence time of 2 seconds after the last secondary air injection [5]. The furnace must therefore have a post combustion or afterburning chamber of a certain height above the grate. The final burnout of the flue gas takes place in that chamber, and secondary combustion air is added in the required amount and in a way that ensure maximum turbulence. Auxiliary burners, if installed, are located in the post combustion chamber; also the injection of ammonia (NH_3) or urea for NOx reduction, according to the Selective Non Catalytic Reduction (SNCR) process, is carried out in this zone.

2.2.6. Flow design and flue gas re-circulation

The total air supply to the combustion process is the sum of the primary and secondary air. To secure complete burnout, it is necessary to operate at a certain surplus of air. The excess air passes all the way through the boiler and the flue gas treatment system to the chimney and, depending on the flue gas temperature in the chimney, it represents a loss of energy and hence a loss of thermal efficiency. The excess air ratio can be reduced, on the primary side, by using a water-cooled grate and on the secondary side by re-circulating a part of the treated flue gas (Flue Gas Recirculation, FGR).

It is also of importance whether the entire gas flow through the furnace is counter flow or parallel flow.

A proportion (approx. 10% – 20% by volume) of the (usually cleaned) flue-gases is re-circulated, normally after pre-dusting, to replace secondary air feeds in the combustion chamber. This technique is reported to reduce heat losses with the flue-gas and to increase the process energy efficiency by around 0.75% - 2%. Additional benefits of primary NO_X reduction are also reported.

2.2.7. Boiler

Boilers for WTE power plants are normally water tube boilers and most often they have four passes: 3 vertical radiation passes and a convective pass. The first of the radiation passes is integrated in the furnace as the post combustion chamber. The convection pass, in which the evaporators, superheaters and economizers are located, may be vertical or horizontal.
When designing a boiler for waste firing, the most important aspect to take into account is the special risk of corrosion. In practice, corrosion problems, as will be described in details later, limit the steam parameters to a maximum of around 450 °C -500 °C and 40 bar - 60 bar. Tubular water boilers are generally used for steam and hot water generation from the energy potential of hot flue-gases. The steam or hot water is generally produced in tube bundles in the flue-gas path. The envelopment of the furnace, the following empty passes and the space where evaporator and superheater tube bundles are located are generally designed with water cooled membrane walls.
In steam generation, it is usually possible to differentiate between the three heat surface areas, shown in Figure 11.

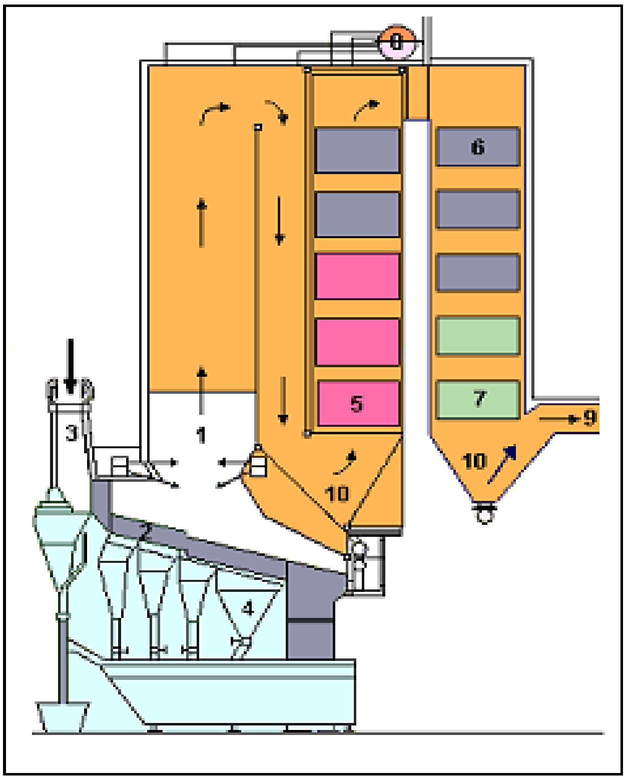


Figure 11 : Typical heat surface areas in a steam generator [5].

Feed-water preheating (Economiser) (7 in Figure 11):
In this area, the boiler feed-water is heated by flue-gases to a temperature close to the boiling point (designed as a bundled heating surface).

Evaporation (5 in Figure 11):
In this area, the water coming out from the economiser is heated until it reaches the saturated steam temperature (designed as a bundled heating surface, envelopment wall of the incineration chamber).

Superheating (6 in Figure 11):
In this area, the saturated steam coming from the evaporator is superheated to the maximum temperature. Spray coolers and surface coolers are used in circulation boilers in order to maintain the exact required steam temperature. It is their function to balance the fluctuations of the steam temperature, these fluctuations being the consequences of load fluctuations, changes in the waste quality, the surplus air, as well as contamination of the heat surfaces (clean or dirty surface).

The preparation of boiler feed water and make up water is essential for a effective operation and to reduce corrosion (inside the tubes) or risk of turbine damage. The quality of boiler water must be higher when increased steam parameters are used.
A compromise is required when determining steam parameters from waste fired boilers. This is because, while the selection of high temperatures and pressures increase waste conversion efficiency into electrical energy, these higher steam parameters can lead to significantly increased corrosion problems, especially at the superheater surfaces and the evaporator; thus, a compromise between high steam cycle parameters and corrosion problems must be looked for. In municipal waste incinerators it is common to use 40 bar and 400 °C, when there is electricity production although higher values are used, especially with pre-treated MSW and prepared Refused Derived Fuel (RDF) (value of 60 Bar and 520 °C are in use with special measures to prevent corrosion). In case of heat production, steam with lower conditions or superheated water may be produced. Based on these rather low (compared to most common fossil fuel power plants) steam parameters, almost exclusively, natural circulation steam boilers are selected.

A feature of waste incineration is the high dust load in flue-gases. The high proportion of ash in flue-gas causes a risk of a correspondingly high contamination of the heat transfer surfaces. This leads to a decline in heat transfer and therefore a performance loss. Thus, heat transfer surface cleaning plays an important role. This cleaning can be

accomplished manually or automatically with lances (compressed air or water jet), with agitators, with soot blowers using steam, with a hail of pellets (sometimes shot cleaning), with sound and shock waves, or with tank cleaning devices.
Different boiler concepts can be used in waste incineration plants. They are from left to right (see Figure 12):

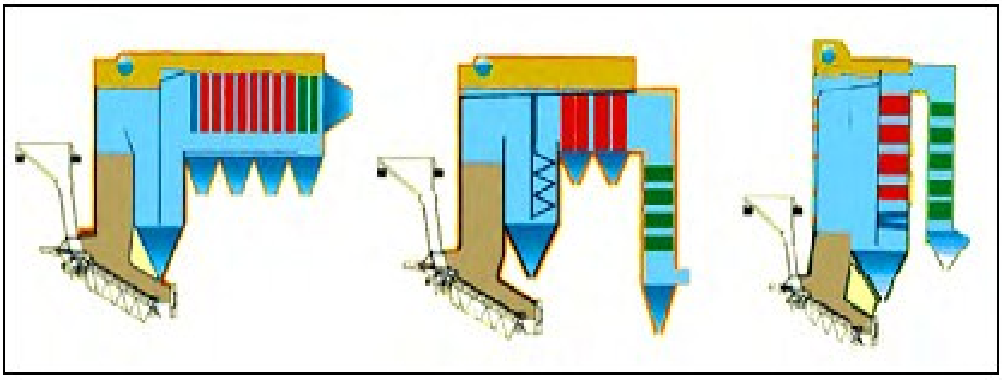


Figure 12 : Overview of boiler systems: horizontal, combination and vertical [5].

• horizontal boilers
• combination of vertical and horizontal boilers
• vertical boilers.

In horizontal and vertical systems usually a number of empty passes with evaporation walls are followed by an arrangement of bundles of heat transfer surfaces i.e. evaporator, superheater and economiser. Horizontal arrangement boilers are characterized by the fact that the flue gas in the convective heating surfaces travels horizontally. One of many advantages of the horizontal design is that the heating surfaces can be cleaned by means of a so-called rapping device which, unlike the traditional steam soot blowers, does not consume steam. As far as the cleaning of the convective heating surface is concerned, the horizontal design means that dirt from the cleaning process enters the hoppers without passing other heating surfaces on its way, thus reducing the risk of blocking the tube bundles and this result in a better availability. Another advantage of the horizontal design is that the support for the heating surfaces can be placed outside the flue gas.
In vertical arrangement boilers the flue gas in the convective heating surfaces travels in the vertical direction. The convective heating surfaces in this type of boiler are usually cleaned by soot blowers. Cleaning with soot blowers is very effective and minimizes the risk of blocking of the tube bundles. To avoid soot blower-induced

erosion, hot tubes (i.e. superheater and evaporator tubes) are protected by stainless steel tube shells.

One of the advantages of the vertical boiler type is that very compact boilers can be designed because the vertical heating surfaces can use a common hopper for ash extraction, thus optimizing performance per ton of steel in the boiler. The arrangement of the tubes means that the tube bundles do not need separate drains - a major advantage in terms of the time required for replacing the bundles.

Typically, the flue gas temperatures in the radiant part of the boiler have to be reduced before entering the convection part to avoid fouling and corrosion. This can be done by introducing membrane baffle walls in the second, or radiant, pass in the boiler. In general, at the entrance of the convection section (located after the third pass) flue gas has a temperature lower than 700-650 °C

2.2.8. Corrosion protection

The modern WTE plants have significantly improved and are far superior compared to the older polluting incinerators. However, the variability in the heating value of the MSW feed and its relatively high content of chlorine and various light metals contribute to a highly corrosive atmosphere that shortens the life of the heat exchanger tubes used in the water wall section and, especially, in the steam superheater sections where the tube temperature is at its maximum. A typical corrosion diagram, derived from practical experience, is shown in Figure 13. The diagram, were horizontal and vertical axis reported respectively flue gas and water walls surface temperature, highlights two different zones, characterized by different corrosion rate.

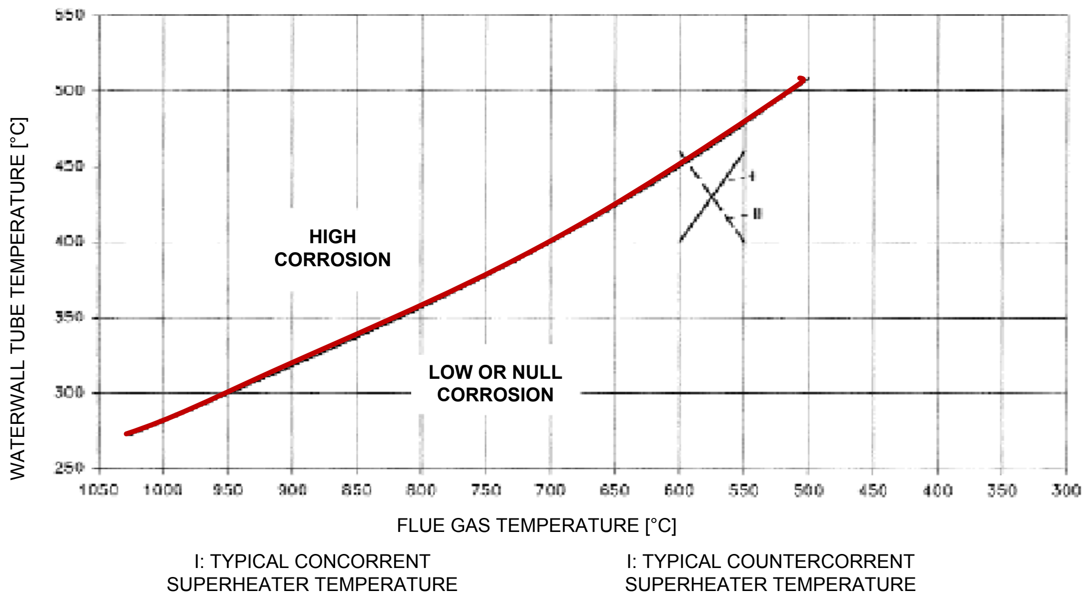


Figure 13 : Typical corrosion diagram [8].

Metal tube corrosion is the major operating problem because it results in downtime and periodic shutdowns in WTE plants and accounts for a significant fraction of the total operating cost of WTE plants. High temperature corrosion has also environmental impacts. Metallic coatings and corrosion resistant alloys such as stainless steels and nickel-base alloys are often used to protect boilers from corrosion and represent a large use of valuable resources. As explained in previous paragraph, the conversion efficiency of steam energy into electricity increases with higher steam temperatures and pressures. However, increasing steam temperature, the heat transfer surfaces are subjected to severe high temperature corrosion, caused by both the metal chlorides in the ash particles deposited on the gas tubes and by the high concentration of HCl in the process. The incineration chamber, the water walls of the first passes and the superheater are boiler's components more affected by corrosion. Under the same operating conditions, superheater tube bundles in a vertical design boiler, which has fewer gas passes will have more critical corrosion problems than the tubes in a horizontal design boiler, because the former are subjected to higher metal temperatures and flow velocities of flue gas. The disadvantage of the horizontal design is that it needs more floor space than the vertical design.

Figure 14 shows the corrosion sensitive areas in a WTE facility [9]. The chlorine concentration in the combustion gas depends entirely on the MSW composition and varies somewhat from region to region. Approximately, one half of the Cl content of MSW is due to natural organics and the other half to chlorinated plastics, mostly PVC. During combustion, nearly all of the chlorine content in the various components of the MSW, both natural organics and chlorinated plastics is volatilized and converted to HCl gas. Assuming that the MSW contains 0.5% Cl, the HCl concentration can be calculated to be about 580 ppmv [9].

Beyond corrosion problems, other negative aspect occurring in a WTE plant is represented by erosion, which is the abrasion of surface material through vertical wear-and-tear, is caused primarily by the ash particles present in flue-gas. Erosion appears mostly in the area of gas redirection. Tube wear is caused by a combination of corrosion and abrasion.

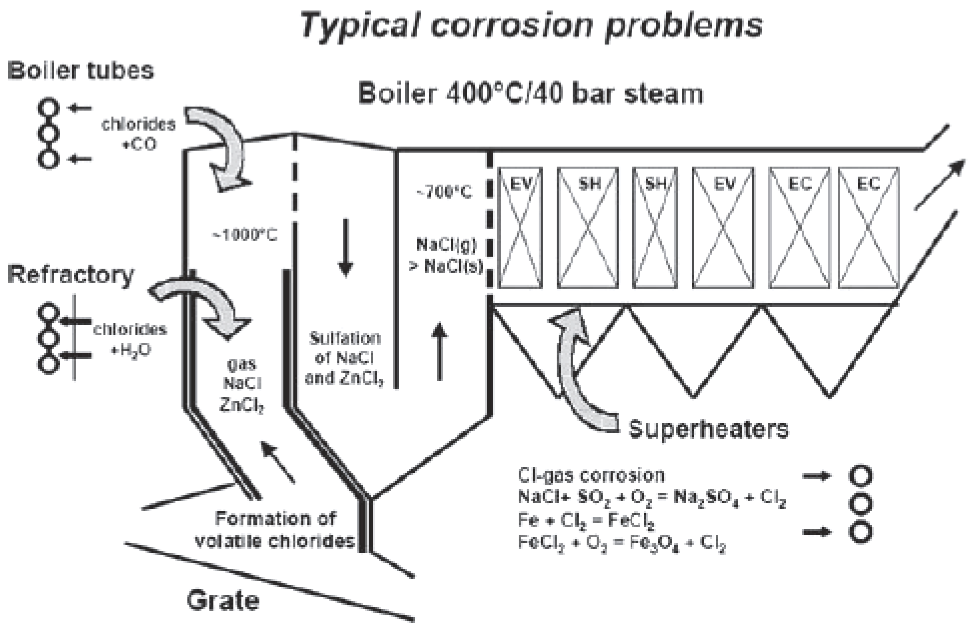


Figure 14 : Corrosion sensitive areas in WTE boilers [9].

Various types of corrosion processes exist in a WTE power plant [10]:

• High temperature corrosion, mainly affecting superheater exchanger.
• Initial corrosion, during start-up operation, due to ferrous chloride formation before the first oxide layer at "blank" steel.
• Oxygen-deficiency corrosion, through $FeCl_2$-formation under deoxygenated flue-gas atmosphere and in the furnace area. This corrosion is observed in individual cases with steam pressures above 30 bar, but more usually above 40 bar.
• Chloride-High temperature corrosion: corrosion by chloride, which is released during the sulphating of alkaline chlorides, and attacks iron or lead hydroxides. This corrosion mechanism is observed in waste incineration plants with flue-gas temperatures greater than 700 °C and at pipe wall temperatures above 400 °C.
• Molten salt corrosion: The flue-gas contains alkali and similar components, which can form eutectics. Eutectic compounds have a lower melting point than the single components which form the eutectic system. These molten systems are highly reactive and can cause severe corrosion of steel. They can react with the refractory lining and lead to the internal formation of compounds which destroy the refractory mechanically. It can also form low viscous melts on the surface consisting of deposited material and refractory material (refractory corrosion).
• Electrochemical Corrosion: This is based on the electrical potential equalisation of different metals. The conductor can be aqueous or a solid that shows sufficient electrical conductivity at the temperatures seen. The conductivity can arise from the water dew point to the sulphuric acid dew point to molten salt.
• Standstill corrosion: based on its high chloride content (especially $CaCl_2$).
• Dew point corrosion: when temperature of flue gas falls beneath the acid dew point, wet chemical corrosions appear on cold surfaces. This damage can be avoided by raising the temperature or by selecting an appropriate material.

Various factors can affect the rate of corrosion in WTE boilers, such as the concentration of chlorine and sulphur in the MSW, the operating temperature of the combustion chamber, temperature fluctuations at a particular location that may disrupt the protective oxide layer, the method used for periodic cleaning of the process gas side of the tubes, and the design of the boiler that should avoid extremely high temperatures (e.g. horizontal vs. vertical disposition of superheater tube arrays). The influence of some important factors can be summarized as follows [10], [11]and [12]:

Metal Surface Temperature. High temperature of the metal surface, due to high radiation fluxes and/or inadequate heat transfer rate to the steam flow inside the tube, results in the melting of deposits and acceleration of the corrosion rate. In general, the metal temperatures of water wall and superheater tubes are maintained at temperatures below 300 °C and 450 °C, respectively. However, as mentioned earlier, operation at higher superheater temperatures increases the thermal efficiency of the steam turbine.

Gas Temperature. The temperature of the combustion gases can affect the deposition rates and also the composition of the deposit and thus accelerate corrosion. The temperature gradient between gas temperature and metal surface temperature is a driving force for the condensation of vaporized species, such as metal chlorides, on the cooled surface.

Temperature Fluctuation. The non-homogeneous physical and chemical composition of the MSW fuel and the corresponding fluctuation in heating value with time result in pronounced fluctuations of the gas temperature within the combustion chamber. Experimental studies have confirmed that the corrosion rate increased several times because of wide temperature fluctuation.

Characteristics of Molten Salt Deposits. As already noted, diffusion of chlorine through the cracks and pores of deposits enhances the rate of corrosion. The presence of chlorides, sulphides, and alkaline and heavy metal components in deposits affects both chemical and physical properties of deposits, such as gas permeability of deposits. Contrary to intuition, the corrosion rate also increases with an increase in thickness of deposits. In WTE boilers, the character of deposits is affected by feed composition and the gas-metal temperature gradient.

Counter measures only help to reduce corrosion damage to an acceptable level.

Primary measures seek to eliminate corrosion by influencing the process conditions in the boiler. Some of these methods include [12]: (i) improvement of process control, in particular minimizing fluctuations in gas temperature; and (ii) design modifications, to alter flow dynamics, enhancing mixing of gas through gas recirculation, and design of the boiler system (e.g. horizontal vs. vertical boiler).

Secondary methods of protection are applied to extend the lifespan of the boiler tubes.

Improvement possibilities are mainly found in the steam generator. Low steam parameters, long reaction times before entry into the heat surfaces, lowering the flue-gas speed, and levelling of the speed profile could all be successful. A compromise must be found in determining the boiler cleaning intensity between the best possible heat transfer (metallic pipe surface) and optimal corrosion protection.

HCl is highly corrosive at high (> 450°C) as well as at low (<110°C) temperatures. To prevent corrosive attacks on the furnace boiler system the heating surfaces in the radiant part is protected by a resistant refractory material and/or welded high-alloy materials. Cladding consists of overlaying a layer of Inconel 625 (21Cr-9Mo-3.5Nb-Ni base) on tubes to protect them from the attack of HCl. This method has been used successfully in the waterwall tubes and part of the superheater tube bundles in many WTE facilities. The key element regarding the cost is how much Inconel 625 is used and where. Some researchers have shown that Inconel 625 applied on waterwall tubes provides excellent corrosion resistance [13] and [14]. Although the price of Inconel 625 is higher than that of a protective refractory lining, the cost of Inconel 625 is partly compensated by avoiding the cost of refractory maintenance. In addition, Inconel 625 has higher thermal conductivity than refractory materials and therefore can reduce gas temperature in the first gas pass. The application of Inconel 625 on superheater tubes is more complicated because the performance of the cladding depends on the metal temperature reached during operation.

In the radiant passes the flue gas is cooled slowly to a temperature of less than 700 °C before it, in the convections pass, is further cooled by the heating surface bundles there. To prevent low temperature corrosion the feed water should be preheated to minimum 125°C before being introduced in the boiler [15].

In 2004, the Waste-To-Energy Research and Technology Council (WTERT) conducted a corrosion survey of several WTE facilities in the U.S. One of the results of the survey showed that the non-scheduled downtime due to corrosion ranged from 0 to 20 days per year (Figure 15). Another result showed that the yearly maintenance cost per boiler unit due to corrosion ranged from \$18˙000 to \$1˙200˙000 (Figure 16); the maintenance cost due to corrosion ranged from \$0.23 to \$8.17 per ton of MSW combusted (see Figure 17). The typical cost is in the range of \$4 per short ton of MSW combusted. Capital cost and maintenance cost account for approximately 60% and 15% of the yearly cost of a WTE facility, respectively [15]. Therefore, the corrosion problems will cost WTE approximately 5% of its yearly total cost, if the corrosion/total maintenance cost ratio of 1/3 applies. The actual cost will be even higher if the revenue loss due to shutdowns because of corrosion is taken into account.

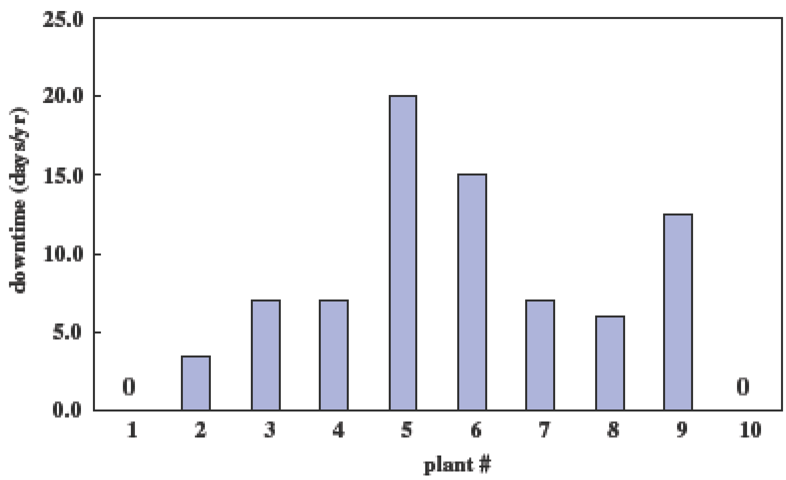


Figure 15 : Non scheduled downtime due to corrosion in different WTE plants [15].

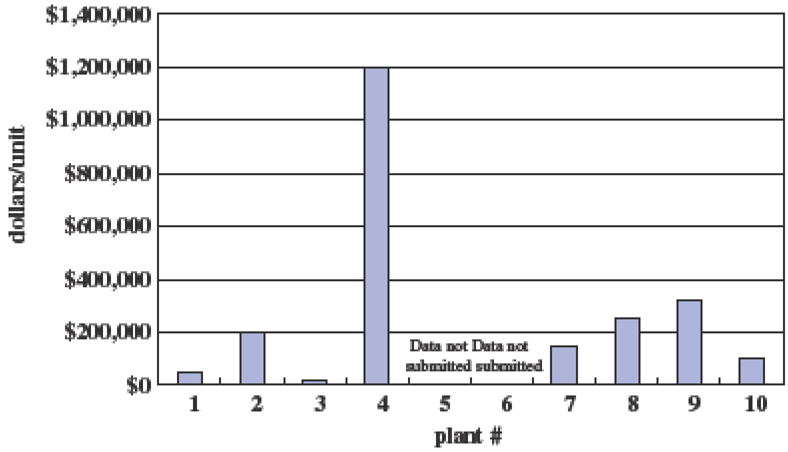


Figure 16 : Yearly maintenance cost per unit due to corrosion in different WTE plant [15].

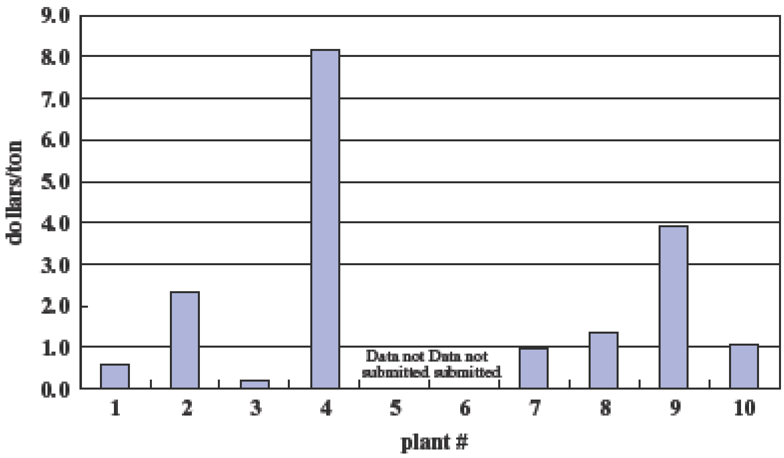


Figure 17 : Cost due to corrosion per ton of MSW combusted in different WTE plant [15].

2.3. *Energy input and output to WTE power plant*

In addition to the energy in the waste, there are other inputs to the incinerator that need to be recognised when considering energy efficiency of the plant as a whole [5].

– Electricity inputs:

Electrical consumption is usually easily calculated. In situations where economic incentives are provided to support the production of electrical energy from incineration (e.g. as a renewable source) there may be a price differential between purchased and exported electricity. Plants may then choose (for economic reasons) to export all of the electricity generated by the incinerator, and import from the grid, which is required to run the incineration process itself. Where this is the case, the incineration plant will often have distinct electricity flows for input and output.

– Steam/heat/hot water inputs:

Steam (heat or hot water) can be used in the process. The source can be external or circulated.

– Fuels:

They are required for several uses. For instance, conventional fuels (typically natural gas) are consumed in order to:

- ensure that the required combustion chamber temperatures are maintained (this then contributes to steam production);
- increase the temperature in the combustion chamber to the required level before the plant is fed with waste (this contributes partially to steam production);
- increase the flue-gas temperature (e.g. after wet scrubbers) in order to avoid bag house filter and stack corrosion, and to suppress plume visibility ;
- preheat the combustion air;
- heat-up the flue-gas for treatment in specific devices, such as SCR or fabric filters.

When considering the overall efficiency of recovery of energy from the waste, it is important to note that some of these primary fuel uses can contribute to steam production and others will not.

Energy outputs from waste incinerators

- Electricity:

The electricity production is easily calculated. The incineration process itself may use some of the produced electricity.

- Fuels:

Fuel (e.g. syngas) is produced in gasification/pyrolysis plants and may be exported or combusted on site with (usually) or without energy recovery.

- Steam/hot water:

The heat released in the combustion of waste is often recovered for a beneficial purpose, e.g. to provide steam or hot water for industrial or domestic users, for external electricity generation or even as a driving force for cooling systems.

Combined heat and power (CHP) plants provide both heat and electricity. Steam/hot water not used by the incineration plant can be exported.

2.4. External factors affecting energy efficiency

The characteristics of the waste will determine the techniques that are appropriate and the degree to which energy can be effectively recovered. Both chemical and physical characteristics are considered when selecting processes.
The chemical and physical characteristics of the waste actually arriving at plants or fed to the incinerator can be influenced by many local factors including:

- contracts with waste suppliers (e.g. industrial waste added to MSW)
- on-site or off-site waste treatments or collection/separation regimes
- market factors that divert certain streams to or from other forms of waste treatment.

Table 3 gives typical net calorific value ranges for some waste types.

Table 3 typical net calorific value ranges for some waste types [5].

Input type	Examples	LHV in original substance (humidity included)	
		Range [GJ/t]	Average [GJ/t]
Mixed Municipal Solid Waste (MSW)	Mixed household domestic waste	6.3-10.5	9
Waste similar to MSW	Waste similar to household waste but arising from shops, offices etc.	7.6-12.6	11
Residual MSW after recycling operation	Screened out fraction from composting and material recovery processes	6.3-11.5	10
Refused derived fuels (RDF)	Pellet or flock material produced from municipal and similar non hazardous waste	10-15	18
Product specific industrial waste	e.g. plastic or paper industrial residues	18-23	20
Hazardous waste	Also called chemical or special waste	0.5-20	9.75

2.5. *Focus on WTE power plants in Italy*

A detailed analysis of the quantity and quality of the treated waste incineration plants at the regional level has been shown in Chapter 1 thus, this paragraph mainly focuses on incinerations and Waste-To-Energy power plants present in the Italian contest. Data, presented in Chapter 1, show for the Italian contest a total amount of Municipal Solid Waste produced, in 2009, equal to 32.5 millions of tones equal to about 541 kg/p.p. The total amount of waste thermally treated are equal to about 5 million of tonnes.

A recent study, conducted by Federambiente for CEWEP country report [16], reported in Table 4, shows 49 operative incinerators at the end of 2009 in Italy. Energy recovery is performed in almost all plants (47 out of 49), electricity production is made in every case while only 8 plants combine both electric and thermal production (CHP).

While plants with electrical energy recovery treated 3.1 million tonnes of waste, recovering more than 1.9 million MWh of electricity, plants equipped with CHP energy recovery have treated 1.8 million tonnes of waste, recovering 1.2 million MWh of electricity and 965000 MWh of thermal energy.

Table 4 : Total electricity and heat generated in WTE plants in Italy in 2009 [16].

	Number of Plants	**Amount of waste thermally treated** [million tonnes/year]	**Electrical Energy** [MWh_e/year]	**Thermal Energy** [MWh_t/year]
Incineration Plants without Energy recovery	2	26˙421	-	-
Incineration Plants with electrical and thermal energy recovery (CHP)	8	1˙838˙942	1˙232˙368	964˙615
Incineration Plants with electrical energy recovery	39	3˙150˙901	1˙939˙478	-
TOTAL	**49**	**5˙016˙264**	**3˙171˙846**	**964˙615**

Accurate studies on Italian WTE design capacity and operative parameters has been performed by ENEA at the end of 2008[2][17]. Complete data are reported in Table 5.

[2] At the end of 2008 51 were the incinerators operative on Italian contest.

Figure 18 shows the distribution of Italian incinerators in terms of thermal capacity (as the product of MSW mass flow rate and LHV) and nominal gross electric power. It is interesting to note that 37 plants out of 51 have a thermal capacity below 50 MW_{LHV}, while only 3 plants (Brescia, Milano and Parona) exceed 100 MW_{LHV}. Figure 18 also shows different values of gross electric efficiency. Looking at the figure it can be observed that only few plants have a gross electric efficiency above 30% while for the most part it is between 18% and 25%. It is important to point out that plants with low electrical energy production or, at least, with no energy recovery, have a relatively old start-up: 16 out of 51 plants started up before 1980.

Collecting the steam cycle operative parameters (evaporative pressure and steam superheated temperature) Figure 19 has been obtained describing the WTE power plant operative in Italy. As stated by the study, evaporative pressure value ranged between 10 bar and 75 bar; 55.1% of cases (27 plants) is within a range between 20 bar and 40 bar, with a peak around 40 bar. The operating temperature is typically 400 °C. Brescia WTE power plants achieved the best performance with 75 bar and 450 ° C of evaporation pressure and steam maximum temperature, respectively.

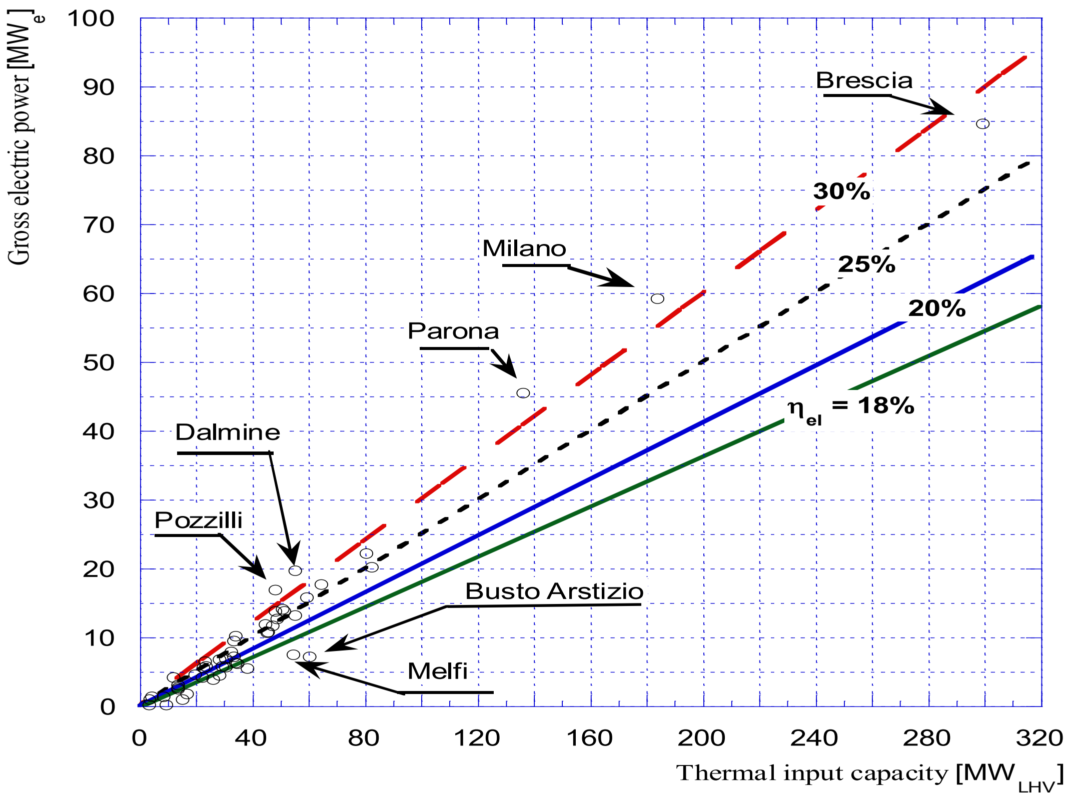


Figure 18 : Gross electric power function of thermal capacity for Italian incinerators (data at 31/12/2008).

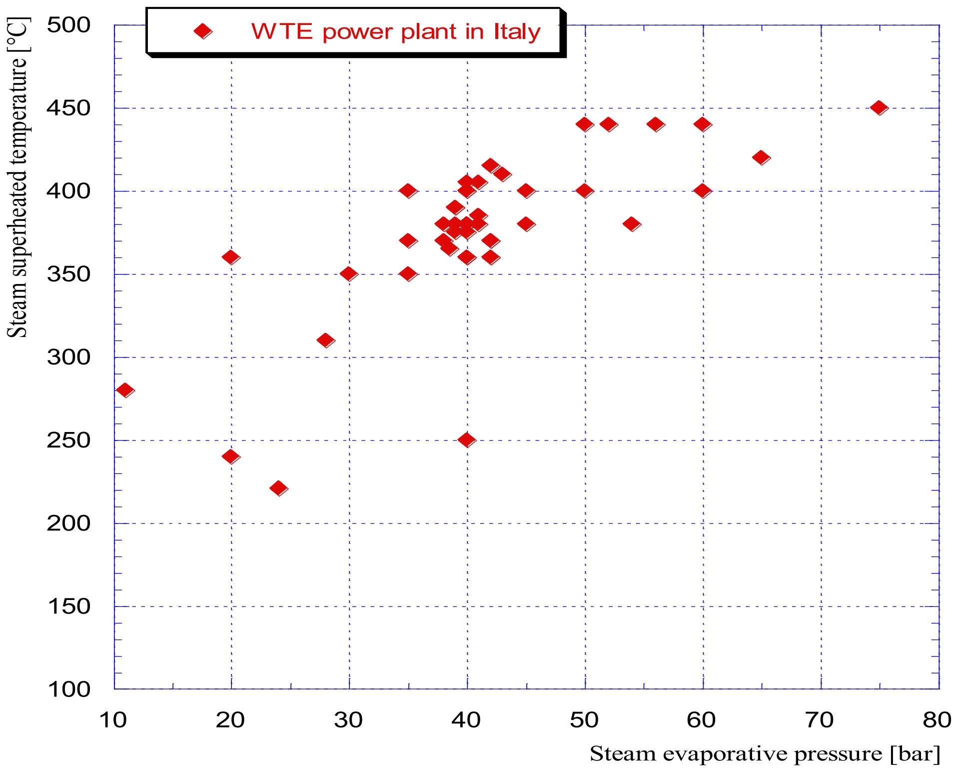


Figure 19 : Operative parameters (evaporative pressure and steam superheated temperature) for the WTE plants in Italy.

Table 5 : Incinerators plant in Italy at 31/12/2008 [17].

N.	Location	N. of lines	Year of startup or repowering	Total waste treated: tonnes/day	Total waste treated: Total capacity [MW]	Type of furnace	Steam parameters: steam Pressure [bar]	Steam parameters: steam Temperature [°C]	Steam parameters: Constructor	Sp[3].	GrossElectric power [MW]	Energy recovery
Piemonte												
1	Prato Michelaccio, Mergozzo (VB)	2	1960/1997	53	6,4	MG	40	360	De Bartolomeis	-	4,0	EE
				53	6,4	MG						
2	Vercelli (VC)	3	1999/2004	74	7,6	MG	35	400	Frassi e De F.	-	2,6	EE
			1995/2003	74	7,6	MG	35	360	Frassi e De F.			
			1992/1997	74	7,6	MG	33	320	Frassi e De F.	-	1,4	
Lombardia												
3	Bergamo (BG)	1	2003	228	48,0	BFB	56	440	CCT / EPI	-	11,5	CHP
4	Brescia (BS)	3	1998	864	100,0	MG	75	450	Ansaldo	-	84,4	CHP
				864	100,0	MG						
			2004	864	100,0	MG						
5	Busto Arsizio (VA)	2	2000/07	252	30,5	MG	40	380	Comef Group	-	7,0	EE
				252	30,5	MG	40	380				
6	Como (CO)	2	1968/2008	172	20,8	MG	38	380	Comef / Galleri	-	5,3	CHP
			1998/2004	150	18,2	MG						
7	Corteolona (PV)	1	2004	226	34,0	BFB	40	405	Kvaerner	-	9,3	EE
8	Cremona (CR)	2	1997/2007	192	17,8	MG	41	385	Crugnola / Saporiti	-	6,0	CHP
			2001	192	17,8	MG						
9	Dalmine (BG)	2	2002	222	27,9	MGW	65	420	Macchi	-	19,5	EE
				222	27,9	MGW						
10	Desio (MI)	2	1976/2003	106	15,0	MG	24	221	Mariotti	-	5,6	CHP
				106	15,0	MG						
11	Milano ("*Silla 2*")	3	2000/2007	480	61,5	MG	52	440	ABB Flakt	2	59,0	CHP
				480	61,5	MG						
				480	61,5	MG						
12	Parona (PV)	2	1999	473	57,0	FCB	60	440	Foster Wheeler	-	19,5	EE
			2007	540	80,0	FCB					25,8	
13	Sesto San Giovanni (MI)	3	2000/2001	79	10,4	MG	40	360	Crugnola Termosud	-	5,5	EE
				79	10,4	MG						
				79	10,4	MG						
14	Trezzo sull'Adda (MI)	2	2002	271	41,6	MGW	40	400	CCT	-	20	EE
				271	41,6	MGW						
15	Valmadrera (LC)	2	1981/2008	139	18	MG	40	400	Frassi De Ferrari	-	10,5	EE
			2006	161	28,0	MG			Sices			
Trentino Alto Adige												
16	Bolzano (BZ)	2	1988/2001	120	14,0	MG	42	360	Sices	-	3,3	CHP
			1994	180	21,0	MG				-	2,8	
Veneto												
17	Cà del Bue (VR)	2	1999	288	35,0	FBB	54	380	Fontana - CCT	-	21,8	EE
				288	35,0	FBB				-		
18	Fusina (VE)	1	1998	174	14,3	MG	41	380	In Steam	-	2,2	EE
19	San Lazzaro (PD)	2	1962/86	150	14,5	MG	42	380	Frassi De Ferrari	-	3,3	EE
			1970/00	150	14,5	MG	42	380	Frassi De F.	-	3,3	
20	Schio (VI)	3	1983/2005	36	6,1	MG	20	240	Giberti	-	7,7	EE
			1991/2006	60	10,2	MG	20	295	Idrotermici	-		
			2003	100	17,0	MG	40	380	Sprinco	-		
Friuli Venezia Giulia												
21	Trieste	3	2000/04	204	21,7	MG	39	380	Insteam	-	17,5	EE
				204	21,7	MG	39	380	Comef			
			2004	204	21,7	MGWC	39	380	Ruths			
Emilia Romagna												
22	Coriano (RN)	1	1976/19912004	192	23,3	MG	40	375	Ruths	-	5,4	EE
23	Ferrara (FE)	2	1993/2007	192	27,9	MGWC	45	400	Ruths	-	13,0	CHP
				192	27,9	MGWC						
24	Forlì (FC)	1	1976/2008	380	46,5	MGWC	45	380	n.d.	-	10,6	CHP
25	Granarolo nell'Emilia (BO)	2	2004	150	20,35	MGWC	49	440	CCT	2	22,0	CHP
				150	20,35	MGWC	49	440				
26	Modena (MO)	3	1980/94	144	9,1	MG	20	360	CCT	1	7,0	EE
				144	9,1	MG			CCT			
				250	15,8	MG			MAW			
27	Piacenza (PC)	2	2002	180	22,7	MG	39	390	CNIM	3	11,7	EE
				180	22,7	MG	39	390				
28	Ravenna (RA)	1	2000	144	24,0	FBB	40	380	CCT	-	6,3	EE
29	Reggio Emilia (RE)	2	1968/2004	100	14,5	MG	11	280	Carimati Sprinco	-	4,3	Cog.
			1968/2005	100	14,5	MG						

[3] Number of regenerative steam turbine bleeds.

N.	Location	N. of lines	Year of startup or repowering	Total waste treated		Type of furnace	Steam parameters			Sp[4].	GrossElectric power [MW]	Energy recovery
							steam					
				tonnes/day	Total capacity [MW]		Pressure [bar]	Temperature [°C]	Constructor			
						Toscana						
30	Arezzo (AR)	1	2000	120	14,5	MG	40	380	Crugnola	-	2,9	EE
31	Castelnuovo di Garfagnana (LU)	1	1997	36	4,3	MG	40	250	Sprinco	-	0,8	EE
32	Pietrasanta, Falascaia (LU)	2	2002	84	12,2	FBB	40	400	Kvaemer Pulping	-	5,6	EE
				84	12,2	FBB				-		
33	Livorno (LI)	2	1974/2003	90	15,6	MGWC	42	370	Frassi De Ferrari	-	6,7	EE
				90	15,6	MGWC			Atzwanger	-		
34	Montale Agliana (PT)	2	1978/2001	60	7,9	ROT	30	350	Crugnola	-	0,8	EE
				60	7,9	ROT				-		
35	Ospedaletto (PI)	2	1980/2002	120	10,2	MG	38	370	Saporiti	-	4,4	EE
				120	10,2	MG				-		
36	Foci, Poggibonsi (SI)	3	1977/2008	29	3,5	MG	40	360	n.d.	-	8,4	EE
				29	3,5	MG			n.d.			
				170	27,9	MGWC	40	370	n.d.			
37	Rufina (FI) "I Cipressi"	1	1975/2005	29	4,0	MG	-	-	*(Progetto di reupero)*	-	-	No rec.
						Umbria						
38	Terni (TR)	2	1998	48	7,3	MG	42	360	Crugnola	-	2,5	EE
				48	7,3	MG						
						Marche						
39	Tolentino (MC)	1	1989/1997	60	9,3	FG	28	310	Snam Progetti	-	1,2	EE
						Lazio						
40	Colleferro (RM), *("Mobilservice")*	1	2002	331	52,9	MGWC	43	410	CCT - Marcegaglia	-	13,6	EE
41	Colleferro (RM), *("EP Sistemi")*	1	2002	336	49,0	MGWC	43	410	CCT - Marcegaglia	-	13,6	EE
42	S. Vittore del Lazio (FR)	1	2002	288	49,0	MGWC	42	415	CCT	-	13,6	EE
						Molise						
43	Pozzilli (IS)	1	2007	258	49,0	MG	60	400	CNIM	-	16,7	EE
						Puglia						
44	Massafra (TA)	1	2003	288	49,5	FBB	50	400	CCT	-	12,5	EE
45	Statte (TA)	2	1976/2001	100	13,3	MG	39	390	Fontana Sud	-	3,7	EE
				100	13,3	MG				-		
						Calabria						
46	Gioia Tauro (RC)	2	2004	206	30,0	FBB	41	405	Kvaerner/CGT	-	15,6	EE
				206	30,0	FBB				-		
						Basilicata						
47	Melfi (PZ)	2	1999/2005	165	20,6	MG	35	350	Macchi	-	7,3	EE
				127	34,9	ROT				-		
48	Potenza (PZ)	2	2005	36	2,6	MG	40	380	Frassi De Ferrari	-	1,2	EE
				36	2,6	MG				-		
						Sardegna						
49	Macchiareddu (CA)	4	1995/2005	120	11,6	MG	38,5	365	Frassi De Ferrari	-	9,4	EE
				120	11,6	MG	38,5	365	Frassi De Ferrari			
			2005	168	16,3	MG	38,5	365	Kawasaki	-	4,5	
				72	12,0	ROT	38,5	365	Frassi De Ferrari			
50	Macomer (NU)	2	1994	72	8,8	FBB	35	370	KTI	2	1,6	EE
			1998	72	8,8	FBB	35	370	Frassi De Ferrari			
						Sicilia						
51	Messina	2	1979/01	50	5,1	MG	-	-	-	-	-	No rec.
51	**Totale**	**97**		**18205**	**2355**						**587**	

Legend:
MG: grate; MGWC: water cooled grate; ROT: rotary klin; FBB: fluidized bed; FCB: circulated bed
EE: electrical energy; CHP: Combined heat and power recovery

[4] Number of regenerative steam turbine bleeds.

2.6. *Focus on “il Frullo”: upgraded WTE power plant*

Born in 1971, "il Frullo" has now been completely renovated and upgraded. From December 23, 2005, after the start-up and testing, Frullo Energia Ambiente (Fea S.r.l) manages the new WTE power plant located in the city of Granarolo dell’Emilia, near Bologna.

The works, started in 2002 and lasted over 2 years, have involved the complete replacement of incineration lines allowing a more efficient energy recovery from waste and improving flue gas cleaning system. The disposal capacity of the new plant has increased by 50% while, electricity sold to the national grid has more than tripled; moreover, the plant, generating both electricity and heat, operates as a Combined Heat and Power plant (CHP).

The plant is able to supply 130 million of kWh/year of electricity and about 24 million Mcal/year to the district heating network, equivalent to the electricity and heating requirements of around 65000 and 3000 households, respectively [18]

Figure 20 : Photograph of “ il Frullo”, WTE plants located in Bologna, Italy.

The benefits achieved with this "advanced" WTE power plant are numerous, both in terms of performance (waste treatment and electricity generation) and in terms of pollutants reduction, thanks to the use of Best Available Techniques (BAT).

These results were obtained through the construction of two new independent lines of waste thermal combustion of about 300 tons/day, each. The facility is mainly fed with municipal solid waste, but the plant is also project to deal with a percentage of non-hazardous special waste and medical waste (equal to about 3500 tons/year)

Plant's nominal power output is equal to about 22 MWe (without generation of heat) while, maximum heat available for district heating is equal to 27.9 MWth. Detailed power plant technical data are reported in Table 7 while a schematic of Frullo's layout is shown in Figure 21.

Waste bunker has a capacity equal to 5400 cubic meters. Progression of waste into the combustion chamber is adjusted, to ensure a combustion temperature above 1000 ° C, by means of special devices acting on grate speed. Water cooled grade are used for waste combustion.

In order to cool the side walls of the combustion chamber to prevent the deposit of molten slag, surfaces are covered by perforated plates of refractory material cooled with tertiary air. Heat removal is also performed by boiler tubes integrated into the combustion chamber. Combustion process is regulated automatically adjusting the waste speed along the grate, distribution of primary, secondary , tertiary air and flue gas recirculation in order to guarantee a right combustion temperature and an optimum concentration of oxygen in each zone of the combustion chamber. Oxygen content in dry exhaust gas is maintained equal to vol 7%- vol 8%. Combustion gases enters the post-combustion chamber at a temperature greater than 850 ° C for, at least, 2 seconds. By means of suitable nozzles, positioned at the entrance of the post-combustion chamber, a percentage of exhaust flue gas (about vol 15 %) and secondary air are injected at high speed, with the aim of reducing the nitrogen oxides and to complete oxidation of combustion products. Two additional natural gas burners are provided with the task of maintaining post-combustion temperature above the allowed value (850 °C) and for furnace start-up operation.

Three vertical channels (1°, 2° and 3° pass) followed by a horizontal zone are present in the boiler convective section where superheater and economizer heat exchangers are placed. To prevent corrosive attacks the first and the second pass are protected with refractory material and tubes cladding consists of a Inconel 625 layer.

The plant has a high thermoelectric efficiency thanks to the integration of the boiler within the combustion chamber and to the heat recovery from exhaust gas, cooled down to a temperature of 180 °C. The boiler is designed to obtain a high thermal efficiency and is therefore equipped with superheaters to increase steam cycle maximum temperature and economizers for heat recovery from flue gas. Superheated steam (440

°C and 50 bar) is sent to a dual stage steam turbine with one controlled (for district heating and deaerator) and one free (for feed water preheating) steam bleeds. A significant amount of heat is extracted by ST bleed for district heating, while second bleed feds a regenerative exchanger to raise temperature of feeding water out of condenser, improving the performance of the thermal cycle. Heat recovery is also performed in a gland condenser for vapor recovery from ST and a heat exchanger downstream of the DeNOx selective catalytic reduction system.

Main and auxiliary condenser cooling is achieved by the use of water cooling towers; condensation pressure is equal at 0.07 bar.

Steam cycle auxiliary consumption is close to 5% of the nominal electric power output while, total power plant auxiliaries consumptions are equal to about 3.3 MW (about 15% of the nominal power output).

Table 7 : Frullo's technical data [18].

WTE section	
Type of waste	Municipal solid waste, non-hazardous special waste and medical waste
Number of lines	2
Consumption of wastes both lines	600-700 tons/day (12.5 t/h each line)
Waste average LHV	2800 kcal/kg
Thermal input capacity	81.41 MWt
Primary air temperature	120 ÷ 130 °C
Secondary air temperature	50 °C
Exhaust gas recirculation	15 %
Exhaust gas recycle temperature	150 °C
Exhaust gas outlet temperature	170 ÷180 °C
Oxygen in dry exhaust gas	7 ÷ 8%
STEAM	
Saturated steam mass flow rate	26 kg/h
Steam evaporative pressure	49 bar
Steam superheated temperature	440 °C
ST bleed for deareator, district heating and primary air preheating	3.5 bar
ST bleed for water preheating	1.3 ÷0.8 bar
Condenser pressure	0.07 bar
ENERGY	
Nominal electric power output (without district heating)	22 MW
Nominal electric power output (with district heating, CHP mode)	17 MW
Maximum heat power for district heating	27.9 MWt
EFFICIENCY AND SAVINGS	
Total Electrical Gross Efficiency	0.27
Overall net plant Efficiency CHP mode	0.55
Thermal Energy recovery	30 Million Mcal/year
Thermal Energy and electricity, representing annual fossil fuels saving	37000 tep/year

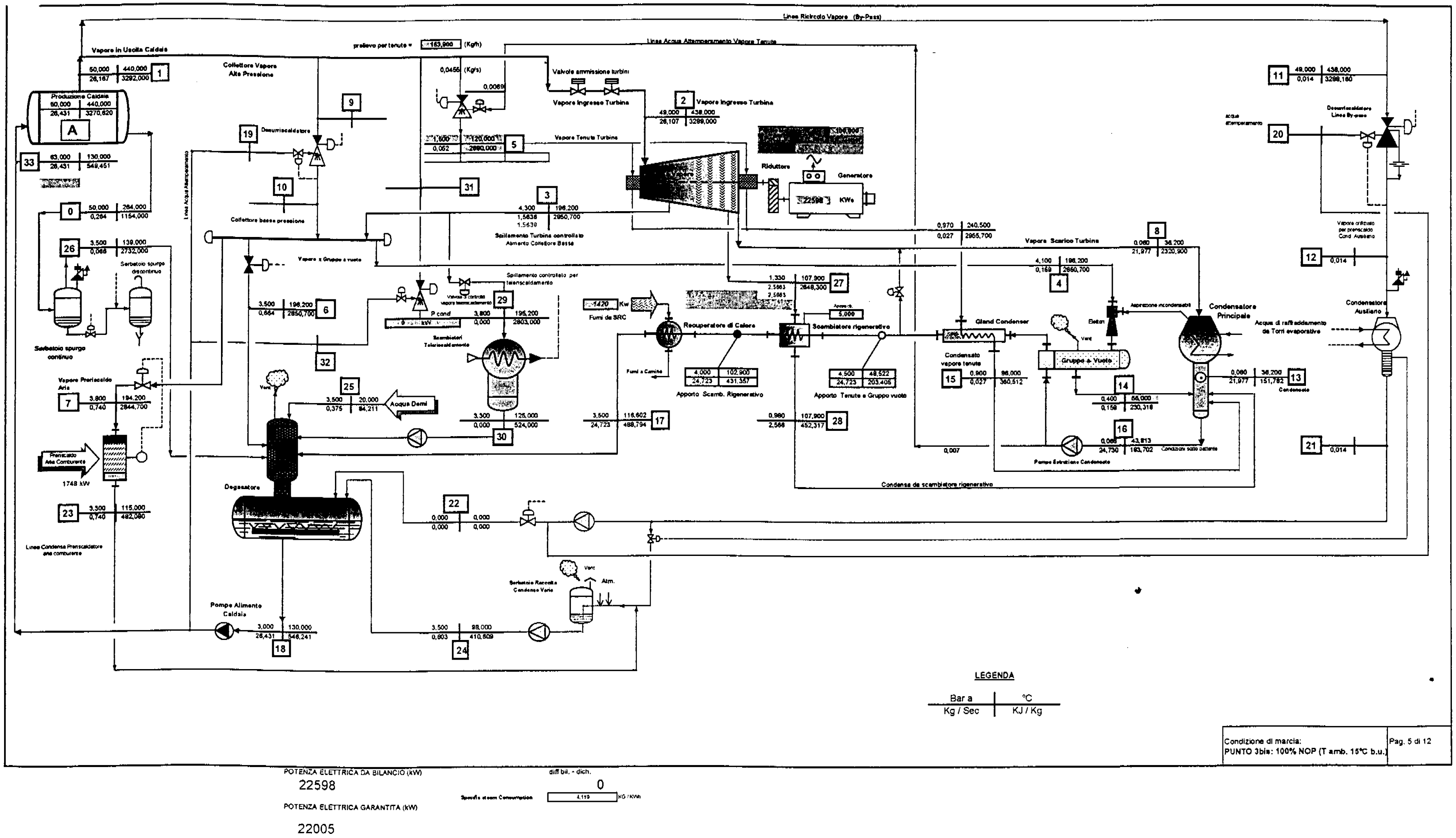


Figure 21 : schematic of Frullo’s layout [18].

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3. The steam cycle in a WTE power plant

The amount of energy recovered from waste combustion can vary significantly with the characteristics of MSW fed into the boiler (composition, mass flow rate and LHV), the combustion technology, the configuration and features of the recovery boiler (adiabatic or integrated) and the characteristics of the thermodynamic cycle.
Due to the heterogeneous nature of waste, some differences with respect to a conventional fossil fuel power plant have to be considered in the chemical-to-electrical energy conversion process.
Corrosion, as explained in Chapter 2, is still the most important problem in MSW boiler; unfortunately corrosion processes have a multiple nature, they change over time according to MSW composition and are strictly related to the characteristic of the steam produced. Erosion, which is the abrasion of surface material, is mainly caused by the ash particles within the flue-gas. Tube wear is caused by a combination of corrosion and abrasion. For boiler normal operative conditions (excluding start-up and shutdown) the kinetics of the corrosion process is influenced by heat exchanger wall temperature and flue gases temperature [1] and [2]. Thus, the characteristics of the produced steam, pressure and temperature play a fundamental role in corrosion generation.
All of these factors expose boiler structure to a variety of critical events; it is therefore necessary to protect the heat exchanger section with refractory materials, reducing the heat flux, to limit the flue gases temperature and to plan a specific heat exchanger section organization.
Aggressive compounds condensation is another limiting factor: in order to avoid it, high WTE boiler outlet temperatures must considered. These limits constrain the steam cycle efficiency, reducing waste to energy conversion.
Corrosion problems bind the inlet turbine temperature to be between 370 and 450 °C. Low superheating temperatures also imply moderate evaporation pressure, in order to limit the liquid fraction at the steam turbine outlet: typically between 40 to 50 bar. Besides, high values of the evaporation pressure (more than 50 bar) makes necessary a protection of the combustion chamber with a noble metal layer to avoid corrosion problems.
It has to be pointed out that the possibility to increase evaporation pressure and steam superheated temperature is mostly limited by economical and plant complexity aspects.

Typical flue gases outlet temperatures range from 180 to 250 °C, significantly higher than those typical for fossil fuel power plants. Values equal or below 160 °C can only be achieved in advanced plants.

The typical size of an incineration plant is smaller than that of a conventional fossil plant. The electric power generated from the combustion of MSW (usually between a few MWe up to a maximum of 70 to 80 MWe) is one or two order less than that of a conventional power plant. Moreover, the layout of the plant must be as simple as possible to reduce the investment costs.

Surface condenser with air cooling towers or air cooler condenser are generally used for steam condensation, implying high condensation pressure (between 0.2 and 0.1 bar), while 0.05 bar is the value for conventional power plant using surface condenser with open cycle.

The net electric efficiency is also negatively influenced by the high air excess necessary for the MSW combustion and, as a consequence, by the high volume of combustion products. Thus, considerable amount of exhaust flue gases are discharged raising the auxiliary power consumptions and the discharged heat.

Because of these reasons the conversion efficiency of the steam cycle for a Waste-To-Energy power plant hardly exceeds 30%, while the net electric efficiency is usually equal to 25% (see Chapter 2 Figure 18 and Figure 19).

3.1. *Steam/water cycle improvements: effects on efficiency and power output*

An initial assessment on the importance of the main steam cycle parameters must be outlined as a preliminary study, in order to understand the possibilities and the benefits of a thermodynamic cycle upgrade for a WTE power plant.

A sensitivity analysis as been carried out to analyze the effect of the most important steam cycle parameters and configuration to improve the performance, in terms of electric efficiency and power output, of a WTE plant. The parameters considered in the study are: condenser pressure (p_k); evaporative pressure (p_{ev}); steam superheated temperature (T_{SH}); steam turbine isentropic efficiency (η_{it}); exhaust MSW flue gases temperature ($T_{O,WTE}$); oxygen content in dry exhaust gases ($\%O_2$); primary and secondary air temperature (T_{air}) and cycle configuration in terms of regeneration.

The influence of each key parameter has been studied considering a fixed capacity of the WTE power plant in terms of MSW composition, mass flow rate and LHV and considering an integrated boiler as MSW combustion technology. In Table 1 is reported the composition and the LHV assumed for MSW fed into the integrated boiler.

The assumed MSW mass flow rate has been considered equal to 1 kg/s to generalize the analysis. Auxiliary consumptions of the steam cycle has been set at 100 kW (about the 3% of the produced gross electric power), the mechanical efficiency of pumps at 0.85 while the alternator efficiency was fixed to 0.9. Steam turbine outlet quality has been lower limited to 0.85 otherwise, lower steam quality values can compromise ST operation.

Being constant the input thermal power with waste, percentage increments achievable in power output, due to an upgrade of the steam cycle parameters or configuration, agree with increments in the electric efficiency value.

The thermodynamic and parametric analysis was carried out by the use of a commercial software *Gate Cycle*^{TM} [3] from General Electric for plant design and simulation. The software, solving mass and energy balance using a lumped model approach, allows to evaluate inlet and outlet conditions of each system's components and to predict plant performance.

Table 1 : MSW composition and LHV assumed for the analysis [4].

<table>
<tr><td rowspan="5">Proximate analysis (% by weight)</td><td>Humidity</td><td>28.0</td><td rowspan="5">% of volatile matter</td><td>C</td><td>52.5</td></tr>
<tr><td>Ashes</td><td>15.0</td><td>H</td><td>7.50</td></tr>
<tr><td rowspan="3">Volatile matter</td><td rowspan="3">57.0</td><td>O</td><td>38.5</td></tr>
<tr><td>N</td><td>1.30</td></tr>
<tr><td>S</td><td>0.20</td></tr>
<tr><td colspan="6">LHV=11.7 MJ/kgMSW (2.800 kcal/kgMSW)</td></tr>
</table>

WTE power plant layout and assumptions: *Base case*

The parametric investigation has been carried out considering a WTE power plant layout (namely “Base case” layout) as reported in Figure 1. In comparison with a Hirn cycle the presence of a steam turbine extraction to feed a deaerator, working at constant pressure, and a supplementary extraction to preheat the combustion primary air can be noticed.

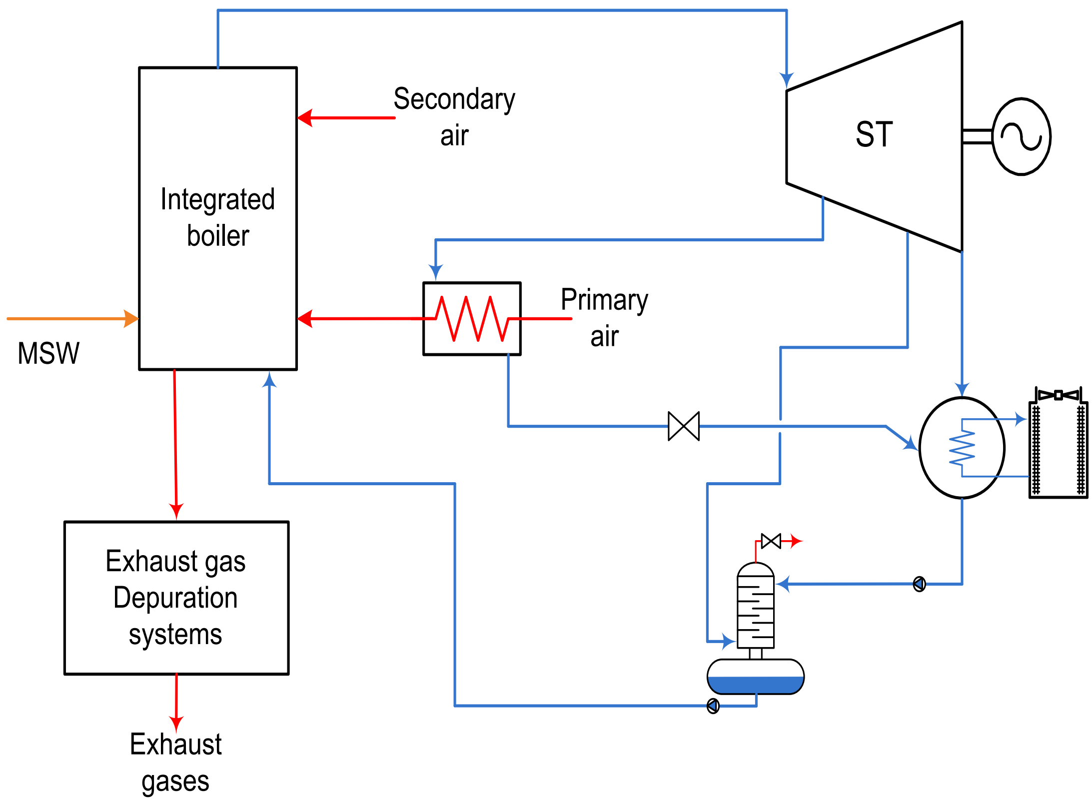


Figure 1 : Base case WTE power plant layout.

In Table 2 the steam cycle design parameters for the base case are shown: plant layout and steam design parameters assumed to represent the base case can be regarded as representative of a small-medium WTE power plant. The performance achieved are also listed in Table 2; it should be observed that ST specific work and steam mass flow are referred to the unit of MSW mass flow rate, while the obtained net electric efficiency value (25.3 %) of the base case plant is in line with previous remarks.

Table 2 : Base case steam cycle design parameters and performances.

BASE CASE	
Evaporation pressure [bar]	50
Steam superheat temperature [°C]	400
Condenser pressure [bar]	0.1
ST isentropic efficiency [-]	0.82
MSW exhaust gases temperature [°C]	190
Oxygen in exhaust dry gases [%]	9
Primary air temperature [°C]	150
Secondary air temperature [°C]	15
Deaerator pressure [bar]	2
ST extraction pressure for air heating [bar]	6
BASE CASE RESULTS	
ST net specific work [kJ/kg_{MSW}]	2717
Electric efficiency [%]	25.3
Steam mass flow [kg/kg_{MSW}]	3.55
ST outlet quality [-]	0.88

Parametric analysis results

The influence of each key parameter has been studied separately or related to others. If not otherwise specified, the parameters assume the default values shown in Table 2. Figure 2 (a) shows the influence of evaporative pressure on electric efficiency for different values of superheated steam temperature. Dotted marks evidence ST outlet quality below the established minimum. (i.e. @ 350 °C and 60 bar ST outlet quality is lower than 0.85). Figure 2 (b) reports the percentage increments on efficiency for a fixed increase in evaporative pressure value equal to 10 bar: i.e. at 400 °C passing from (p_{ev1}) 30 bar to (p_{ev2}) 40 bar the increase in efficiency is equal to 4.2% . This figure clearly highlights that higher benefits can be found for lower value of $p_{ev,}$ passing from

4% (from 30 to 40 bar) to about 1% (from 90 to 100 bar). High value of T_{SH} slightly narrow the achieved increase.

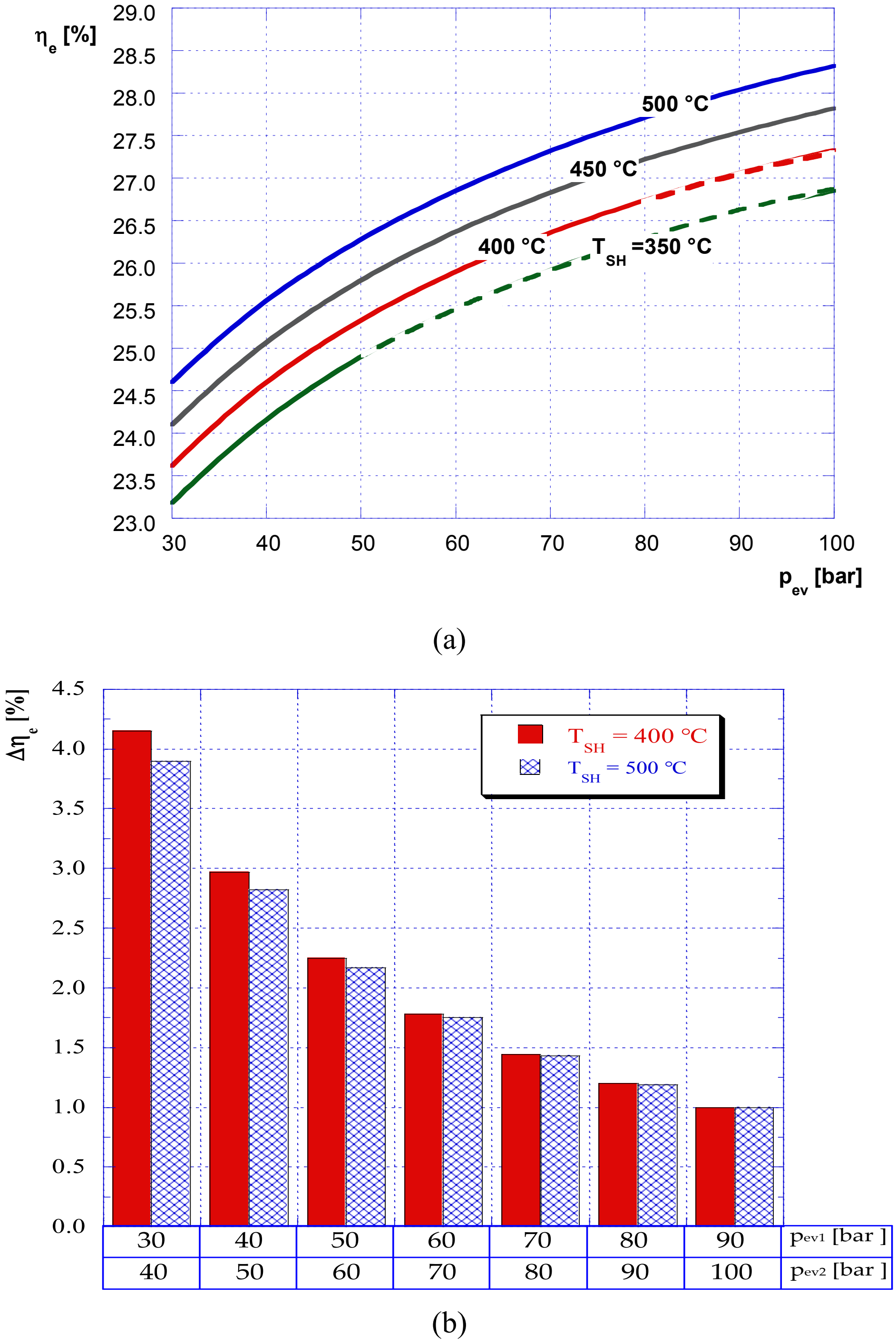
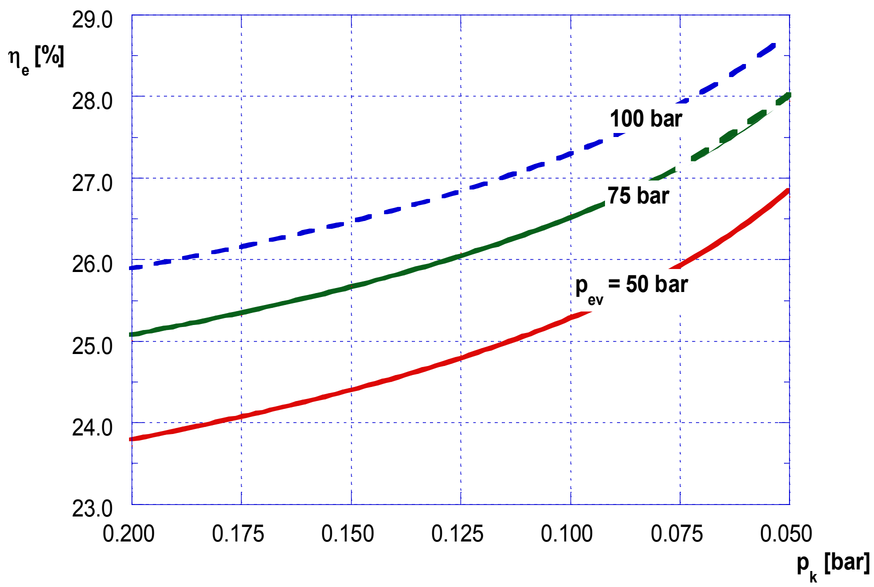
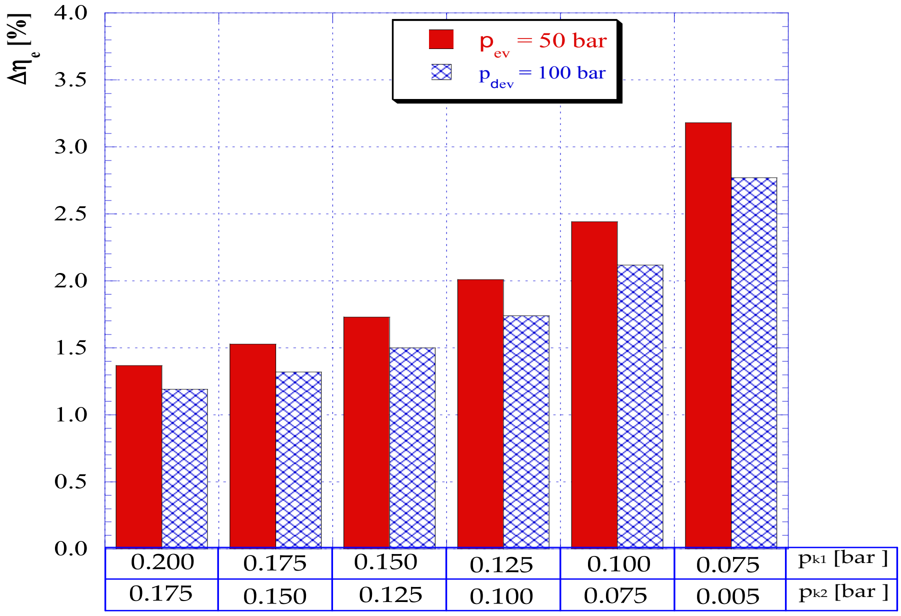


Figure 2 : Electric efficiency (a) and percentage increments (b) as a function of evaporative pressure for different values of steam superheated temperature.

Figure 3 (a) shows the effect of condenser pressure on electric efficiency for different values of evaporative pressure. The possibility to reduce the p_k have considerable effects on the steam cycle efficiency. It should be noted that lower p_k values (i.e. 0.05 bar) can be achieved only by the use of surface condenser with an open cycle; this possibility is strictly limited by the presence of a natural water source close to the WTE power plant. Figure 3 (b) reports the percentage increments on efficiency for a fixed decrease (equal to 25 mbar) in condenser pressure value. Figure clearly highlights that higher benefits can be found for lower value of p_k passing from more than 3% (from 75 to 50 mbar) to about 1% (from 2 to 1.75 mbar). Percentage gains increase if the evaporative pressure is lower.



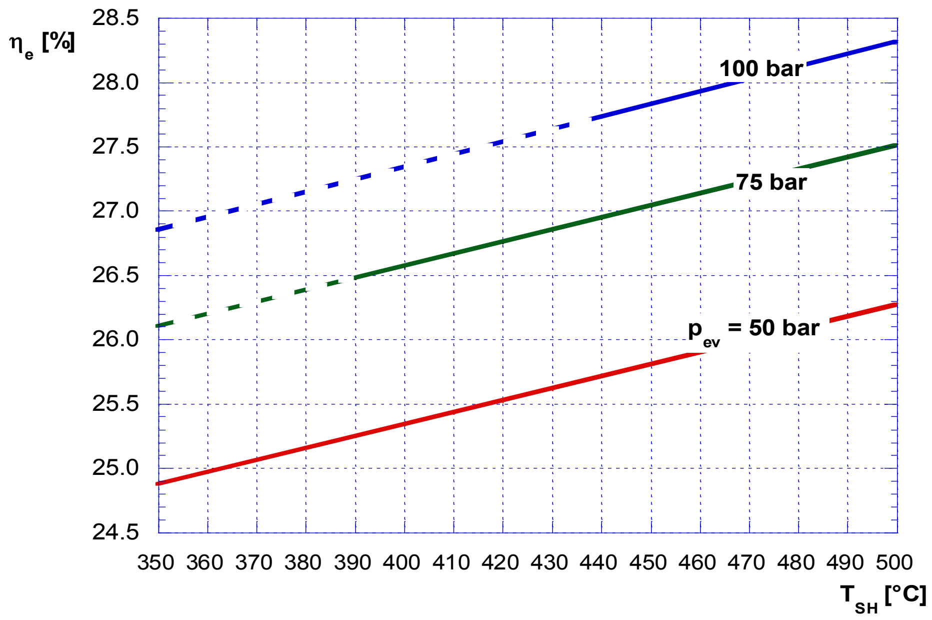
(a)



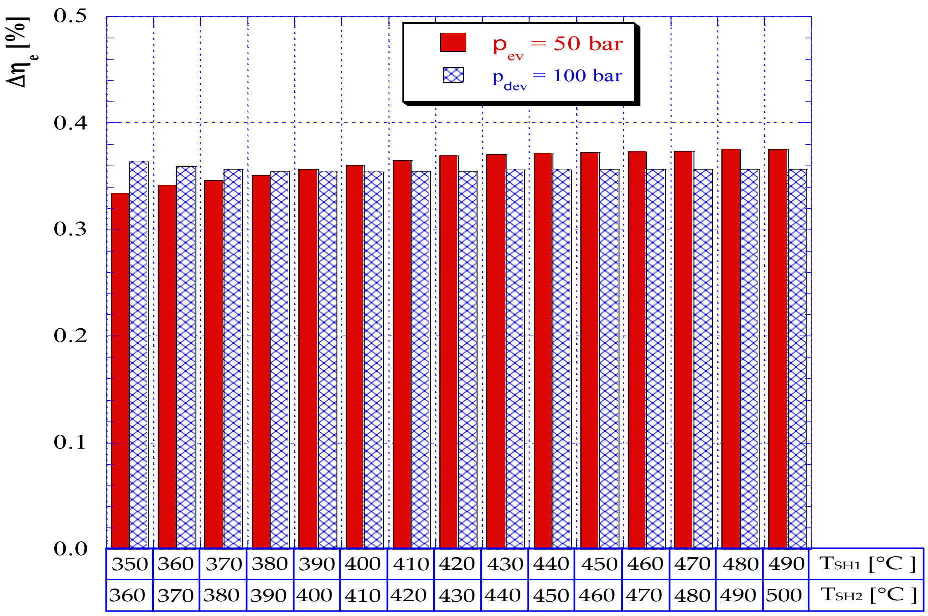
(b)

Figure 3 : Electric efficiency (a) and percentage increments (b) as a function of condenser pressure for different value of evaporative pressure.

The increase in superheated steam temperature (T_{SH}), as obvious, has positive effect on the efficiency of the steam cycle. The electric efficiency trend are shown in Figure 4 (a) for three different value of evaporative pressure. Percentage increments on electric efficiency are shown in Figure 4 (b) for a fixed steam superheated temperature increment, equal to 10°C. The slope of the curve is slightly influenced by T_{SH} actual value (as evaporative pressure is equal to 100 bar), on the contrary, for lower p_{ev} value percentage increment on efficiency increases with the increase of T_{SH} actual value.



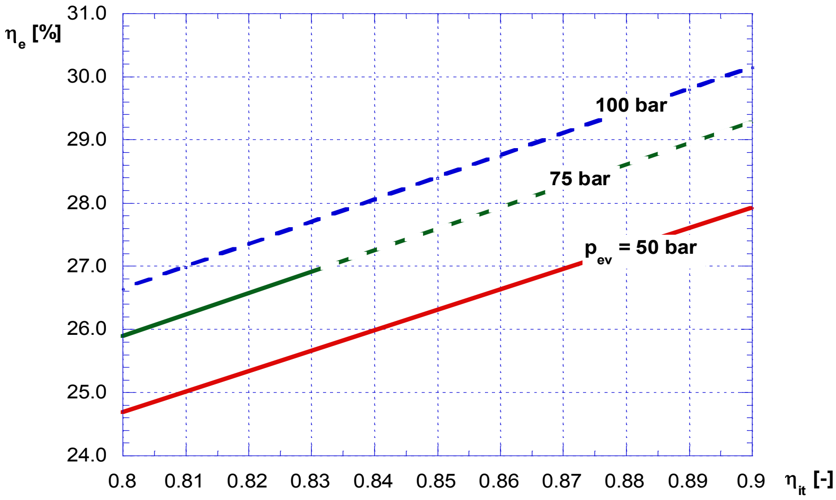
(a)



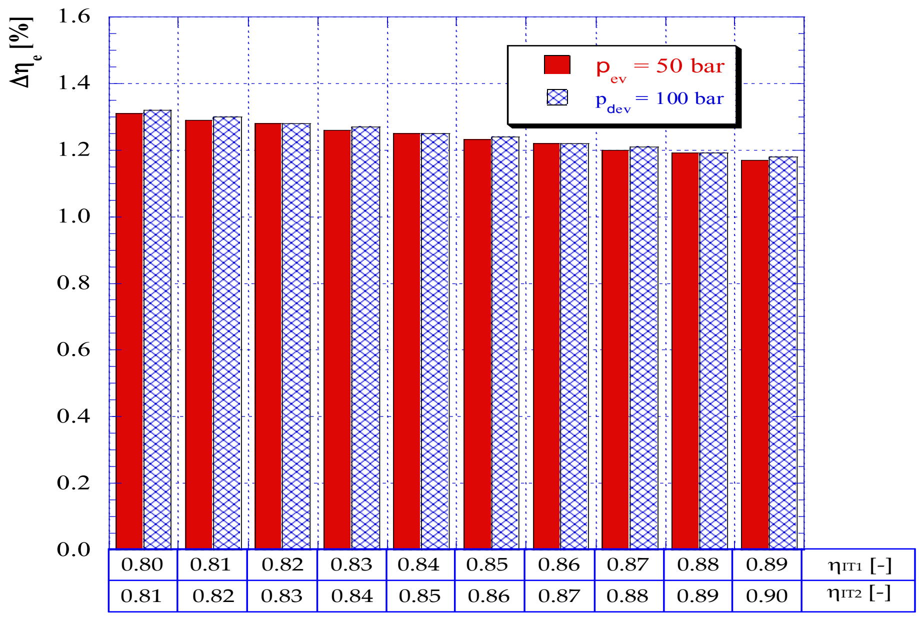
(b)

Figure 4 : Electric efficiency (a) and percentage increments (b) as a function of steam superheated temperature for different value of evaporative pressure.

High value of ST isentropic efficiency (η_{it}) are also related to the size of the steam turbine and, as a consequence, to the plant's size. The electric efficiency related to ST isentropic efficiency is shown in Figure 5 (a) for three different values of evaporative pressure. Even for this parameter, the percentage increments on electric efficiency are shown in Figure 5 (b) for a fixed η_{it} increment, set equal to 0.01; this figure points out that increments are not influenced by the evaporative pressure value.



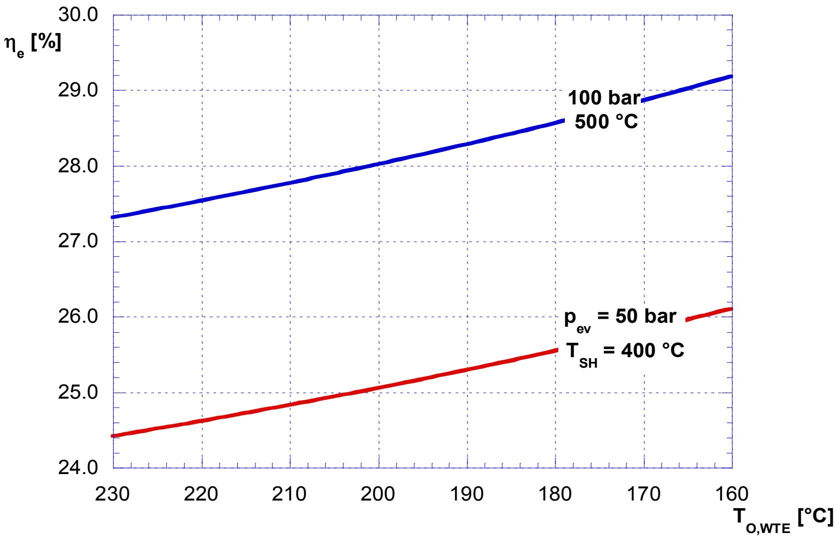
(a)



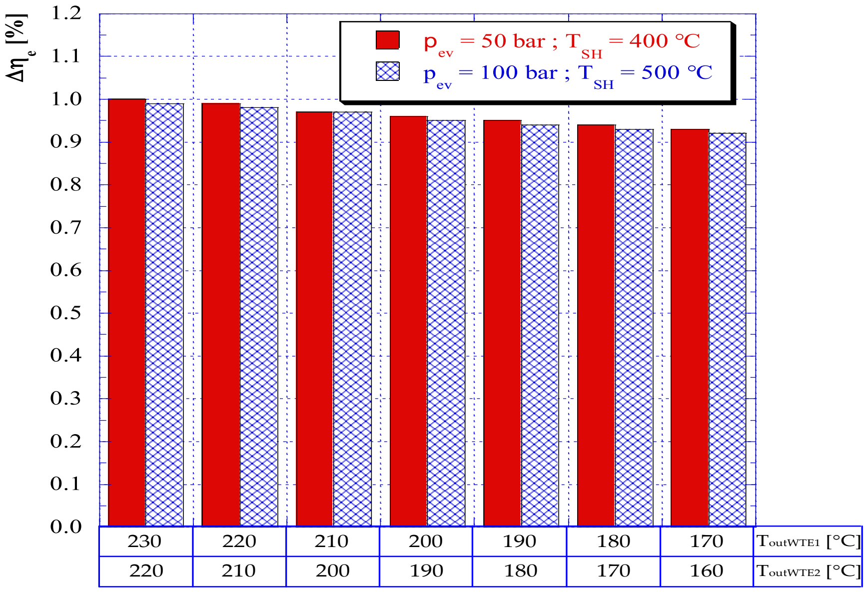
(b)

Figure 5 : Electric efficiency (a) and percentage increments (b) as a function of ST isentropic efficiency for different values of evaporative pressure.

In Figure 6 (a) and (b) is shown respectively the electric efficiency performance as function of the flue gases outlet temperature, and the percentage increments gain for a fixed decrease in $T_{O,WTE}$ equal 10 °C. Increments are almost constant, not dependent on evaporative pressure and steam superheated temperature. It should be noted that, in the considered range, the lowest value (160 °C) is typical of advanced WTE power plants.



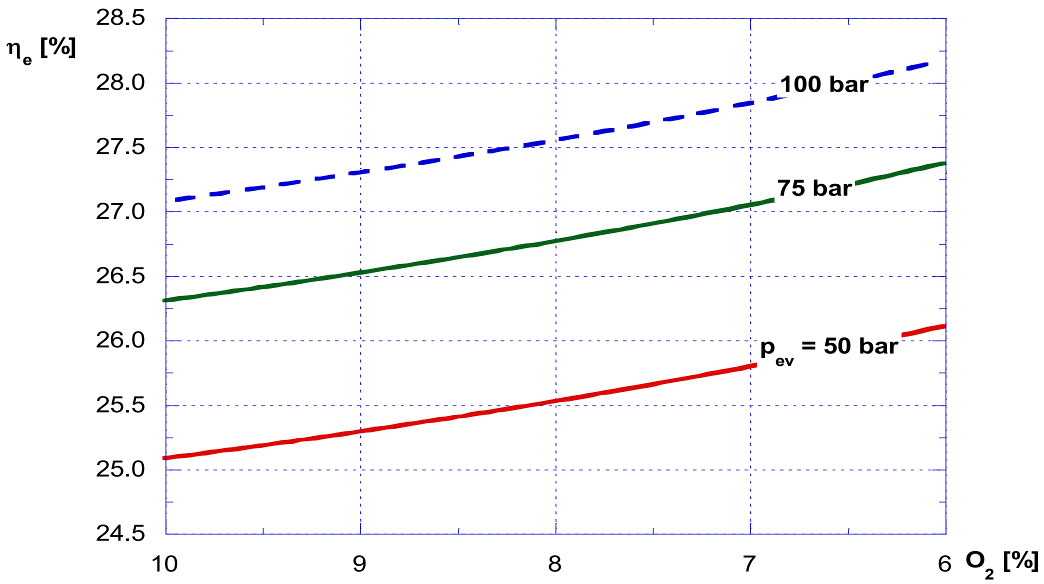
(a)



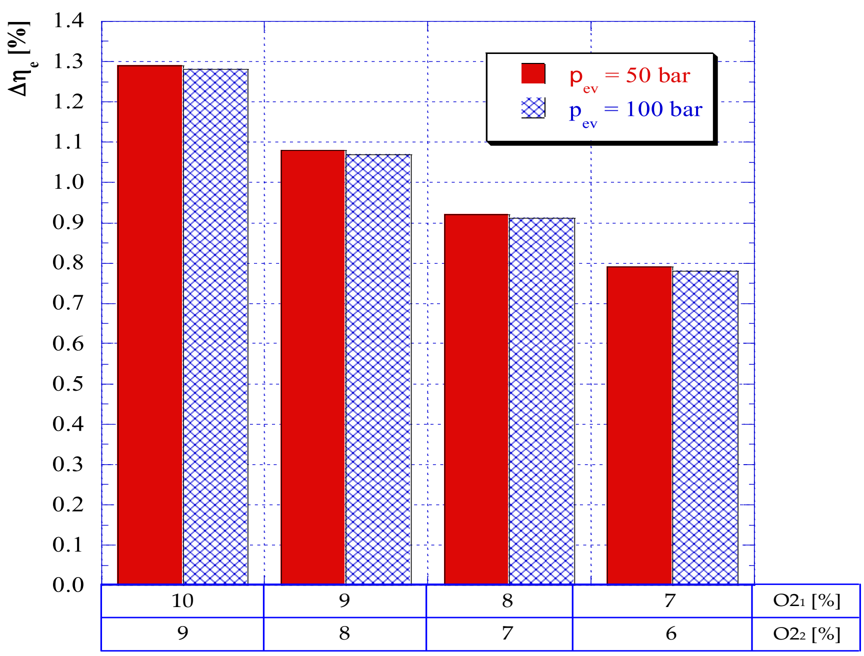
(b)

Figure 6 : Electric efficiency (a) and percentage increments (b) related to exhaust gases outlet temperature for different value of evaporative pressure.

Even a reduction of oxygen content (vol. $\%O_2$) into the dry exhaust gases at WTE boiler outlet causes a reduction in the amount of heat lost to the stack. Benefits on efficiency are shown in Figure 7 (a) for two different values of evaporative pressure. Figure 7 (b) emphasizes that the percentage gain for a fixed percentage point of decrease are function of $\%O_2$ actual value, but not dependent on p_{ev} value.



(a)



(b)

Figure 7 : Electric efficiency (a) and percentage increments (b) as a function of volume oxygen content in exhaust dry gases for different values of evaporative pressure.

The influence of secondary air temperature on electric efficiency has been also analyzed. A ST extraction pressure of 6 bar has been considered feeding both primary and secondary combustion air heater; percentage increase in electric efficiency of more than 1.6% (25.7% of electric efficiency) has been found by increasing secondary air temperature from ambient to 150°C.

<u>WTE power plant layout and assumptions: *Advanced case*</u>

The layout of the steam cycle has a relevant impact on WTE plants performances. To better analyze the influence of a regenerative steam cycle on electric efficiency a more complex cycle configuration has been considered (namely Advanced case). This layout is shown in Figure 8. In this configuration, besides the presence of the deaerator, a regenerative steam turbine extraction has been introduced, to heat the feeding water at the deaerator exit; moreover, the preheating of the secondary combustion air has been added too. To reduce the steam turbine complexity, the extraction pressure is unique both for air and water preheating. Nevertheless this unique ST extraction pressure allows the achievement of appreciable values of electric efficiency.

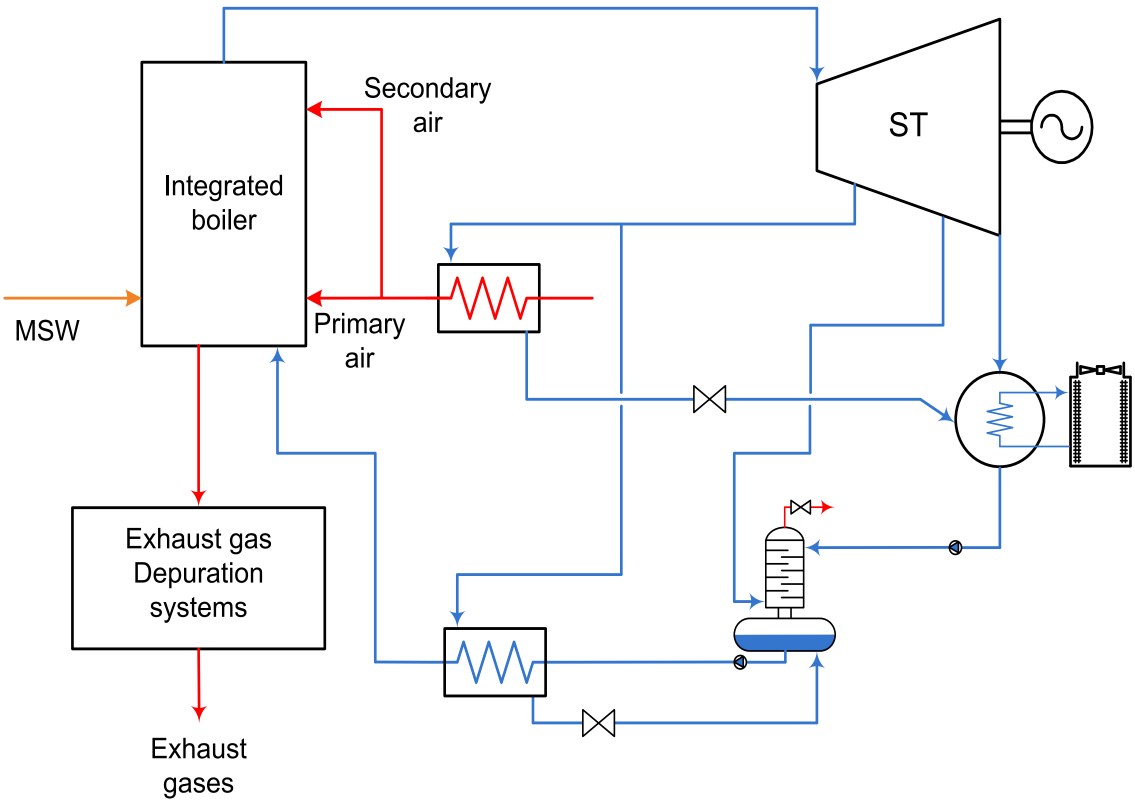


Figure 8 : Advanced case WTE power plant layout.

The steam cycle design assumptions and achieved performances, for the advanced case, are listed in Table 3. As a consequence of plant layout complexity and steam cycle parameters upgrade, the electric efficiency achieved, above 32%, is representative of a high performance WTE power plant.

Table 3 : Advanced case steam cycle design parameters and performances.

ADVANCED CASE	
Evaporation pressure [bar]	100
Steam superheated temperature [°C]	500
Condenser pressure [bar]	0.1
ST isentropic efficiency [-]	0.85
MSW exhaust gases temperature [°C]	160
Oxygen in exhaust dry gases [%]	7
Primary and secondary air temperature [°C]	150
Deaerator pressure [bar]	2
ST extraction pressure for air and feed water heating [bar]	6
ADVANCED CASE RESULTS	
ST net specific work [kJ/kg_{MSW}]	3496
Electric efficiency [%]	32.6
Steam mass flow [kg/kg_{MSW}]	3.71
ST outlet quality [-]	0.87

The influence of the advanced case layout (ST extraction to heat feed water and secondary combustion air) has been analyzed with base case assumptions; the achieved electric efficiency value is 26.7% with an increment, with reference to the base case value, of more than 5.5%.

In Table 4 further results of the analysis in terms of electric efficiency are shown for both base case and advanced case layouts. Results account the electric efficiency value achieved with the upgrade of only one parameters (main diagonal) or with the change of two different parameters. Spotlight cells present a ST outlet quality below 0.85.

Thus, these results point out the influence of the layout configuration, switching from base to advanced layout, and how upgrading key parameters affects performance of waste conversion efficiency.

Table 4 : Electric efficiency for both, the base case and advanced case layout.

Base case layout / **Advanced case layout**	p_{ev} 50→100 bar	T_{SH} 400→500 °C	η_{it} 0.82→0.85	$T_{out\ WTE}$ 190→160 °C	$\%O_2$ 9→7%
p_{ev} 50→100 bar	**27.3** / **29.3**	**-**	**-**	**-**	**-**
T_{SH} 400→500 °C	**28.3** / **30.3**	**26.3** / **28.1**	**-**	**-**	**-**
η_{it} 0.82→0.85	**28.4** / **30.4**	**27.3** / **29.1**	**26.3** / **28.1**	**-**	**-**
T_{out_WTE} 190→160 °C	**28.1** / **30.1**	**27.0** / **28.8**	**27.1** / **28.9**	**26.1** / **27.8**	**-**
$\%O_2$ 9→7%	**27.9** / **29.7**	**26.8** / **28.4**	**26.8** / **28.5**	**26.5** / **28.1**	**25.8** / **27.5**

3.2. Conclusive remarks

The carried out analysis, identifying two different WTE power plants, namely a "base case" and an "advanced case", highlights the influence of both the main steam cycle parameters and layout configuration on the electric efficiency.

Obtained results show that different strategies can be performed to increase the electric efficiency of a small-medium WTE power plant. It has to be pointed that some strategies (like increasing evaporative pressure and steam superheated temperature) may involve specific solutions to protect the integrity of the waste fired boiler; on the contrary, other strategies are simpler, more economic and can be applied without considerable changes.

This study can be considered a useful preliminary tool to evaluate and compare all the configurations and the possible strategies to improve the efficiency of a WTE plant.

As a conclusion of the carried out analysis, Figure 9 identify an area where, depending, on steam cycle parameters, the thermodynamic efficiency of a WTE power plant can be obtained. The area is bordered between two curves: the lower one stands for low steam cycle key parameters, the upper stands for upgraded values. Points (from A to G) show the thermodynamic efficiency achieved with intermediate steam cycle values (e.g. point D with a steam superheated temperature equal to 450 °C, a condenser pressure of 0.07 bar and a ST isentropic efficiency of 0.845 reach a η_{th} equal to about 32.5 %). It can be pointed out that, depending on the cycle key parameters, efficiency can have a wide variation passing from 26 % to 36 %.

As suggested from the study, also the steam water cycle layout have an important impact on the steam cycle thermodynamic efficiency. As evidenced from Figure 10, feed water preheating (regenerative cycle, black curve) has considerably effect on steam cycle thermodynamic efficiency.

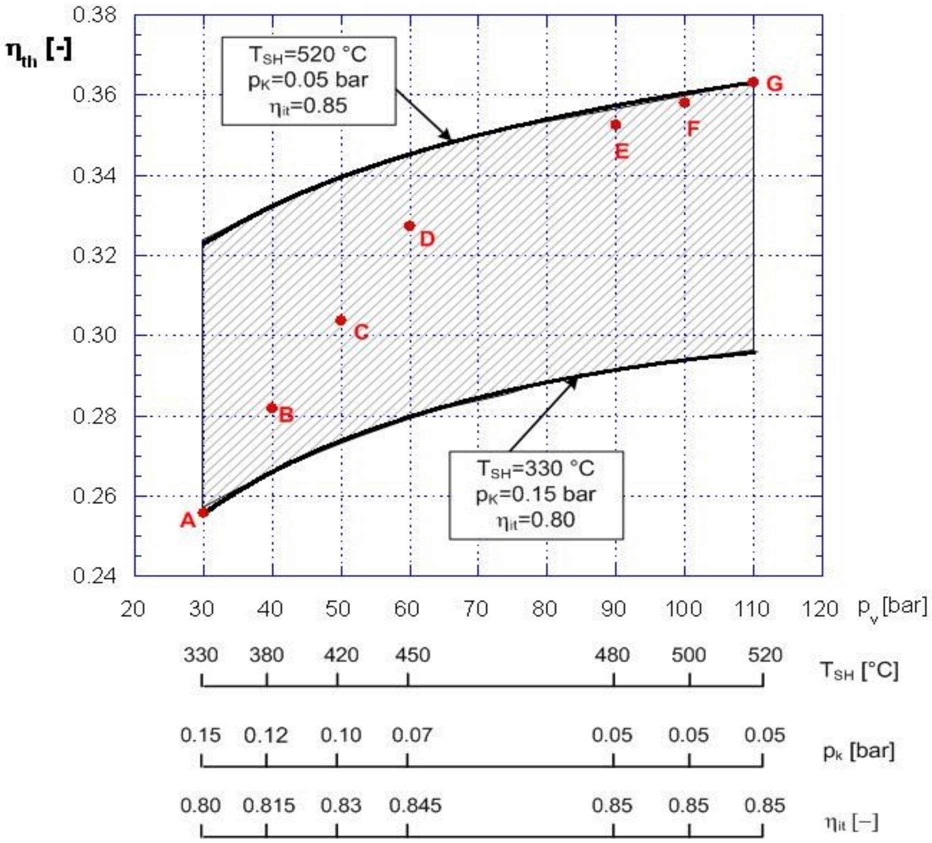


Figure 9 : steam cycle thermodynamic efficiency function of key cycle parameters.

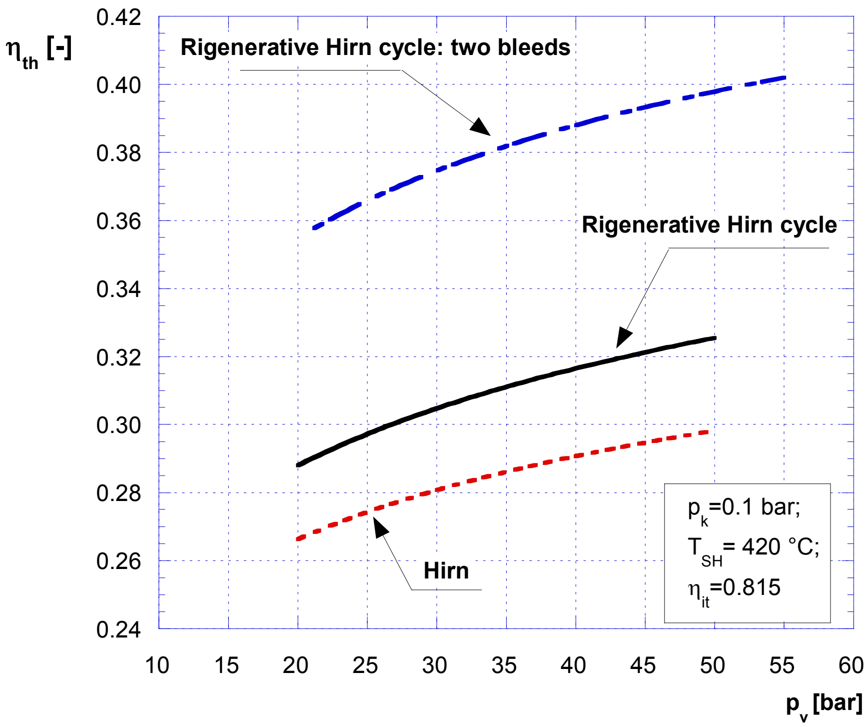


Figure 10 :Thermodynamic efficiency for different steam cycle configurations.

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4. WTE-GT integrated power plant

4.1. The Hybrid Combined Cycle concept

To better understand the concept of WTE and Gas Turbine (GT) integrated power plant, a general overview on hybrid combined cycle is presented in this section.

Even if a standardized terminology to address the thermal connection between different systems does not exist, Hybrid Combined Cycle (HCC) is the definition commonly used to address it. The Hybrid Combine Cycle (HCC) concept is not new. The first suggestions for such a thermal connection between a Topping Cycle (TC) and a Bottomer Cycle (BC) originate as long back as the idea for a Combine Cycle Gas Turbine itself.

The term “Topping Cycle” (TC) addresses the power cycle of any heat engine that accepts thermal energy at high temperature level and whose remaining exhaust heat is exploited in another cycle at a lower temperature level. Typical examples, well developed and widely used today as TC, are GTs and Internal Combustion Engines (ICE). TC systems, typically, utilise high-grade gaseous or liquid fuels.

The term “Bottomer Cycle” (BC) refers to any power cycle whose heat supply comes in the form of rejected heat from another power cycle. The BC itself rejects heat at the lowest possible temperature level. A typical example of a BC is the well known Rankine cycle, working with water/steam or any other two-phase fluid. Power units of this type can have their own fuel input, while serving as bottoming cycles to a topping engine.

The term Hybrid Cycles denotes specifically “dual-fuel” combined power cycles where different fuels are used for the topping and bottoming cycle [1]. This is one of the big advantages of the HCC, namely the possibility to utilise low-grade fuels (solid fuels) in the bottoming cycle, together with exploiting the full potential of high-grade fuels (gaseous or liquid) in the topping cycle.

The BC, then, has its own individual combustion chamber, in which the bottoming fuel is fired. Two basic types of hybrid dual-fuel combined cycle arrangements are possible (see Figure 1) depending on TC purpose:

- Windbox repowering, where the TC exhaust, with or without pre-cooling, is supplied to the bottoming boiler and used as combustion air for firing the BC fuel.
- Steam/water side integrated HCC, where thermal energy from the TC exhaust is utilised for feedwater preheating and/or steam superheating and/or additional steam generation parallel to the BC.

Windbox repowering cycles can be further divided in two main types: hot windbox and cold windbox. In the hot windbox type GT exhaust is fed directly into the BC boiler. In the cold windbox type the GT exhaust is first cooled down to a lower temperature level (by various options as, for example, supplying heat for parallel steam generation or feedwater preheating), after which is fed to the BC boiler. Cold windbox arrangements actually allow for features typical to both the windbox repowering and steam/water side integrated hybrid cycles.

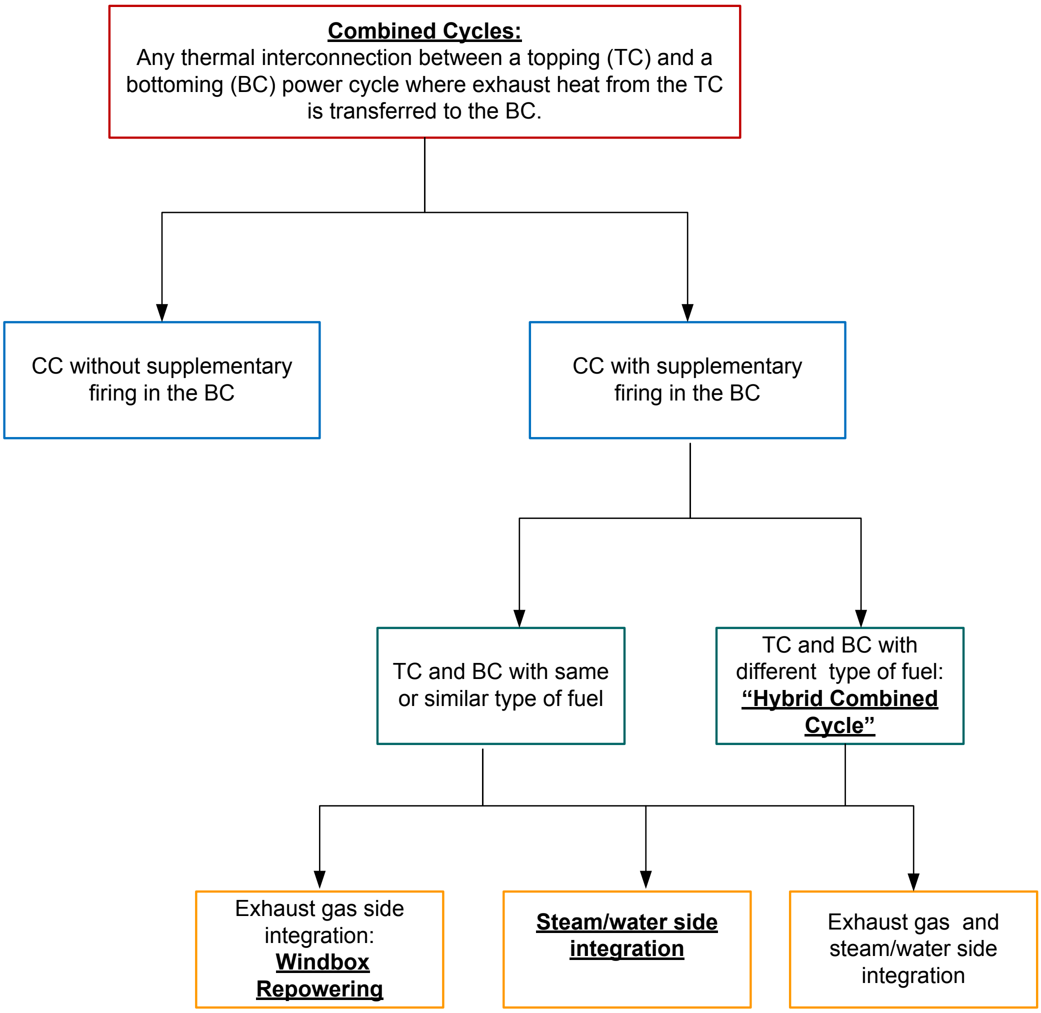

Figure 1 : Schematic diversification of combined cycle.

4.2. *WTE-GT steam/water side integration*

Focus on steam/water side integration, the WTE plant basically acts as an additional source of saturated steam and, if convenient, hot water for the combined cycle. All the equipment for power production is concentrated in the "island" comprising the combined cycle. The waste treatment island comprises the equipment for waste handling, waste combustion and flue gas treatment. Figure 2 schematically shows a possible WTE-GT steam/water side integrated power plant layout.

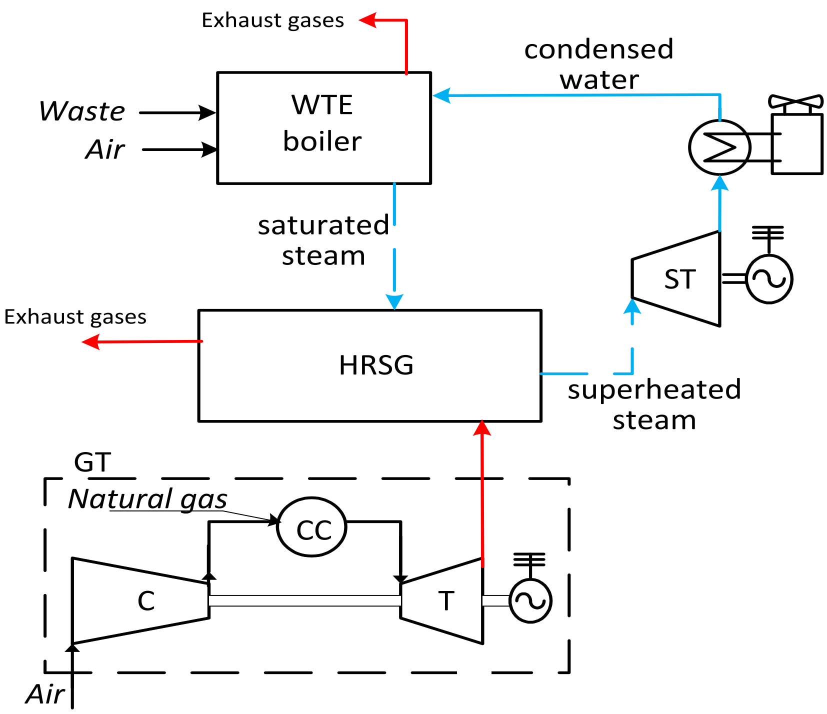

Figure 2 : Schematic of the steam/water side integration between a WTE and a GT.

The waste treatment island is basically the same as current grate combustor/steam Rankine plants, except for the lack of:

- Steam turbine, electric generator, condenser and feedwater pump, because their function is carried out within the steam section of the combined cycle;

- Steam superheater, because the whole steam flow can be conveniently superheated in the HRSG by the clean combustion products of natural gas.

Being almost exempt from corrosion, the HRSG can take steam to temperatures typically reached in fossil fuel power plants 520-560°C; in turn, higher steam temperatures allow higher evaporation pressures without undue liquid fractions in the last stages of the steam turbine. Also, generation of saturated steam outside the HRSG (in the WTE boiler) gives a more favourable temperature profile (smaller ΔT between gas and steam or water), whereby lower irreversibility and higher efficiencies.

The flue gas treatment section and thus pollutant concentrations at the stack can be identical to those of conventional WTE plants; this also applies to emissions specific to the amount of MSW, because the integration causes no dilution of the combustion products. No dilution also means no penalties on the cost for flue gas treatment.

Increasing steam temperature externally in the Rankine cycle will improve the efficiency of electricity generation for MSW fired power plants. This can be achieved by MSW and natural gas hybrid combined cycles that involve two different thermodynamic cycles with two types of fuel. In the combined cycles, the topping cycle consists of a gas turbine, while the bottoming cycle, driven by low quality fuel MSW, is a steam cycle. In a dual-fuel combined cycle system, there must be a well-designed thermal link between the toping cycle and the bottoming steam cycle. The integration have to provide thermodynamic and operating advantages for both the topping cycle and the bottoming cycle. Generally, steam superheating by turbine exhaust heat is viable. This arrangement can substantially increase the efficiency of MSW energy conversion while avoiding the described corrosion problems (see Chapter 2).

The potential advantages of a WTE-GT plant match, with reference to a stand alone WTE, can be summarized as follow [2]:

- *increasing of the maximum temperature of the stem cycle* transferring superheater from the WTE to HRSG, the most problematic component for what concern high temperature corrosion;
- *reduction of the HRSG irreversibility* caused by high mean temperature differences, moving the generation of saturated steam from the HRSG to the WTE section;
- *increase steam turbine efficiency* increasing its size: the steam turbine serving both the WTE section and the HRSG is greater than that typically used in conventional WTE plants;

Moreover, from an economic point of view, the integrated system leads to further benefits such as the reductions of:

- *maintenance costs of the WTE section, due to elimination of the superheater heat exchanger;*
- *capital costs because a significant number of equipment and services are shared in the integrated plant.*

4.3. WTE-GT windbox integration

Windbox repowering is accomplished by installing a gas turbine to provide extra power and direct its exhaust into the original boiler's windbox with or without precooling. Figure 3 shows a schematic WTE-GT hot windbox integrated layout.

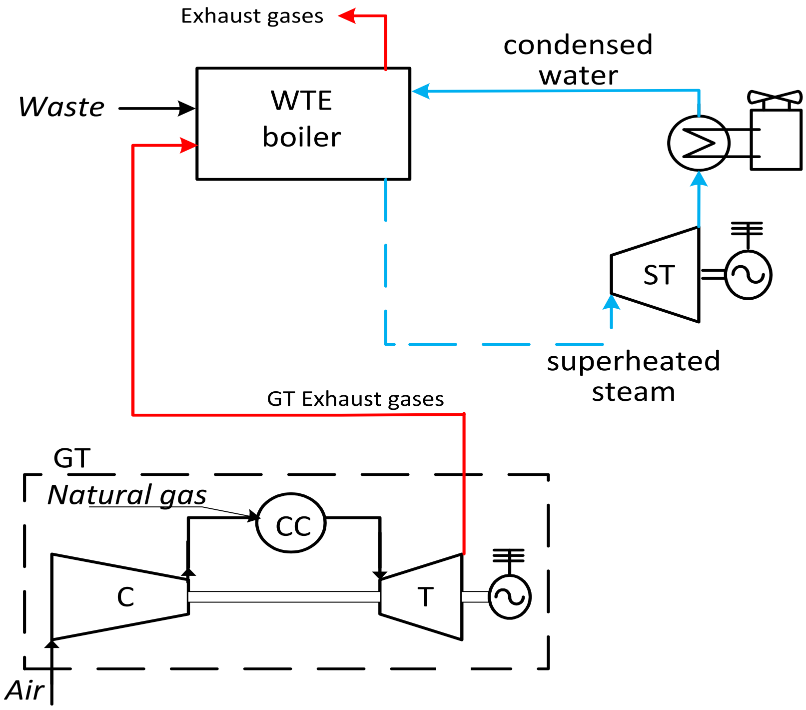

Figure 3 : Schematic of a hot windbox integration between a WTE and a GT.

Gas turbines can be used for retrofitting existing steam plants (hot windbox repowering). GT are operated with large amounts of excess air. Thus, the exhaust of gas turbines contains a high concentration of oxygen (vol 14 % – vol 16 %) and the exhaust temperature is also high (480–600 °C). This makes a gas turbine well-suited for integration with an MSW boiler in the combined cycles [3]. Moreover, modern gas turbines are generally optimized with respect to maximum power density (output per unit air flow) rather than efficiency. This coincidentally meets well the requirements of optimum efficiency of the combined cycle plants.

The hot windbox repowering has the highest degree of technical complexity of all the combustion-turbine-based repowering options. The air heaters may need to be modified based on the revised air and gas flows, and the ductwork must be upgraded to accommodate the higher temperature and larger volume of air. The furnace burners must be modified or replaced because of the lower oxygen content of the flow from the combustion turbine exhaust. Furthermore, the lower oxygen content of the combustion air will change the heat release profile in the furnace and some derating of the boiler or a redesign of the convective parts of the furnace may be necessary. Other necessary modifications can include bypass ducts for admitting variable amounts of combustion turbine exhaust, a steam air heater to allow independent operation of the existing boiler when the combustion turbine is not available, an induced draft fan to reduce the back pressure on the combustion turbine, and a combustion turbine bypass stack for unit start-up [4].

A variant to the hot windbox repowering approach includes a HRSG or heat exchangers to reduce the temperature of the combustion turbine exhaust and, for example, produce additional steam.

4.4. MSW-fired bottoming cycles: literature overview

Hybrid cycles with MSW as bottoming fuel, during the last 15 years, have become a very attractive topic of scientific and industrial research. Due to the growing interest in energy utilisation from MSW, the waste-to-energy conversion efficiency is becoming an important issue. Interest in improving the electrical efficiency of WTE, with cost-effective methods, is based on the fact that MSW-fired power cycles, as already detailed in Chapter 2 and Chapter 3, are typically characterized be very modest steam cycle parameters and consequently very low electrical efficiencies (far below 30%). Improved electrical efficiency from MSW-fired steam units can be achieved by tolerating higher corrosive rates for increased superheat temperatures, and consequently higher maintenance costs.

A more cost-effective method for increasing the electrical efficiency without new materials or expensive investments is the possibility to superheat the steam in a separate heat exchanger by combusting cleaner fuels. The incorporation of the MSW boiler as a BC into a HCC, where the topping exhaust provides superheating, can substantially improve the MSW conversion efficiency into electrical energy using only conventional technology. This is relevant to any steam cycle with low steam parameters.

One of the first suggestions focused on external superheating in a natural gas burner has been presented by Eber et al. [5]. Results of the study, where energy input form natural gas has been limited to 25% of the total energy input, highlighted that from a thermodynamic as well as economic point of view the solution is viable.

First idea of integrating a MSW combustor in HCC with natural gas fired topping GT have been put forward by Lowry and Martin [6] and Wiekmeijer [7]. Lowry and Martin evaluate a simplified arrangement of a GT whose exhaust gas superheats the MSW generated steam. On the other side, Wiekmeijer's study focuses on a more complicated steam/water side integrated cycle arrangement with economiser and final superheater in the HRSG behind the GT [7].

Several Japanese authors studied possible arrangements and developments of HCC plants for waste fired combustor. Terasawa and Ogura [8] shortly mention the HCC alternative in their evaluation of systems for rationalization of waste incineration practices. Ito et al. [9] evaluated the economic and energy characteristics of a MSW boiler topped by a GT in CHP mode. Sue [10] suggests application of a steam-injected GT as TC in a MSW fired power unit. Otoma et al. [11] firstly performed a life-cycle analysis for the general case of MSW-based electrical production in Japanese conditions (with very low steam parameters and electrical efficiency of 15%). Results show that electricity production from this basic waste-to-energy plant is 9.5 times higher than the energy involved in supporting activities over the whole life-cycle of the facilities. Then, Authors evaluate also two options for topping their base-case boiler with a GT. Holmgren [12] focuses on a thermodynamic and sensitivity analysis of the overall performance of three different HCC configurations of a GT combined with a MSW-fired boiler in CHP mode. Results of the comparative analysis highlight the advantages of the hybrid configurations in comparison with separate plants (existing single-fuel MSW-fired, CHP plants and GTCC plants). Korobitsyn et al. [13], as shown in Figure 4, Figure 5 and Figure 6, examine three different MSW and GT HCC configurations. All feature final superheating of the steam by the GT exhaust gas. Results of the proposed cycles, Figure 7, clearly presents the HCC electrical efficiency compared to the average efficiency of two individual single-fuel plants (one simple MSW-fired steam cycle with efficiency equal to about 25% and one natural gas fired GTCC). Moreover, Korobitsyn and co-authors were the first to suggest definitions to evaluate the electrical efficiency attributable to the MSW fuel within the HCC.

Consonni' s study [2] also proposes MSW cycle configurations with a GT as TC, where all superheating is done in the HRSG behind the gas turbine. His work focuses on steam/water side integrated HCC layouts, mentioning the advantage of keeping the GT exhaust gas flow separated from the MSW exhaust. Representative configurations of the proposed single pressure level and two pressure level integrated plants are shown respectively in Figure 8 a) and b). Consonni also emphasized the importance of estimating the efficiency attributable to MSW within the HCC. His results (Figure 9) show that electrical efficiency based on MSW can reach close to 36% net, with an increase compared to simple MSW steam Rankine cycle equal to about 1.5 times. Consonni also recognizes the effects of scale in his efficiency comparisons and presents a thorough cost calculation and economic analysis of MSW energy utilisation plants for European conditions.

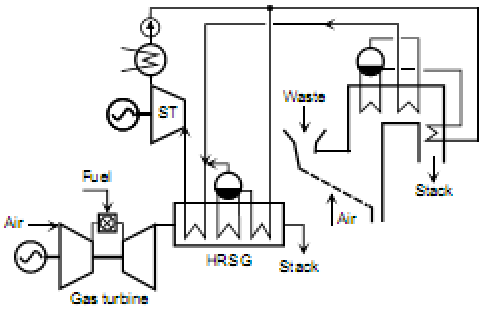

Figure 4 : HRSG incineration boiler parallel configuration with an exhaust bypass (Case 1) [13].

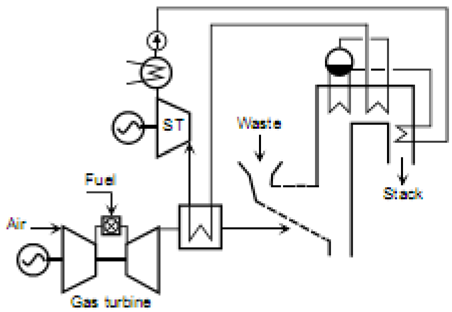

Figure 5 : HRSG incineration boiler parallel configuration with an exhaust bypass (Case 3) [13].

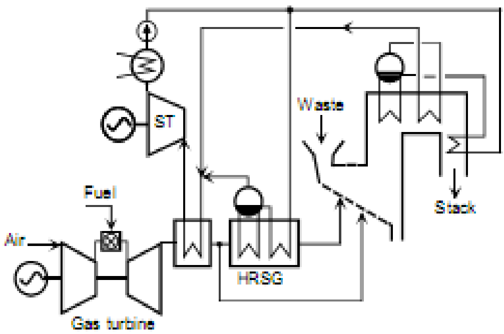

Figure 6 : HRSG incineration boiler parallel configuration with an exhaust bypass (Case 4) [13].

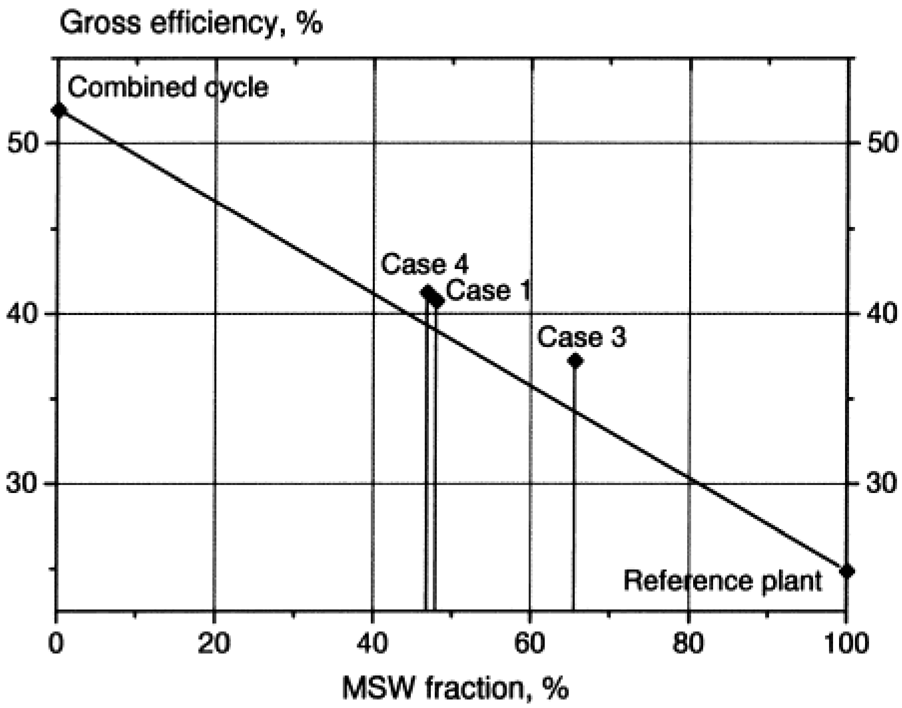

Figure 7 : The MSW fraction in the total fuel input versus gross efficiency [13].

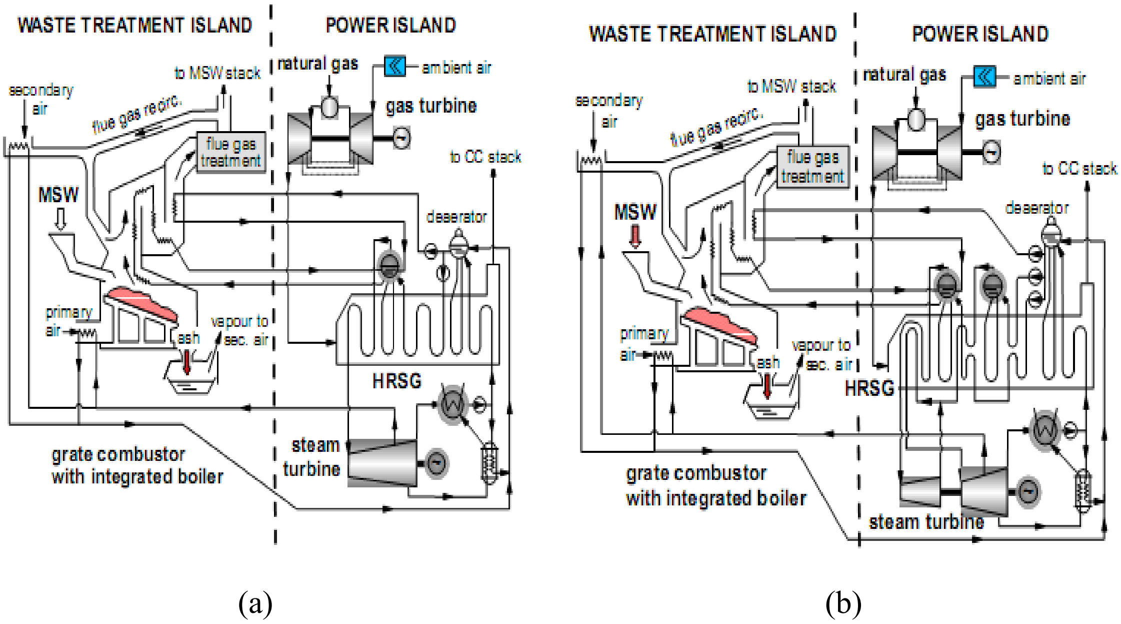

Figure 8 : Schematic of an integrated combined cycle WTE plant comprising a single level (a) and a two-level steam cycle with reheat (b) [2].

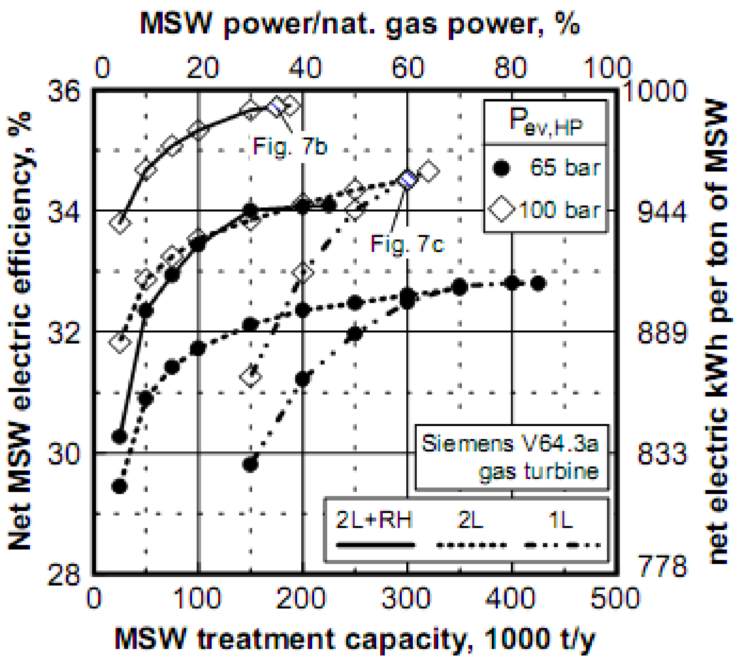

Figure 9 :Net electric efficiency for MSW vs MSW treatement capacity for integrated WTE-CC plants [2].

4.5. Existing WTE-GT integrated power plant

The diffusion of integrated WTE-GT power plant is really poor, to confirm that this is a new and still under research and investigation solution. Two are the operative WTE-GT power plants in Europe, in Spain and in the Netherlands. Another operative plant is located in Japan. By the way, only the Spanish power plant has been designed with the idea of integrated power plant.

4.5.1. Zabalgarbi WTE-CC power plant: the SENER solution

The integrated power plant of Zabalgarbi has been built by Constructions Industrielles de la Méditerranée (CNIM) as expert in Municipal Solid Waste with energy recovery construction and operation, and SENER Ingeniería y Sistemas, S.A. as Engineering Company for the designing and development of the project in Bilbao, Spain (see Figure 1) [16].

Plant construction started in September of 2001 to be completed approximately 36 months after. Start-up of the plant was in 2004.

The WTE-CC power plant is based on "steam/water side" integration. It utilizes the SENER-2 high efficiency thermodynamic energy cycle, designed and patented by SENER to minimize corrosion problems which are normal in conventional plants. As stated into plant summary report, the aim of the project is to promote a new concept for generation of electricity from MSW to allows efficient power generation, efficient waste disposal and low environmental impact.

Figure 1 : A picture of Zabalgarbi integrated WTE-CC power plant [16].

The process and a schematically layout of the plant are shown in Figure 2 while technical data are summarized in Table 1.**Errore. L'origine riferimento non è stata trovata.**

Main components in the system are: a MSW furnace with boiler for the generation of saturated steam, a Gas Turbine generator, a Heat Recovery Steam Generator (HRSG) working at 100 bar (equipped with auxiliary burners), a steam turbine generator, a main condenser and an auxiliary one and flue gas cleaning system.

The saturated steam goes out of the main boiler (located into WTE section) at 350 °C and 100 bar. To protect the boiler against corrosion, all exposed furnace tubes are coated with Inconnel 625.

Saturated steam gets in the HRSG where is heated up to 540 °C with heat coming from the exhaust of the GT and auxiliary burners. Superheated steam generated at 540 °C and 100 bars goes into the high pressure steam turbine to be expanded. Steam comes out of the high pressure Steam Turbine at a lower pressure (25 bar) goes again to the HRSG to be reheated at the same temperature (540 °C) and finally goes to the low pressure Steam Turbine. After condensation and before entering furnace's boiler water is preheated with a low pressure steam turbine extraction.

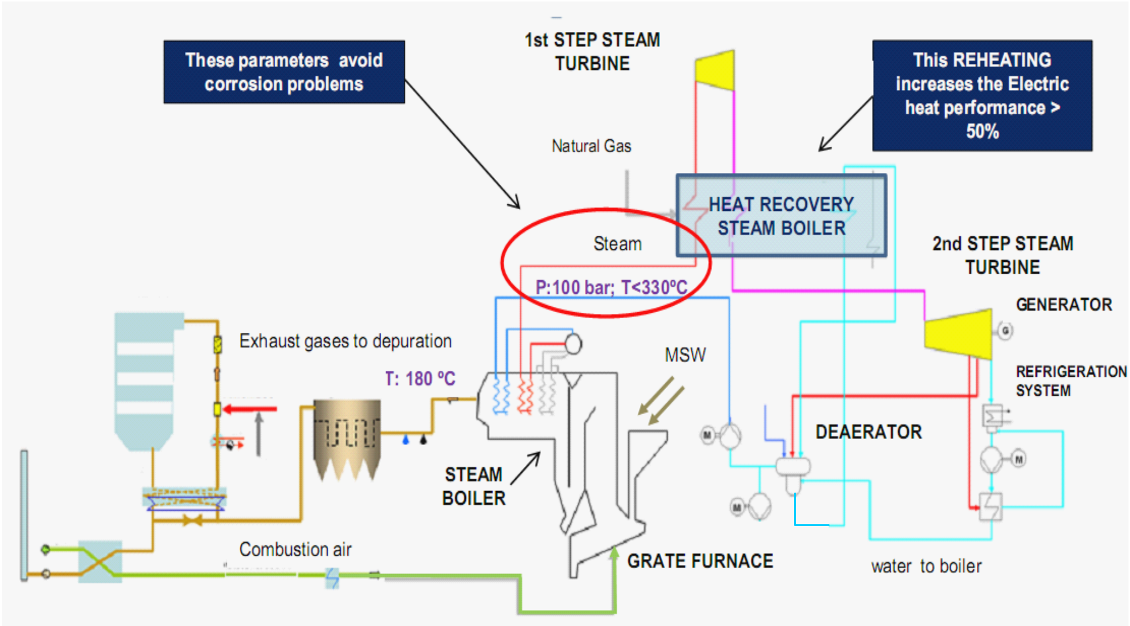

Figure 2 : Zabalgarbi integrated WTE-GT layout [17].

The gas turbine generator chosen is a General Electric LM-6000 of 43 MW average gross capacity, fuelled with natural gas and equipped with intake air cooling system (compression chiller). The GT exhaust temperature of LM6000 is equal to 455 °C, thus, to generate superheated at 540 °C, auxiliary burners (using natural gas and fresh air) are necessary, working at temperature around 650-700 °C.

LM6000 has been chosen considering that, at the time of the project and authorizations, the plant was considered in a special Spanish regime (renewables and cogeneration), where maximum size of the integrated plant should not exceed 100 MW, thus, considering WTE and consequently ST average capacity, the installed GT could not exceed 45 MW.

The effect of eliminating the superheated exchangers from the WTE section has been clearly stated: in conventional plants, maintenance (exchange of tubes) must be done every 1-2 years, in Zabalgarbi, the first rows of tubes have been exchanged in the 6th year of operation.

Due to systems integration the power plant has a high flexibility. The following operations mode are feasible:

- Normal operation;
- Combined cycle operation;
- Fresh air mode
- Incineration and gas turbine through bypass stack;
- Incineration only;

- Shutdown.

Based on Zabalgarbi's operational and economic success, the Vizcaya Regional Government has focused its 2016 Integral Waste Management Plan on doubling the first plant's capacity. The project for Zabalgarbi's second production line will be launched in 2009.

Table 1 : Zabalgarbi technical data [16].

WTE section	
Type of waste	Municipal solid waste and assimilated
Number of lines	1
Consumption of wastes	33,08 tons/h
Waste average LHV	8000 kJ/kg
Productive capacity	240000 t/year
Thermal power from wastes	73.52 MW
STEAM	
Saturated steam mass flow rate	100 t/h
Steam evaporative pressure (HP and LP)	100 and 25 bar
Steam superheated and reheated temperature	540 °C
NATURAL GAS	
Consumption of natural gas (GT+HRB)	13.870 Nm^3/h
LHV of natural gas	38.992 kJ/Nm^3
Thermal power introduced with natural gas	150 MW
ENERGY	
Gross electric power	99,176 MW
Internal consumption	4.745 MWh (5.200 MWh with externals)
Net power (grid connection) Steam Turbine read	54.320 MW
Net power to the grid	94.431 MW
Net power (grid connection) Gas turbine read	40.111 MW
Net electricity generation	730000 MWh/year
EFFICIENCY AND SAVINGS	
Total Electrical Gross Efficiency:	44, 31 %
Overall net plant Efficiency:	42%
Savings on conventional primary energy (thermal plant combined cycle with natural gas comparison).	47%
CO2 emission avoided	300000 t/year

4.5.2. Moerdijk WTE-GT power plant: the Netherlands solution

The history of the WTE-GT power plant of Moerdijk, located in south Netherlands, is due to the nearest of a MSW incinerators (properties of AZN company) and a combined cycle (properties of EPZ, the electric company owning the south of Netherlands). The two systems have been commissioned in the same period (1996 for the incinerator and 1997 for the combined cycle power plant). Thus, the proximity and their contemporary construction, are the main reason for the idea of the integrated plant. For this reason, the combined cycle has all the components and the design of an autonomous system. Figure 3, schematically presenting the layout of the integrated plant; it can be seen a complex three pressure levels combined cycle with reheat. The heat recovery steam generator is used to superheat the steam coming from the furnace boiler to avoid corrosion problems into the WTE section. Steam turbines are serving both the CC and WTE sections. The steam coming from WTE boiler (at 100 bar) is mixed with the steam generated into the HRSG section. The superheated steam at 520 °C is then fed to the high pressure steam turbine. Middle pressure and low pressure steam are generated only into the HRSG section.

Incinerator has a capacity equal to about 80 t/h of waste with a LHV equal to about 10˙450 kJ/kg. Thus, the power introduced with waste is equal to 232 MWth. The combined cycle is equipped with three gas turbines, each with an electric power equal to 60 MW. The total power of the integrated system is equal to about 330 MW with an increase, with reference to separate production, of 18 MW.

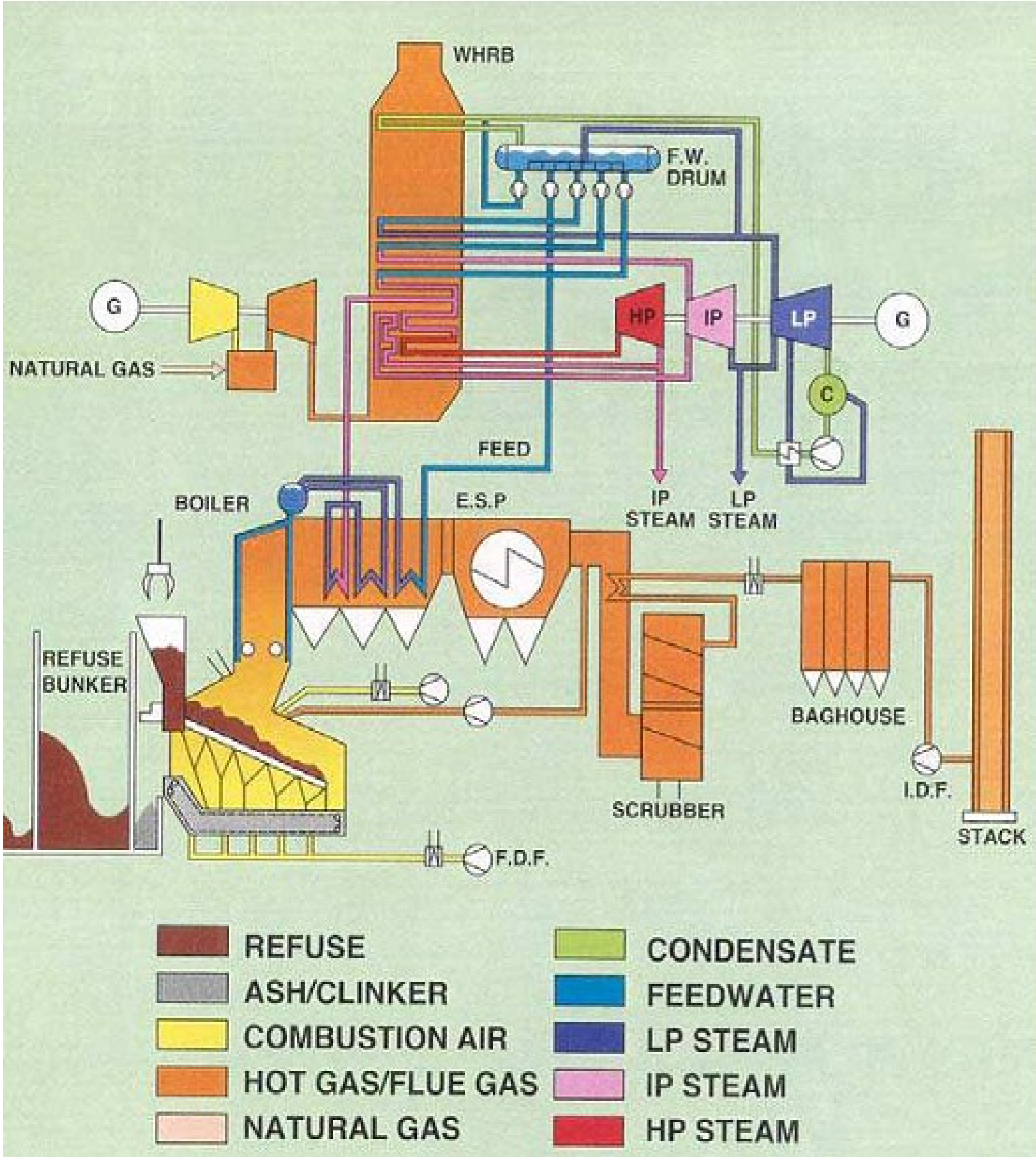

Figure 3 : Moerdijk integrated WTE-CC layout [18].

4.5.3. Takahama WTE-CC power plant: the Japanese solution

Near the city of Takahama in Japan, in 1988 a municipal solid waste incineration plant was built, able to treat 18 tons/h of waste with an average calorific value of about 8˙400 kJ/kg. The primary objective of the incinerator is the disposal of waste, while energy recovery was limited to the generation of only 1.3 MW, essentially equal to the auxiliary and internal consumptions of the plant.

Eight years later came out the idea of increasing the electricity production of the plant, integrating the WTE section with a small gas turbine and an heat recovery boiler where saturated steam generated in the incinerator at a temperature of 255 °C and a pressure of 20 bars could be superheated. In 1996 the integrated plant was realized: a 15 MW gas turbine with a heat recovery boiler, a 10 MW steam turbine and the incinerator. The layout of the Takahama integrated power plant is shown in Figure 4. It can be seen from figure that water, out of the economizer section, is split in two streams thus saturated steam (at 20 bar) is generated both into the WTE and HRSG section; after the mixing, the steam is superheated into the heat recovery boiler at a temperature of 400 °C. The integrated plant has a total capacity of 25 MW.

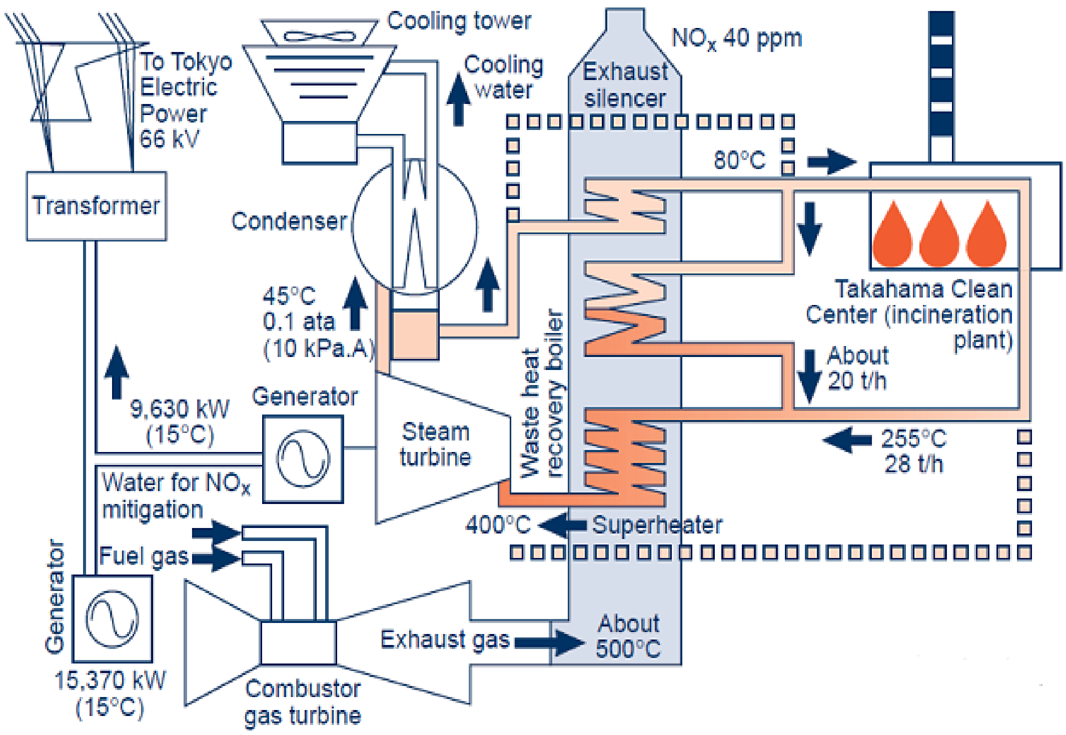

Figure 4 : Takahama integrated WTE-CC layout [19].

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5. WTE-GT steam/water side integration: thermodynamic analysis on one pressure level

This chapter focuses on WTE-GT integrated configurations concerning one pressure level HRSG. The thermodynamic and parametric analysis on steam/water side integrated WTE-GT power plant has been carried out, first of all, with the aim to investigate the logic governing plants match in terms of steam production as function of the thermal powers introduced. Steam generation, optimum plants match condition, heat exchangers inlet and outlet conditions, as a consequence of systems integration, are analyzed and explained. A sensitivity analysis, varying evaporative pressure and HRSG inlet conditions, is also presented in order to investigate the influence of operative parameters on steam mass flow rate.

Starting from a simple integrated plant layout, several configurations are proposed and analyzed. Positive aspects and limitations of each one of the presented layouts were investigated and discussed.

The following paragraphs assess and define, for a given layout and operative conditions, the optimum WTE-GT plant match in terms of system input thermal powers, to maximize steam generation, plant performance and to minimize discharged outlet temperature.

The thermodynamic and parametric analysis was carried out by the use of a commercial software *Gate Cycle*^{TM}*[1]* from General Electric for plant design and simulation.

The software predicts and analyzes design and off-design performance of both the gas and steam sides of power plant. Its component-by-component approach lets modelling any type of system; solving mass and energy balance using a lumped model approach, allows to evaluate inlet and outlet conditions of each system's components and to predict plant performance.

5.1. *Thermodynamic analysis on steam production*

Focusing on a WTE-GT water side integrated power plant, the simplest layout for a one pressure level HRSG is shown in Figure 1; this configuration can be regarded as the starting point, for an integrated plant, to investigate the logic concerning steam production.

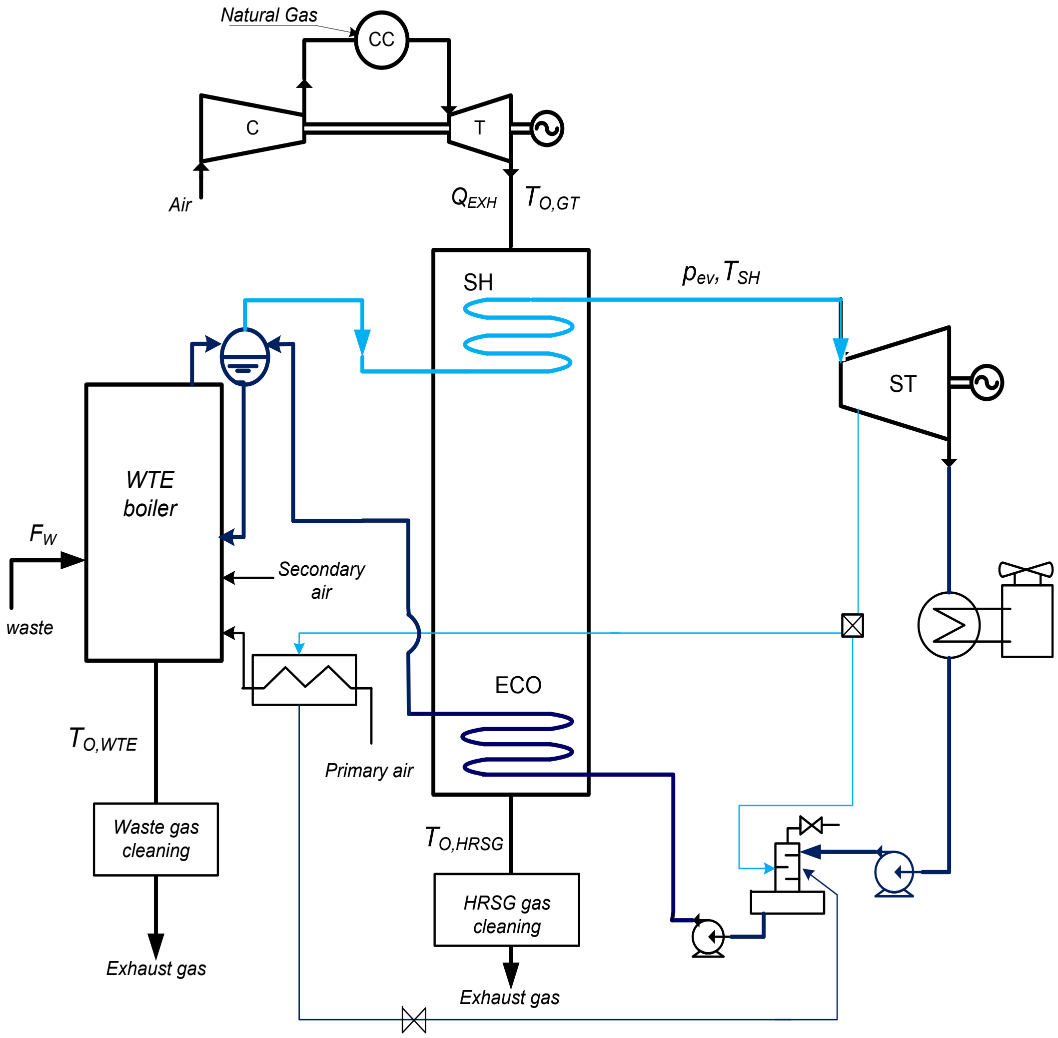

Figure 1 : WTE-TG integrated power plant layout.

Waste boiler has the task of producing saturated steam while water preheating and steam superheating are performed respectively into the HRSG economizer (ECO) and superheater (SH); one bleed from the steam turbine (ST) is present to preheat the primary air necessary for waste combustion and to feed the deaerator.

Focusing on steam production, steam generation is affected by:

- the thermal power discharged from GT entering the HRSG section:

$$Q_{EXH} = \dot{m}_{exh} c_{p,exh} \left(T_{O,GT} - T_{ref}\right) \qquad (1)$$

where $T_{O,GT}$ is the GT outlet temperature, T_{ref} is the reference temperature assumed equal to 15 °C and $c_{p,exh}$ is the exhaust gas specific heat, function of $T_{O,GT}$.

- the input power introduced with waste in the WTE boiler:

$$F_W = \dot{m}_W LHV_W \qquad (2)$$

where $\dot{m}_W$ and LHV_W are waste mass flow rate and lower heating value, respectively.

Consequently, energy balances allow to evaluate the superheated steam produced in the HRSG ($\dot{m}_{s,HRSG}$) and the saturated steam generated by the WTE boiler ($\dot{m}_{s,WTE}$), with the following equations:

$$\dot{m}_{s,HRSG} = \frac{\varepsilon\, Q_{EXH}}{\Delta h_{SH} + \Delta h_{ECO}} \qquad (3)$$

$$\dot{m}_{s,WTE} = \frac{\eta_{boil}\, F_W}{\Delta h_{lat} + \Delta h_{sc}} \qquad (4)$$

where:

- ε is the HRSG effectiveness, mainly depending on the HRSG inlet and outlet temperatures, $T_{O,GT}$ and $T_{O,HRSG}$;
- η_{boil} is the WTE boiler efficiency, primarily due to WTE outlet temperature, $T_{O,WTE}$, and to the ratio between air and fuel waste mass flow rates;
- Δh_{SH} and Δh_{ECO} are steam and water specific enthalpy increases in the superheater and in the economizer, respectively;
- $\Delta h_{lat} + \Delta h_{sc}$ is the specific enthalpy rise in the evaporator, sum of latent heat and sub-cooling enthalpy difference. In particular, once selected the evaporation pressure value (p_{ev}), Δh_{lat} is constant while Δh_{sc} is equal to the difference between evaporative and ECO water outlet temperature. Being related with ECO outlet temperature, the value of Δh_{sc} depends on Q_{EXH}; thus, it can be equal or higher than a minimum value imposed to avoid water evaporation in the ECO section depending on Q_{EXH}.

The adopted layout imposes to have, in the HRSG and in the WTE section, the same value of steam mass flow rate. Consequently, by equating Eq.s (1) and (2), the following must be satisfied:

$$\frac{Q_{EXH}}{F_W} = \frac{\eta_{WTE}}{\varepsilon} \frac{\Delta h_{SH} + \Delta h_{ECO}}{\Delta h_{lat} + \Delta h_{sc}} \qquad (5)$$

Figure 2 shows qualitatively how Eq. (5) could be solved in order to calculate the steam production, considering different GT discharge thermal power values and assuming constant values of $T_{O,GT}$, p_{ev} and F_W.

In particular, $\dot{m}_{s,max}$, calculated from Eq. (4), represents the upper limit of steam production, regardless of the HRSG size, determined only by the thermal power introduced with waste, F_W.

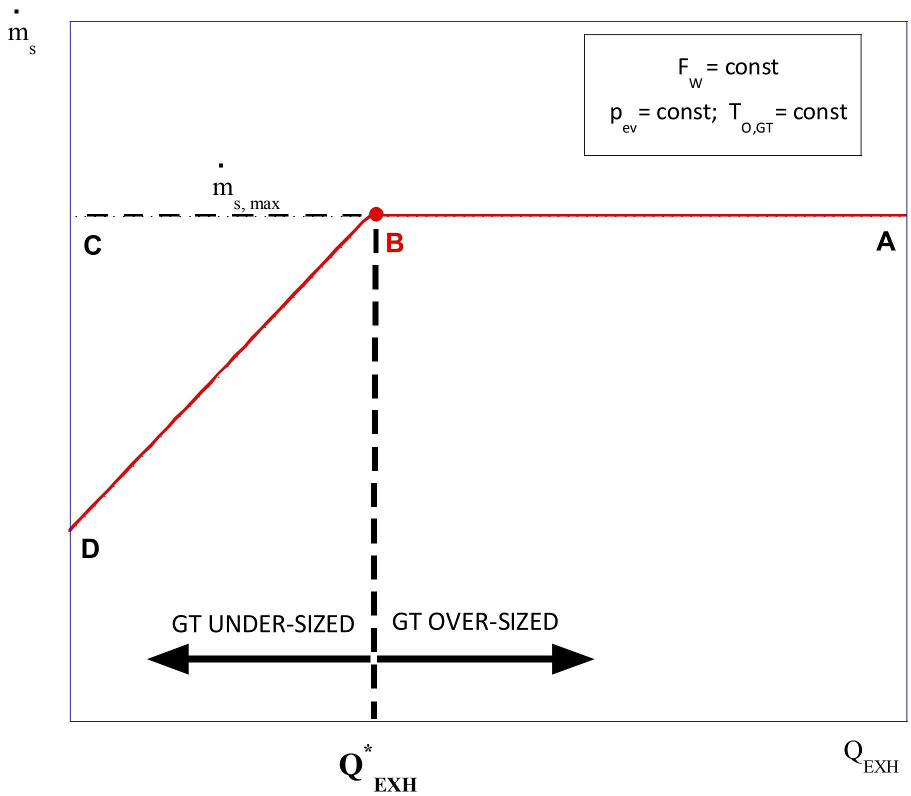

Figure 2 : steam mass flow rate as function of GT discharged thermal power.

For a discharged GT thermal power higher than Q^*_{EXH} (point B), moving from B to A, it occurs:

– HRSG outlet temperature ($T_{O,HRSG}$) increases (decreasing HRSG effectiveness, ε), overcoming its minimum value (Figure 3 a);

- no changes occur to sub-cooling temperature difference, ΔT_{sc} (Figure 3 b) and to the superheated steam temperature, T_{SH} (Figure 3 c); both keep respectively their minimum and maximum values.

A reduction of Q_{EXH} under Q^{*}_{EXH} can be managed with two possible strategies:

i) dotted line B $\rightarrow$ C: to keep steam mass flow rate at its maximum value $\dot{m}_{s,max}$, a reduction of T_{SH} is required. In this case the thermal power discharged from GT is not enough to superheat at the maximum temperature all the steam that the WTE boiler could produce.

ii) continuous line B $\rightarrow$ D: T_{SH} can be kept constant to not penalize the steam cycle efficiency, both reducing the steam mass flow rate and increasing ΔT_{sc}.

In this case, the difference between the evaporative and ECO outlet temperatures is higher than the minimum value allowed; this sub-cooling temperature increase is quite equivalent to the introduction of an economizer section in the WTE[1] boiler.

In conclusion, point B represents the best choice (maximum T_{SH}, minimum ΔT_{sc} and $T_{O,HRSG}$) for the assumed layout.

Anyway, it is possible to operate keeping the thermodynamic parameters at their optimum values (maximum T_{SH} and minimum $T_{O,HRSG}$, line B-D) but, in this case, steam mass flow rate is lower than its maximum value. Thus, WTE section turns out to be undersized.

[1] Actually, if an economizer section was placed into the WTE boiler, point D will show an increase in steam mass flow rate: thanks to a lower water temperature entering the WTE boiler, a further reduction of $T_{O,WTE}$ occurs.

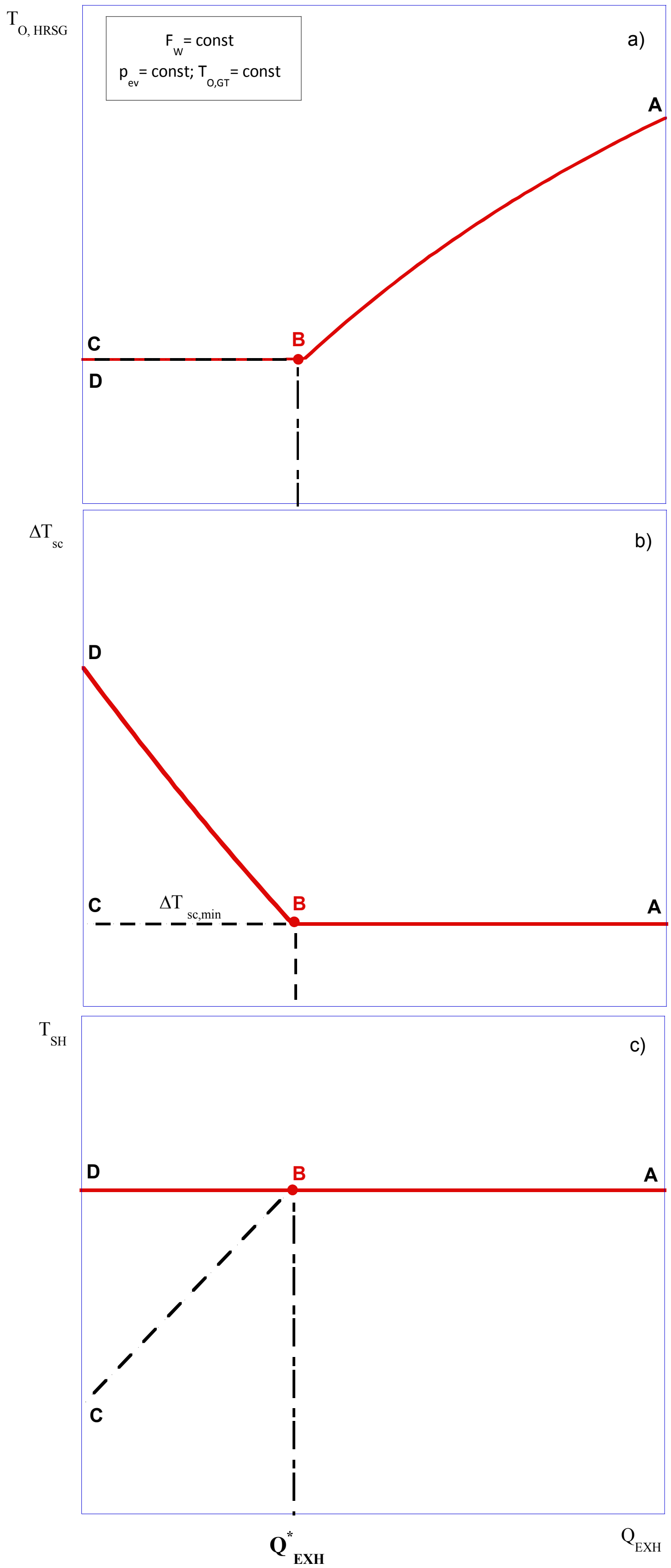

Figure 3 : $T_{O,HRSG}$ (a), ΔT_{sc} (b) and T_{SH} (c) realized as function of GT discharged thermal power.

On the contrary, there is no convenience to operate keeping the maximum mass flow rate $\dot{m}_{s,max}$, but with Q_{EXH} far from Q^*_{EXH} (line A-B or line B-C): this means to operate with low HRSG effectiveness (A-B) or with low thermodynamic steam cycle efficiency (B-C).

The performed analysis highlights that, for a given evaporative pressure and GT outlet temperature, an optimum plant match, in terms of thermal powers ratio, Q^*_{EXH}/F_W , is found.

Beyond this value, an increase in HRSG input thermal power, does not lead to additional benefits in terms of generated steam mass flow or HRSG effectiveness.

Figure 4 shows the T-Q diagram for the HRSG section, in correspondence to the optimum value (point B, Q^*_{EXH}/F_W).

As previously described, one of the advantage of a WTE-GT integrated plant is the reduction of the HRSG irreversibility caused by high mean temperature differences. As evidenced form figure, removing the evaporator from the HRSG reduces the distance between the exhaust gas and steam/water lines, with respect to a typical one pressure level HRSG T-Q diagram.

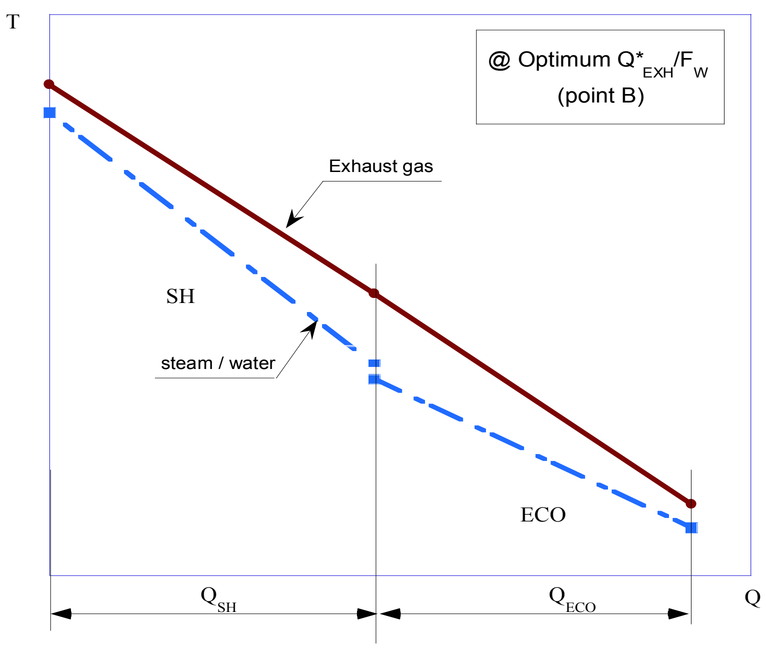

Figure 4 : T-Q diagram for HRSG section of the integrated plant.

Finally, for the investigated layout, it must be noticed that $T_{O,WTE}$ strongly depends on evaporation temperature; thus the evaporation pressure choice mainly affects η_{boil}: the higher is p_{ev} the lower is the WTE boiler efficiency.

Influence of evaporative pressure and GT outlet temperature on steam production

In order to investigate the influence of steam cycle parameters on optimum WTE-GT plant match, a parametric analysis has been performed varying steam evaporative pressure and GT outlet temperature.

In particular, Figure 5 and Figure 6 present, respectively, the influence of three different evaporative pressure ($p_{ev1} < p_{ev2} < p_{ev3}$) and $T_{O,GT}$ values ($T_{O,GT_1} < T_{O,GT_2} < T_{O,GT_3}$).

Focusing on Figure 5, an increase in evaporative pressure value, decreasing Δh_{lat}, can lead to an increase in steam mass flow rate as expressed in Eq. (4).

If the thermal power introduced with GT exhaust is not enough to reach steam maximum flow rate (points B, B' and B''), the amount of produced steam is lower than the maximum value and its variation is quite not dependent from p_{ev} (lines are overlapped).

Increasing Q_{EXH} over Q^*_{EXH} a constant trend is reached for each investigated evaporative pressure value. Moreover, increasing the evaporation pressure, the optimum condition (B, B', B'') is characterized by a greater optimum thermal powers ratio Q^*_{EXH}/F_W value.

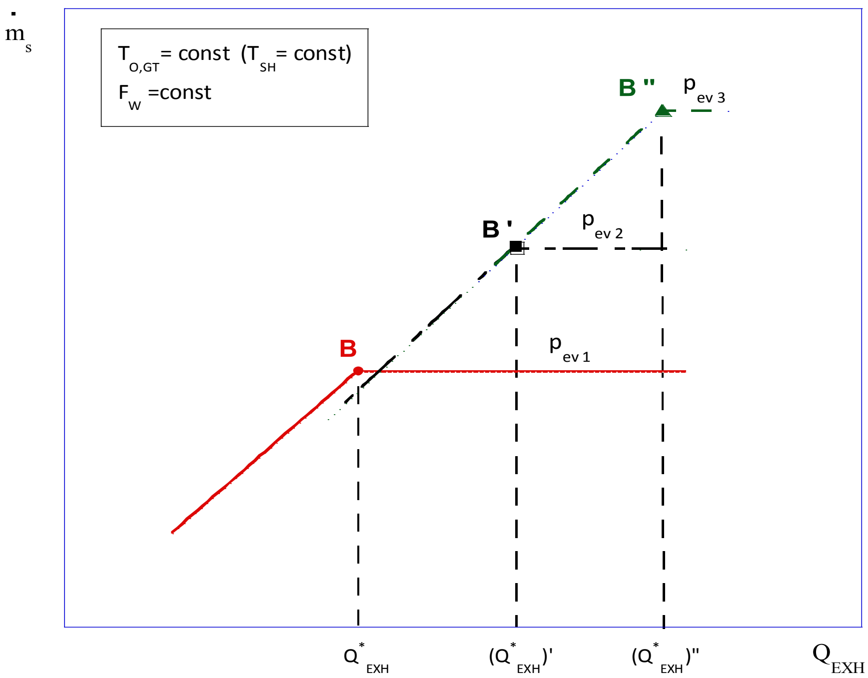

Figure 5 : steam mass flow rate as function of GT discharged thermal power for different evaporative pressure value.

The influence of $T_{O,GT}$ (and consequently, T_{SH} having assumed a constant approach difference) is presented in Figure 6: for Q_{EXH} below the optimum point, an increase in GT outlet temperature (and on steam superheated one) decreases the steam mass flow rate. Once reached the optimum point (B, B_2, B_3), steam production is function only of the input waste thermal power, and it is not affected by $T_{O,GT}$.

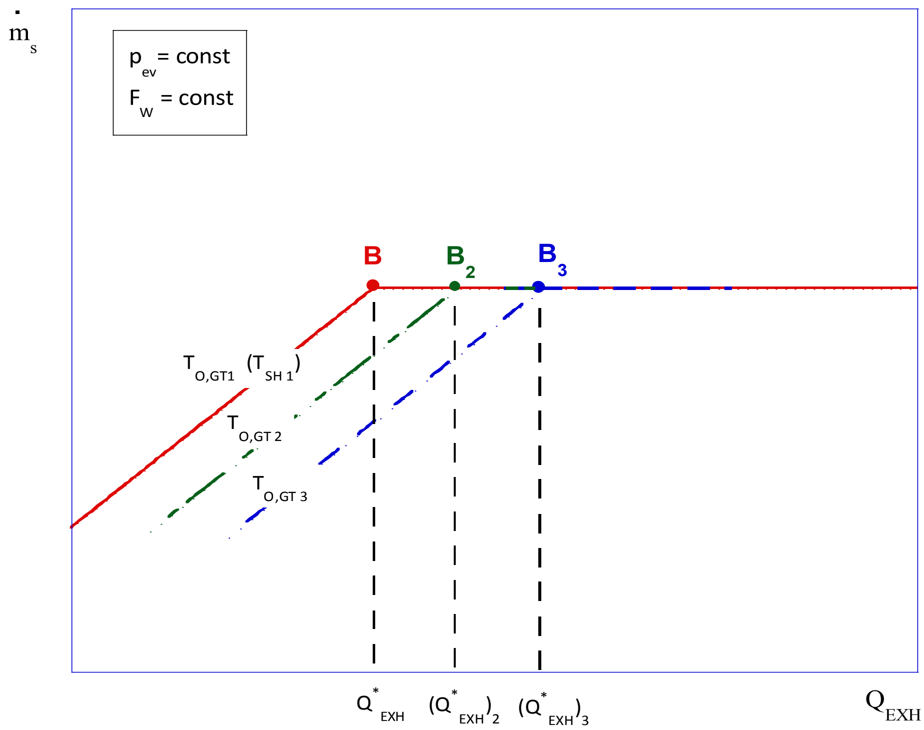

Figure 6 : steam mass flow rate as function of GT discharged thermal power for different GT outlet temperature.

Numerical results

The main results of the study are collected together and shown in Figure 7. A constant waste composition and LHV_W value have been assumed, as reported in Table 1. In Table 2 additional assumptions used for numerical analysis are listed.

The grid of Figure 7 represents optimum plant match conditions (points B) for different evaporation pressure and GT outlet temperature values.

By intersecting line at constant pressure with line at constant temperature, the corresponding steam mass flow rate and the optimum ratio between thermal input powers are obtained.

As an example, selecting an evaporative pressure equal to 60 bar and a GT outlet temperature of 500 °C (corresponding to 480 °C of superheated steam) about 6 kg/s of steam mass flow rate is generated for every kg/s of waste. Moreover, the optimum plant match is found in correspondence to $Q^*_{EXH}/F_W = 0.93$.

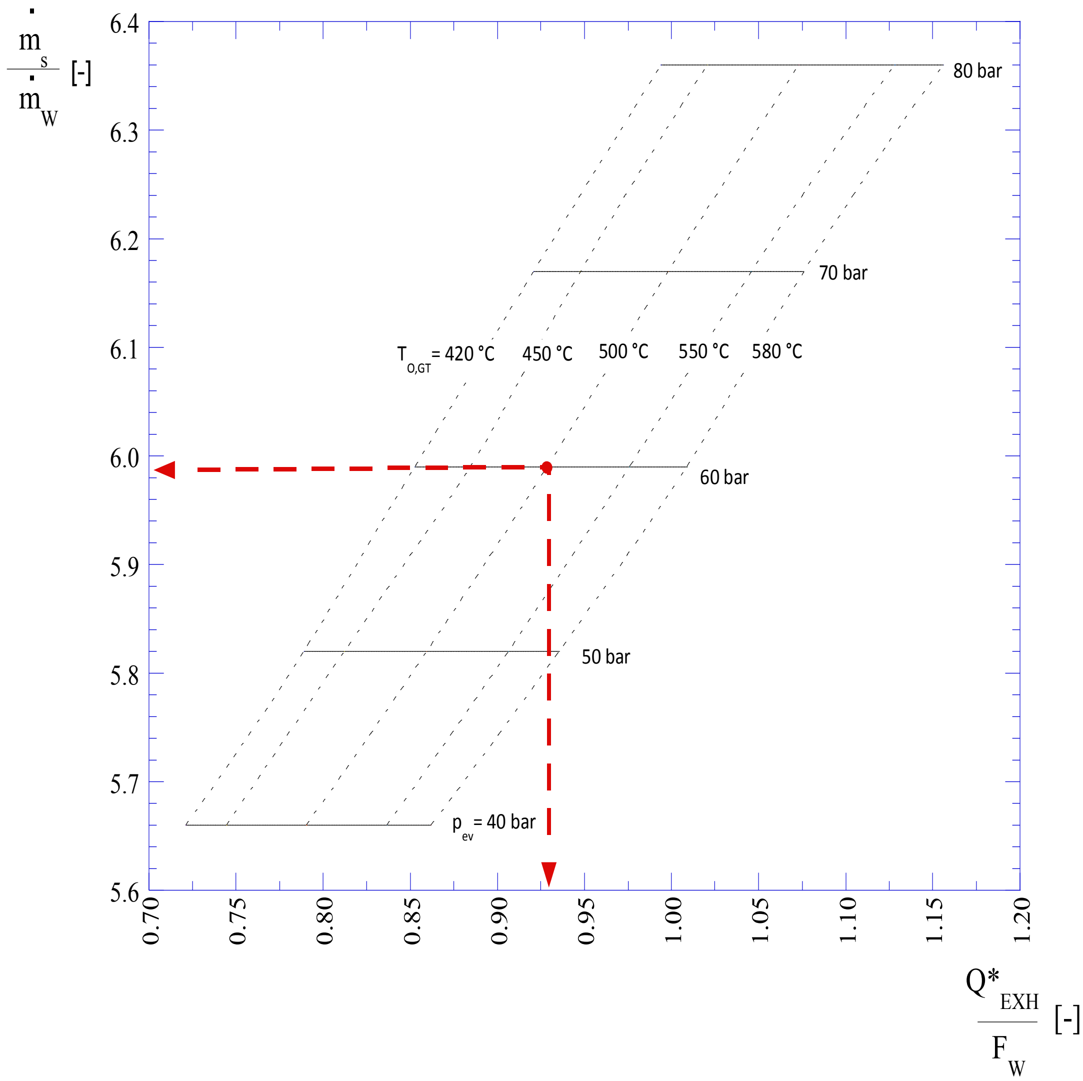

Figure 7 : grid representing optimum plant match for a given evaporative pressure and GT outlet temperature as function of thermal input powers ratio.

Table 1 : MSW composition and LHV assumed for the analysis.

Proximate analysis (% by weight)	Humidity	23.0	% of volatile matter	C	52.5
	Ashes	15.0		H	7.5
	Volatile matter	62.0		O	38.5
				N	1.3
				S	0.2
LHV_W =11.85 MJ/kg_W (2.83 $kcal/kg_W$)					

Table 2 : Main assumptions

ST extraction pressure for primary air heating and deaerator [bar]	3.5
ST isentropic efficiency [-]	0.85
ST outlet quality [-]	0.84 ÷ 0.95
Condenser pressure [bar]	0.1
Dearetor working pressure [bar]	3
Oxygen in exhaust dry gases [%]	7
WTE exhaust gas recirculation [%]	15
WTE exhaust gas recirculation temperature [°C]	150
Primary air temperature [°C]	130
Secondary air temperature [°C]	50
ΔT pinch point [°C]	10
ΔT approach [°C]	≥ 20
ΔT_{sc}[°C]	≥ 10
$T_{O,WTE}$ [°C]	≥ 160
$T_{O,HRSG}$ [°C]	≥ 110

As highlighted in Figure 7, high evaporative pressure and steam superheated temperature means high Q^*_{EXH}/F_W . Thus, the higher are the steam cycle parameters the higher must be the GT discharged thermal power with respect to thermal power introduced with waste.

Optimum plant match in terms of electric powers ratio

In order to evaluate the GT electric size, P_{GT}, that must be chosen to realize the integrated power plant of Figure 3, the following equation can be written:

$$\frac{P_{GT}}{P_{WTE}} = \frac{P_{GT}}{Q_{EXH}} \cdot \frac{Q_{EXH}}{F_W} \cdot \frac{F_W}{P_{WTE}} \qquad (6)$$

where P_{WTE} is the electric capacity of a traditional WTE power plant fed with the same waste input of the integrated system. As highlighted from Eq. (6) P_{GT}/P_{WTE} depends on: GT and WTE electric efficiency and system input thermal power ratio.

Taking into account that:

- the electric capacity of a GT is a fraction of the discharged thermal power Q_{EXH} ranging from 0.3 to 0.8, depending on the GT characteristics (see Figure 8, showing performance of several GT commercial units);
- for a traditional WTE power plant, the efficiency P_{WTE}/F_W, typically ranges from 0.25 to 0.30.

Eq. (6) can be re-written as:

$$\frac{P_{GT}}{P_{WTE}} = \left\{ \left(0.3\frac{1}{0.3}\right)\frac{Q^{*}_{EXH}}{F_W} ; \left(0.8\frac{1}{0.25}\right)\frac{Q^{*}_{EXH}}{F_W} \right\} \approx \left\{ 1\frac{Q^{*}_{EXH}}{F_W} ; 3\frac{Q^{*}_{EXH}}{F_W} \right\} \qquad (7)$$

highlighting that the optimum matching requires to select a GT with an electric power output up to three times the WTE original plant electric power size.

To minimize the GT size the P_{GT}/Q_{EXH} ratio has to be the lowest. Since a relationship between P_{GT}/Q_{EXH} and the GT efficiency (η_{GT}) can be identified as follows:

$$\eta_{GT} \approx \frac{1}{1 + \left(\frac{Q_{EXH}}{P_{GT}}\right)} \qquad (8)$$

this means that the GTs with low efficiency are best choice for this purpose.

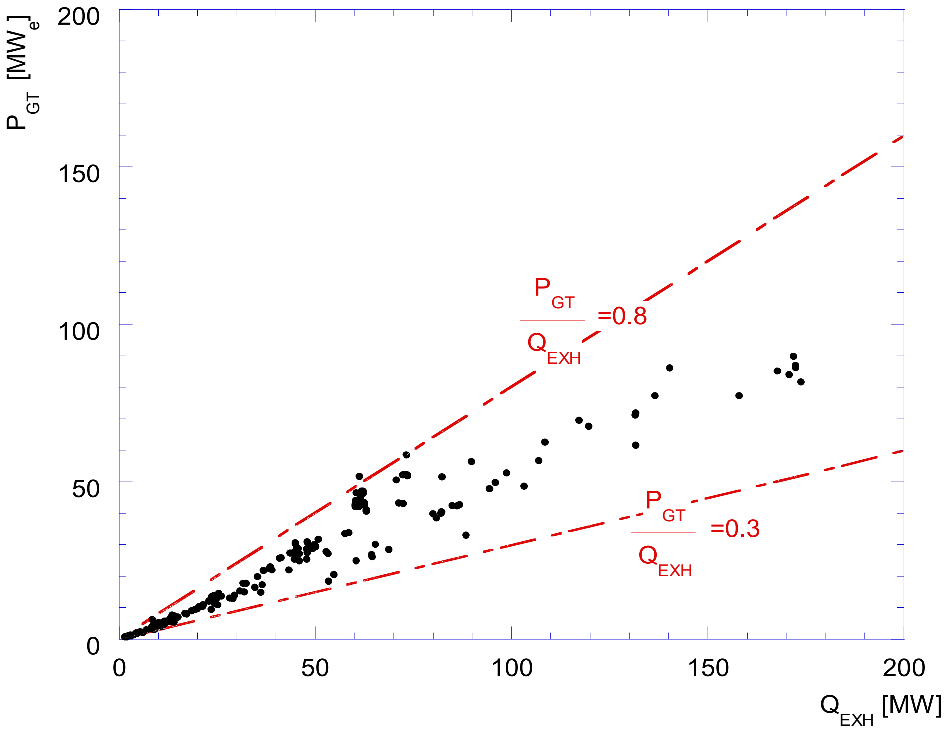

Figure 8 : GT power output as function of GT discharged heat for commercial GT units.

Figure 9 shows, as example, the steam to waste mass flow ratio as function of electric powers output $\frac{P_{GT}}{P_{WTE}}$. Figure refers to a WTE efficiency equal to 30 % and a ratio between GT electric power output and discharged heat equal to 0.3. Thus, optimum values expressed in terms of thermal powers ratio coincide with that found in terms of electric powers output ratio $\left(\frac{P_{GT}}{P_{WTE}} = \frac{Q^*_{EXH}}{F_W}\right)$.

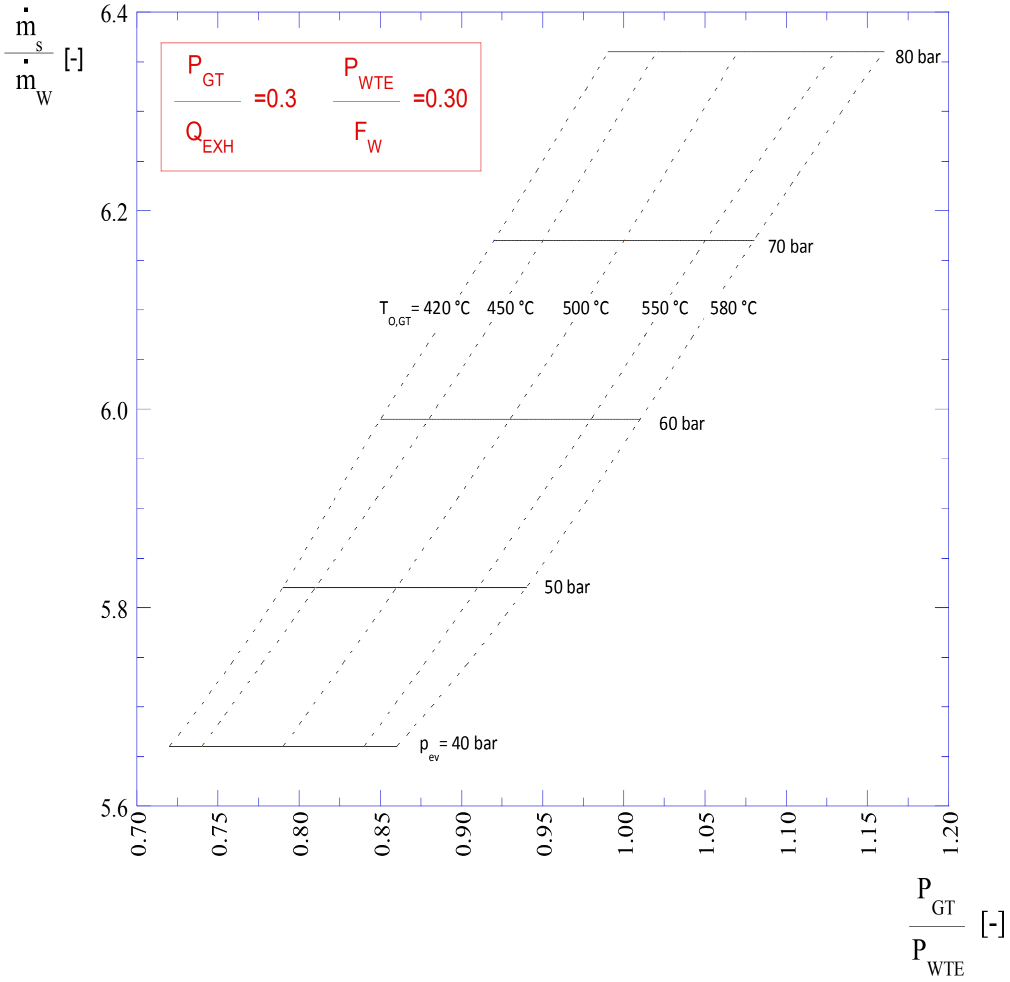

Figure 9 : grid representing optimum plant match as function of output powers ratio for a WTE efficiency equal to 30% a ratio between GT electric power output and discharged heat equal to 0.3.

Instead, Figure 10 shows that optimum values of electric powers ratio are about three times more, assuming a WTE efficiency equal to 25% and a ratio between GT electric power output and discharged heat equal to 0.8.

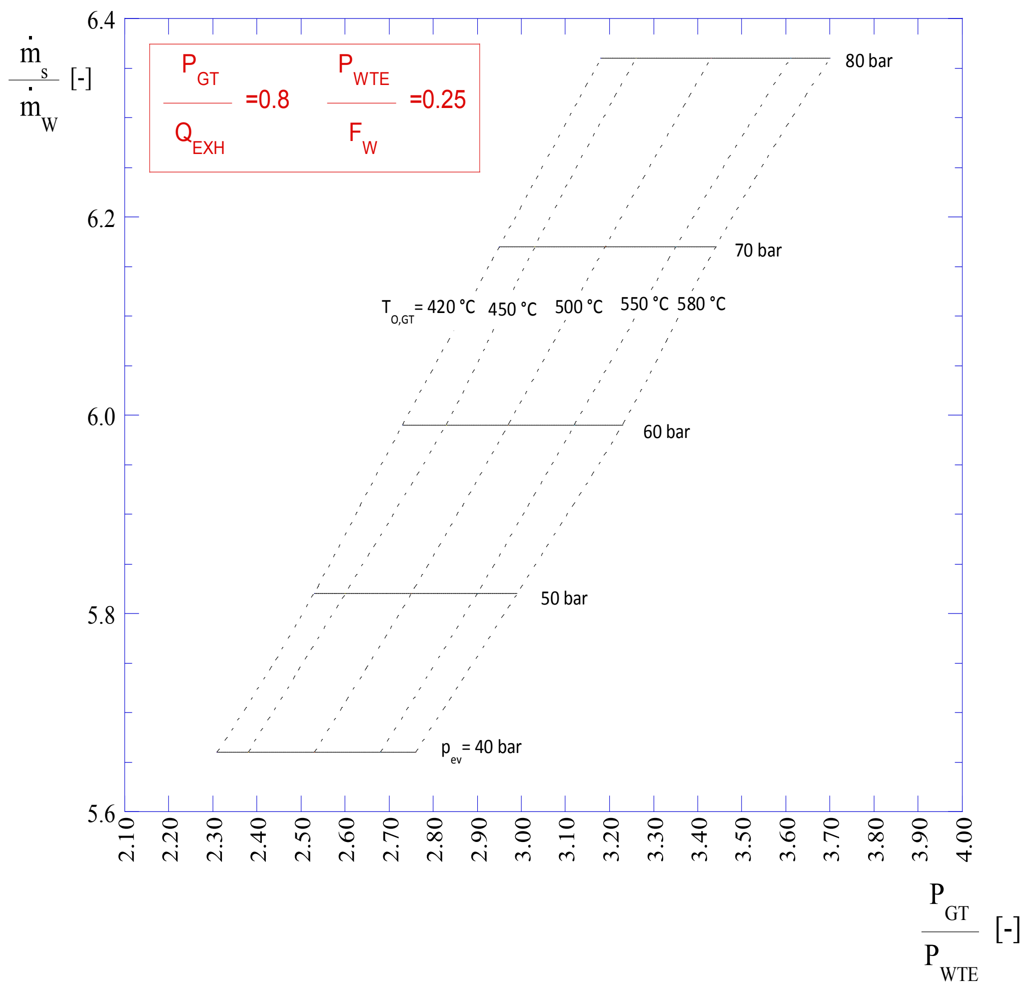

Figure 10 : grid representing optimum plant match as function of output powers ratio for a WTE efficiency equal to 25% and a ratio between GT power output and discharged heat equal to 0.8.

<u>Traditional WTE vs. integrated plant: steam turbine power output</u>

Comparing the steam mass flow rate produced by the integrated plant with that generated by a traditional WTE fed with the same amount of waste input power, a considerable increase is found. A WTE stand alone power plant working at the same evaporation pressure and steam superheated temperature (respectively equal to 50 bar and 480 °C) and receiving, with waste, a thermal input power equal to 100 MW would generate about 32.09 kg/s of saturated steam mass flow rate and would have a ST capacity equal to 29.7 MW.

As a consequence, the capacity of the steam turbine in the WTE-GT integrated system will be higher than the one employed in a stand alone WTE power plant. It can be expressed as:

$$P_{ST} = P_{WTE} + \Delta P_{ST} \qquad (9)$$

where ΔP_{ST} is quite proportional to the steam mass flow increase.

Figure 11 shows percentage increment of steam turbine power output on the respect of a WTE stand alone power plant, highlighting that it ranges, for the considered cases, from 35 % to 70 %, reaching its highest values for the maximum evaporative pressure and steam superheated temperature.

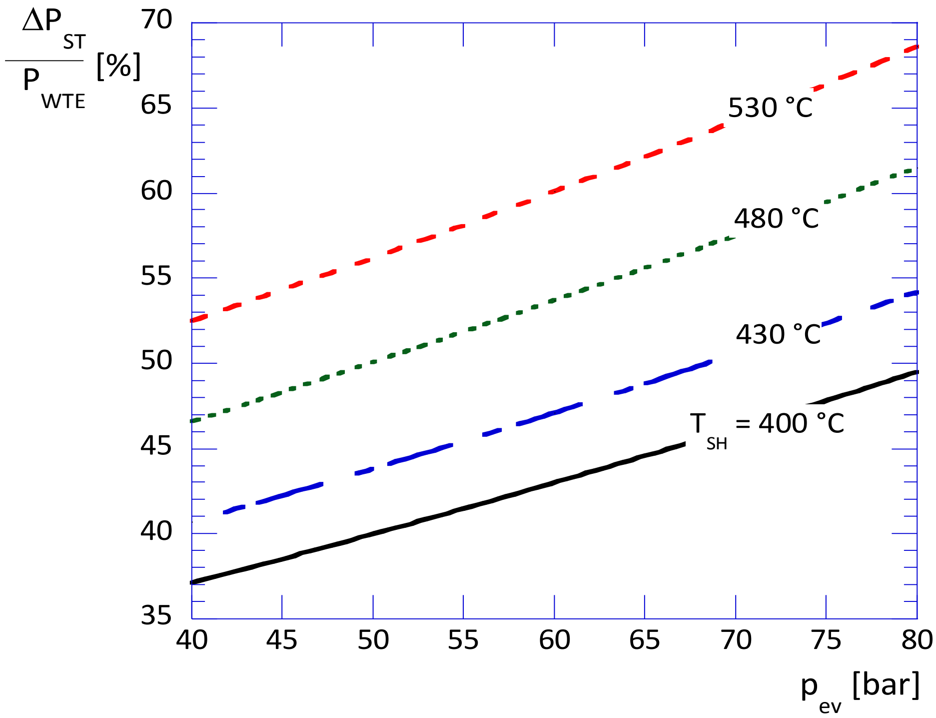

Figure 11 : increase in steam turbine power output as function of evaporative pressure and GT outlet temperature.

5.2. Concluding remarks on thermodynamic analysis

Results of the study suggest that an optimum WTE-GT plant match in terms of system input thermal powers must be pursued to maximize steam generation, steam superheated temperature and to minimize exhaust gases temperature. Moving far from optimum condition it means: i) oversize the gas turbine without additional benefits in terms of generated steam mass flow rate or Heat Recovery Steam Generator effectiveness, ii) depress the WTE section decreasing the amount of generated steam or, iii) working with low steam cycle thermodynamic efficiency. Thus, the carried out thermodynamic and parametric analysis provide useful guidelines in selecting optimum gas turbine size to match WTE-GT maximum performance. Results of the study highlight that the higher are the steam cycle parameters (evaporative pressure and steam superheated temperature) the higher must be the gas turbine discharged thermal power on the respect of thermal power introduced with waste.

A correspondence between optimum thermal and electric powers ratios can be achieved only if the gas turbine efficiency is low; otherwise, optimum values in terms of electric powers ratio can be tree times that found in correspondence to input thermal powers.

Comparing steam mass flow rate produced in the WTE-GT integrated system with a WTE stand alone, significant increases have been found. Moreover, percentage gains in steam mass flow rate increase increasing evaporative pressure and steam superheated temperature. Being increments in steam turbine power output quite proportional with increments in steam mass flow rate, values of steam turbine new capacity for the integrated plant have been derived: steam turbine power output increase ranges from 35 % to 70 %, reaching its highest values for the maximum evaporative pressure and GT outlet temperature.

5.3. WTE-GT proposed layouts for a one pressure level HRSG

In this paragraph WTE-GT layouts for a one pressure level HRSG are proposed and investigated. Integrated system layouts will be optimize, case after case, in order to exploit both: the thermal power input with waste and the GT discharged thermal power. Aim of the analysis is to improve HRSG and WTE boiler configurations in order to maximize steam generation minimizing the optimum plant match condition in terms of Q_{EXH}. For each one of the proposed configurations, steam mass flow generation is analyzed and discussed as function of GT discharged thermal power and the optimum plant match condition is identified. Based on common assumptions, briefly described in the following section, numerical results for each one of WTE-GT integrated plant layouts are highlighted.

5.3.1. <u>Assumption of the thermodynamic analysis</u>

Municipal solid waste composition and, as a consequence, waste lower heating value (LHV) assumed for the analysis has been taken according to Table 3. The assumed waste composition and LHV is in line with typical values of residual waste after recycling operation as suggested in Chapter 2, paragraph 2.5.

Table 3 : MSW composition and LHV assumed for the analysis.

<table>
<tr><td rowspan="5">Proximate analysis (% by weight)</td><td>Humidity</td><td>23.0</td><td rowspan="5">% of volatile matter</td><td>C</td><td>52.5</td></tr>
<tr><td>Ashes</td><td>15.0</td><td>H</td><td>7.5</td></tr>
<tr><td rowspan="3">Volatile matter</td><td rowspan="3">62.0</td><td>O</td><td>38.5</td></tr>
<tr><td>N</td><td>1.3</td></tr>
<tr><td>S</td><td>0.2</td></tr>
<tr><td colspan="6">LHV_W =11.85 MJ/kg$_W$ (2.83 kcal/kg$_W$)</td></tr>
</table>

Being interest in evaluate the optimum plant match condition in terms of thermal power discharged by GT exhaust, a typical GT exhaust gases composition has been assumed according to [3]. The reference composition entering the HRSG section, shown in Table 4, refers to an air to fuel (CH_4) mass flow rate ratio equal to 68.94. Taking into account stechiometric ratio between combustion air and methane (17.23), the excess of air for the exhaust gases composition assumed turns out to be equal to 4, in line with typical GT combustion values.

Table 4 : GT exhaust gases composition assumed [3].

Molecular Weight	28.6344
Molecular fraction composition	
Oxygen, O2	0.1531
Nitrogen, N2	0.7610
Water, H2O	0.0510
Carbon Monoxide, CO	0.0000
Carbon Dioxide, CO2	0.0258
Methane, CH4	0.0000
Hydrogen, H2	0.0000
Argon, AR	0.0091
Carbonyl Sulfide, COS	0.0000
Hydrogen Sulfide, H2S	0.0000
Sulfur Dioxide, SO2	0.0000
Specific henatlpy value	
h_{exh} (@ 500 °C) [kJ/kg]	519.78
h_{rif} (@ 15 °C) [kJ/kg]	-57.04

For what concern waste combustion, assumptions shown in Table 5 and the following aspects have been considered in the analysis:

- The total air requirement for the waste boiler is setting in order to obtain a prescribed oxygen concentration in dry exhaust gases; based on oxygen content in the exhaust gas, the program automatically calculates the excess of air with the respect to a stechiometric combustion. For the assumed value of oxygen content in dry exhaust gases the combustion air to waste mass flow ratio turns out to be equal to 6.55 (thus, the excess air fraction is equal to 0.63).

- Moreover, in the program, the primary air is specified in relation to the mass flow rate of the fuel, i.e. as a mass fraction (mass flow of primary air/mass flow of fuel) assumed equal to 4.1, thus, 63%of the total amount of combustion air is identified as primary air while, the part is secondary combustion air.
- Primary and secondary combustion air are supposed to be preheated up to a temperature equal to 50°C by exploiting grate cooling.
- The integrated boiler (representing three vertical radiation passes, see Chapter 2 paragraph 2.3.7) has been modeled taking into account both, the contribution of thermal radiation and convective heat transfer; the overall heat transferred to evaporators, water walls bordering the combustion chamber, is mainly due to thermal radiation (about 87%).
- The assumed MSW composition, the calculated combustion air mass flow, the assumed exhaust flue gases recirculation and combustion parameters lead to an adiabatic combustion temperature equal to about 1185 °C which is in line with typical values for a WTE integrated boiler.
- A convective heat exchanger section (convection pass, see Chapter 2 paragraph 2.3.7) has been also modeled. The temperature at the inlet of the convective section has been fixed, according to typical WTE value, equal to 650 °C (see Chapter 2 paragraph 2.3.7).
- Effectiveness of heat exchanger has been upper limited to limit heat exchanger surfaces.

The remaining steam cycle common assumptions are reported in Table 5

Table 5 : Main assumption

ST extraction pressure for primary air heating [bar]	3.5
ST isentropic efficiency [-]	0.85
ST outlet quality [-]	0.84 ÷ 0.95
Condenser pressure [bar]	0.1
Dearetor working pressure interval [bar]	2÷3.5
Oxygen in exhaust dry gases [%]	7
WTE exhaust gas recirculation [%]	15
WTE exhaust gas recirculation temperature [°C]	150
Primary air temperature [°C]	130

Secondary air temperature [°C]	50
ΔT pinch point [°C]	10
ΔT approach [°C]	20
ΔT_{sc}[°C]	≥ 10
$T_{O,WTE}$ [°C]	≥ 160
$T_{O,HRSG}$ [°C]	≥ 110

LAYOUT_1

As described above, the first WTE-TG integrated layout proposed (Layout_1) is shown in Figure 12. Waste boiler has the task of producing saturated steam while water preheating and steam superheating are performed respectively into the HRSG economizer (ECO) and superheater (SH); one bleed from the steam turbine (ST) is used to preheat the primary air necessary for waste combustion and to feed the deaerator. In this configuration, only evaporator heat exchanger is present in the WTE boiler. For Q_{EXH} equal or above Q^{*}_{EXH} the economizer is completely disposed into HRSG section. Figure 13 shows a T-Q diagram for the HRSG section in correspondence to the optimum point. The 56% of the thermal power introduced in HRSG is exploited, quite equally shared between SH and ECO.

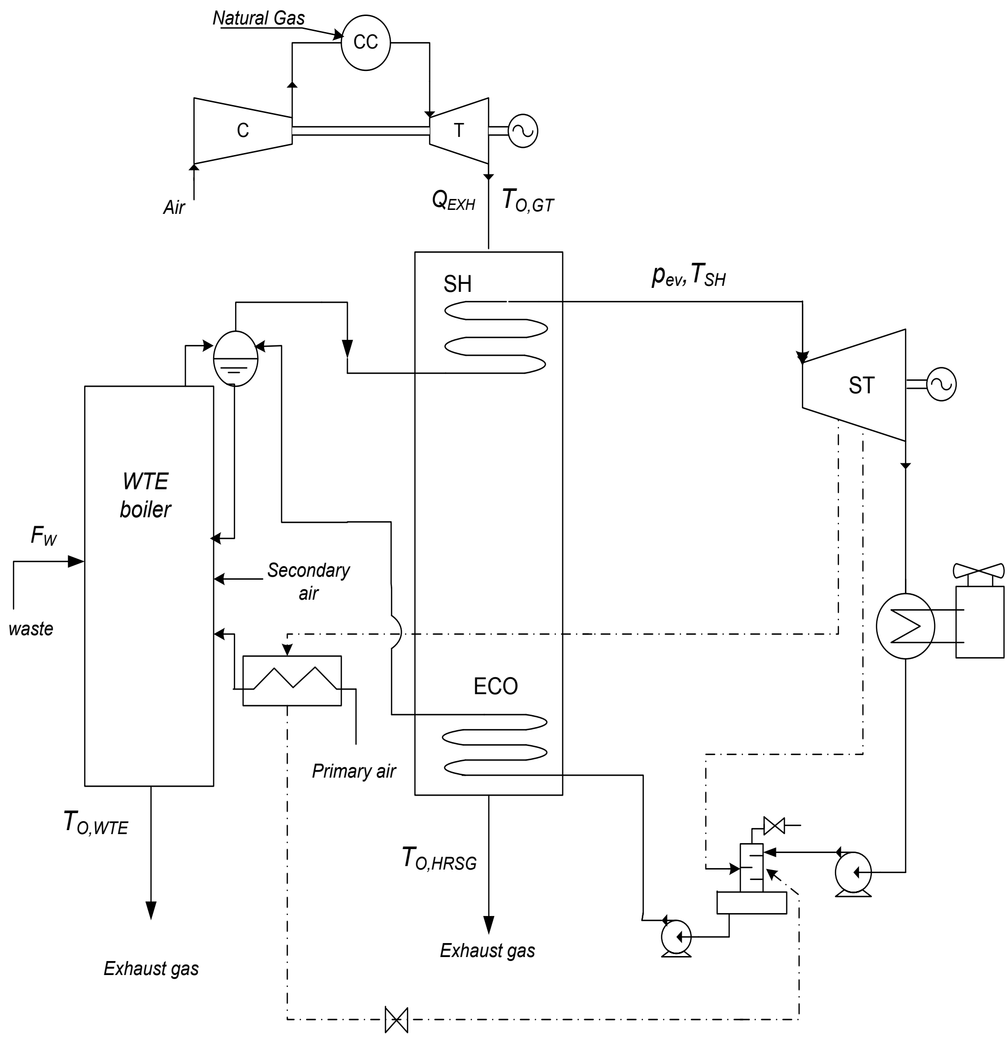

Figure 12 : Layout_1

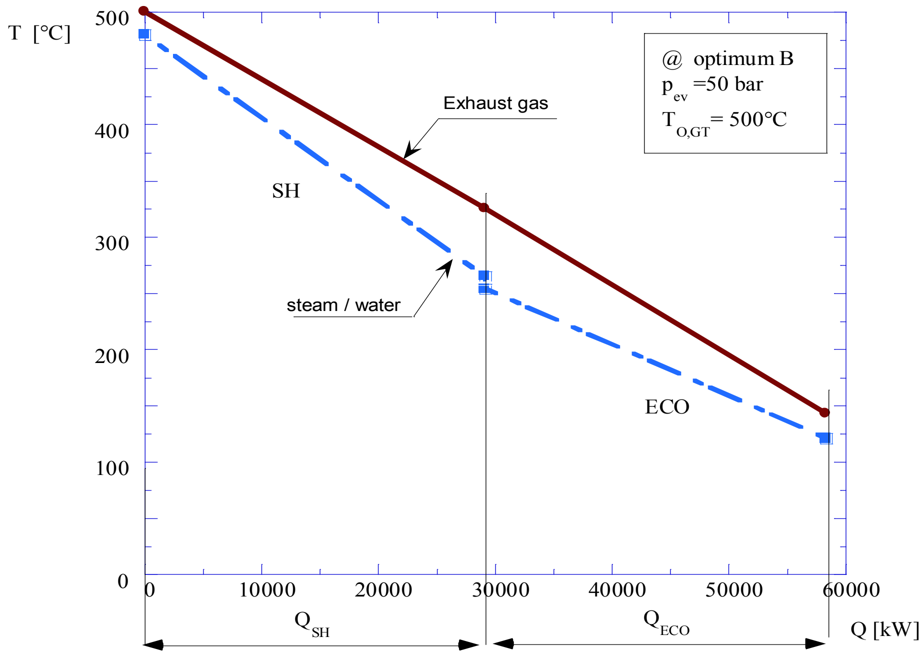

Figure 13 : T-Q diagram for HRSG section for Layout_1.

Figure 14 shows the steam mass flow rate generated as function of GT discharged thermal power for a fixed evaporative pressure and GT outlet temperature (or steam superheated temperature) and assuming a constant thermal input with waste. As previously described, the steam mass flow rate trend increases quite linearly, reaching the maximum value in correspondence to the optimum plant match (Q^*_{EXH}, point B), above which a constant trend is reached. Optimum plant match condition, for the considered layout turns out to be equal to about 87.7 MW corresponding to maximum steam mass flow rate of 49.12 kg/s. The lowest Q_{EXH} considered value (corresponding to a steam mass flow rate equal to about 44 kg/s) agree with the highest effectiveness assumed for SH heat exchanger. On the contrary, there is no upper limitation on Q_{EXH} but the constant trend has been interrupted in proximity to point B because, as previously described, there is no convenience in operating with values of Q_{EXH} higher than Q^*_{EXH}.

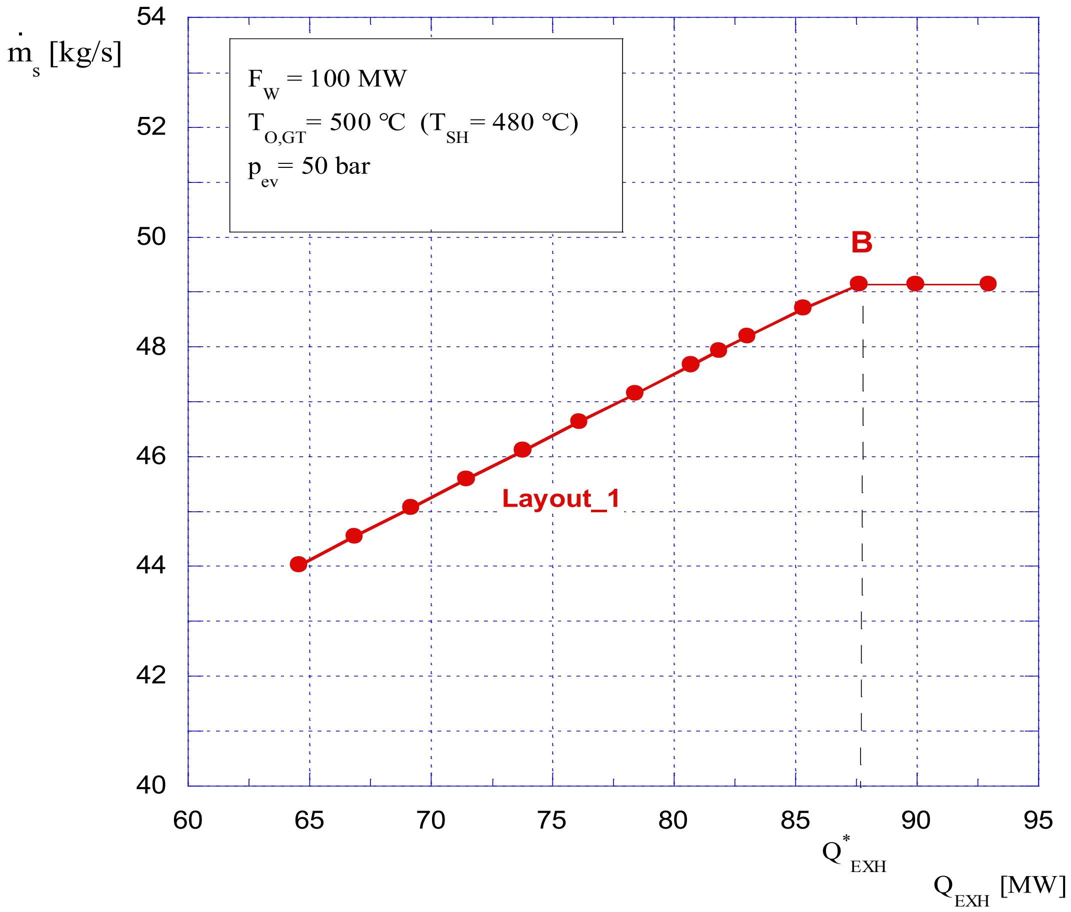

Figure 14 : steam mass flow rate for Layout_1 as function of GT discharged thermal power.

For the investigated layout, it must be noticed that $T_{O,WTE}$ strongly depends on evaporative temperature; having assumed only the presence of evaporative heat exchanger into the WTE integrated boiler, the exhaust gases temperature is function only of the evaporative pressure and pinch point assumed, as reported in Table 6.

Table 6 : WTE outlet temperature as function of evaporative pressure

p_{ev} [bar]	$T_{O,WTE}$ [°C]
40	260
50	274
80	305

Main results of the proposed layout, in correspondence to optimum plant match condition (point B), are summarized in Table 7. The optimum Q^*_{EXH}/F_W found in correspondence to an evaporative pressure of 50 bar is below the unit (0.88). This means that the thermal power introduced with waste is greater that that introduced with GT exhaust gases thus, the biggest supply in terms of thermal powers introduced is assigned to WTE section.
The main negative aspect for the proposed layout is the high WTE exhaust gases temperature: a relative high amount of heat is discharged from WTE boiler without being completely exploited down to the minimum allowed temperature (160 °C).

Table 7 : Layout_1 main results.

	Layout_1
F_W [MW]	100
p_{ev} [bar]	50
$T_{O,GT}$ [°C]	500
T_{SH} [°C]	480
m_{exh} [kg/s]	152
Q^*_{EXH} [MW]	87.68
Q^*_{EXH}/F_W [-]	0.88
$m_{s,max}$ [kg/s]	49.12
$T_{O,HRSG}$ [°C]	143
$T_{O,WTE}$ [°C]	274
P_{ST} [MW]	45.76

LAYOUT 2 and LAYOUT_2bis

Next proposed layout (namely Layout_2) is schematically shown in Figure 15. As suggested from the figure, two economizer sections are present respectively in the WTE and in the HRSG section. Water out of deaerator is firstly fed to the ECO1, placed in the WTE convective pass, where part of the economization is performed; economization is completed in the ECO2, inside the HRSG section.

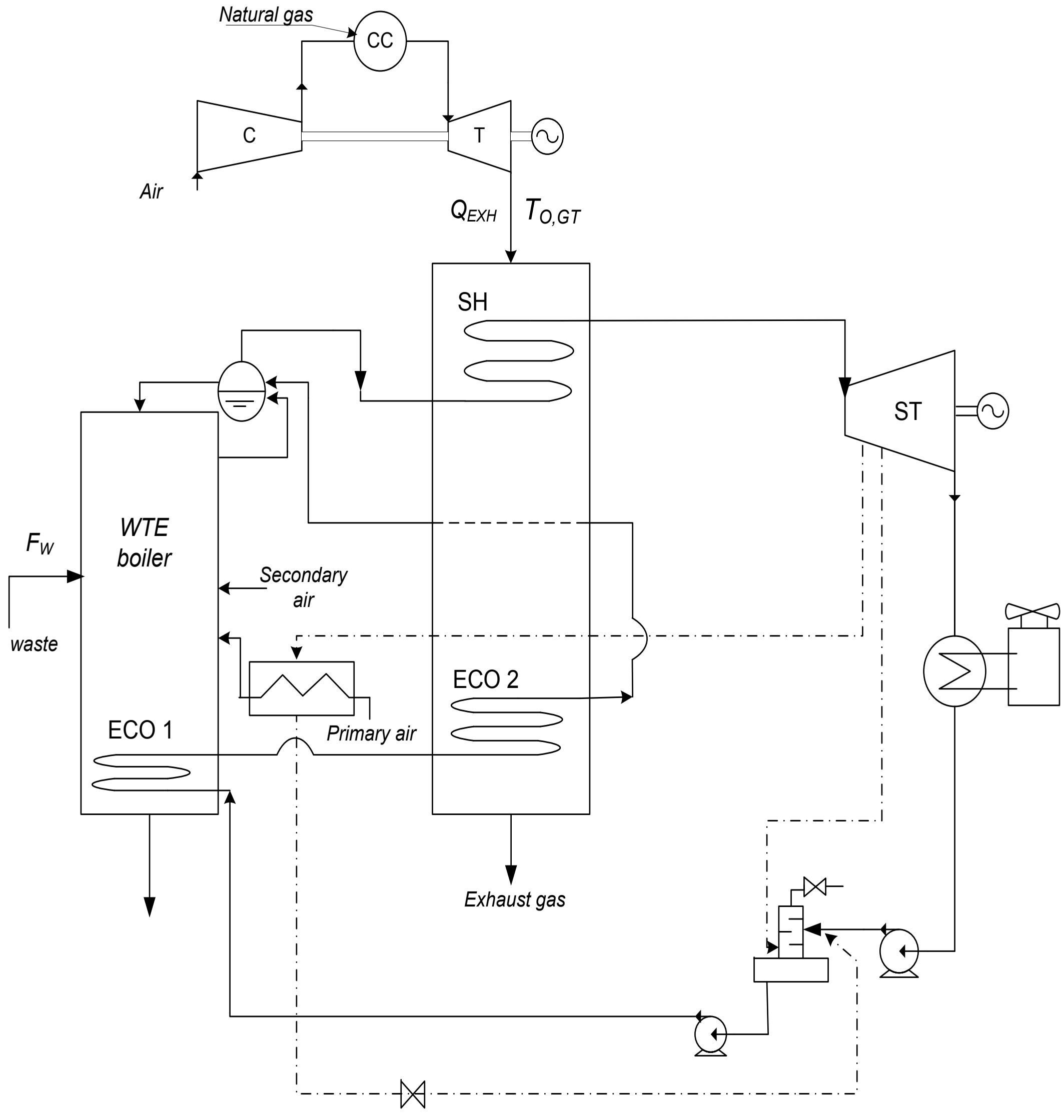

Figure 15 : Layout_2

The idea to share water economization comes from results of the Layout_1 trying to exploit the thermal power available in the boiler, minimizing WTE outlet temperature.

If compared to the previous layout, a considerable reduction of Q^*_{EXH} has been found in correspondence to the optimum point (point B'', Figure 16). Optimum plant match condition is found in correspondence to $\left(Q^*_{EXH}\right)'$ equal to 81.9 MW, A decrease in Q^*_{EXH} equal to 6.6%, with respect to Layout_1, has been obtained while the maximum amount of steam produced still remains the same. Focusing on steam production, for a fixed evaporative pressure and steam superheated temperature, Layout_2 shows also an increase in steam mass flow rate, with the respect of Layout_1, for Q_{EXH} lower than optimum value (see Figure 16). The increase in steam mass flow can be explained considering the presence of a first economizer section in the WTE: allowing an increase in water inlet temperature at ECO 2, for a fixed value of Q_{EXH} , increases the amount of generated steam. On the contrary, once reached the optimum condition, being the amount of generated steam determined only by thermal power available with waste F_W , the two layouts reache the same maximum value.

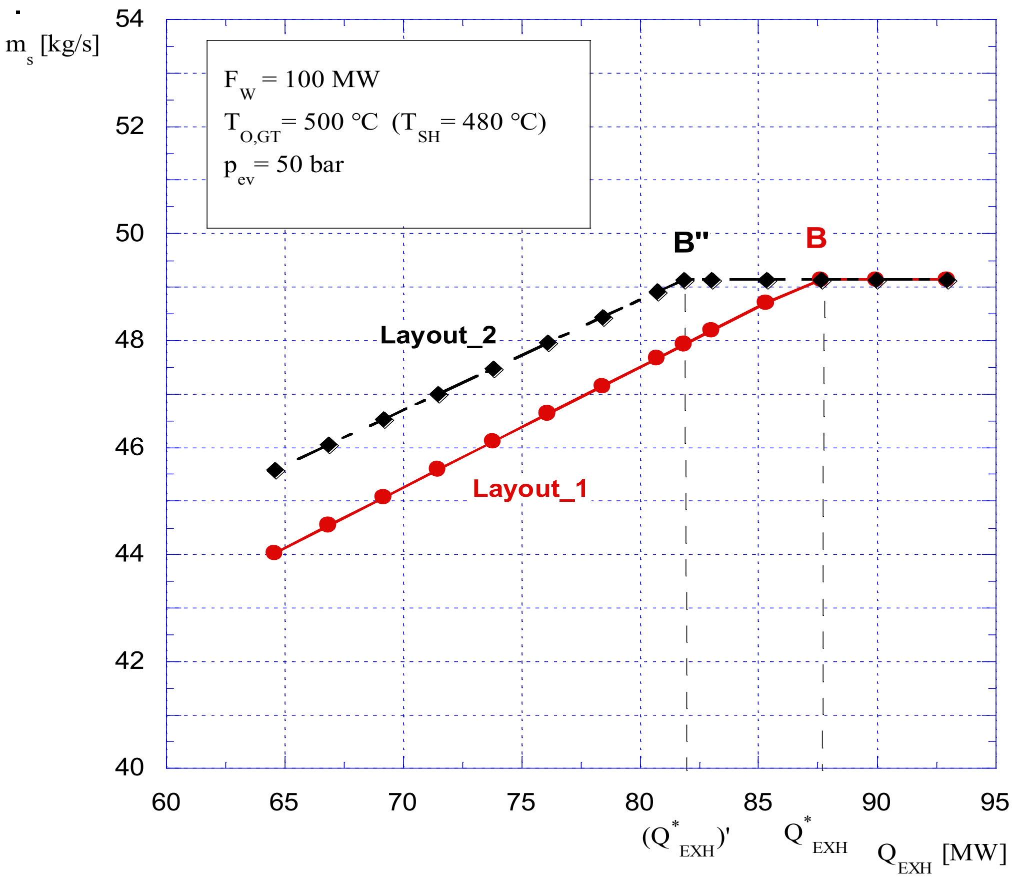

Figure 16 : steam mass flow rate for Layout_1 and 2 as function of GT discharged thermal power.

On the contrary, in correspondence to optimum value, an increase in HRSG outlet temperature has been found if compared to Layout_1. In this configuration, water enters ECO 2 with a higher temperature. The T-Q diagram for HRSG section of Layout_2 is shown in Figure 17. The 71 % of the thermal power introduced in HRSG is exploited, of which 60% in the SH section.

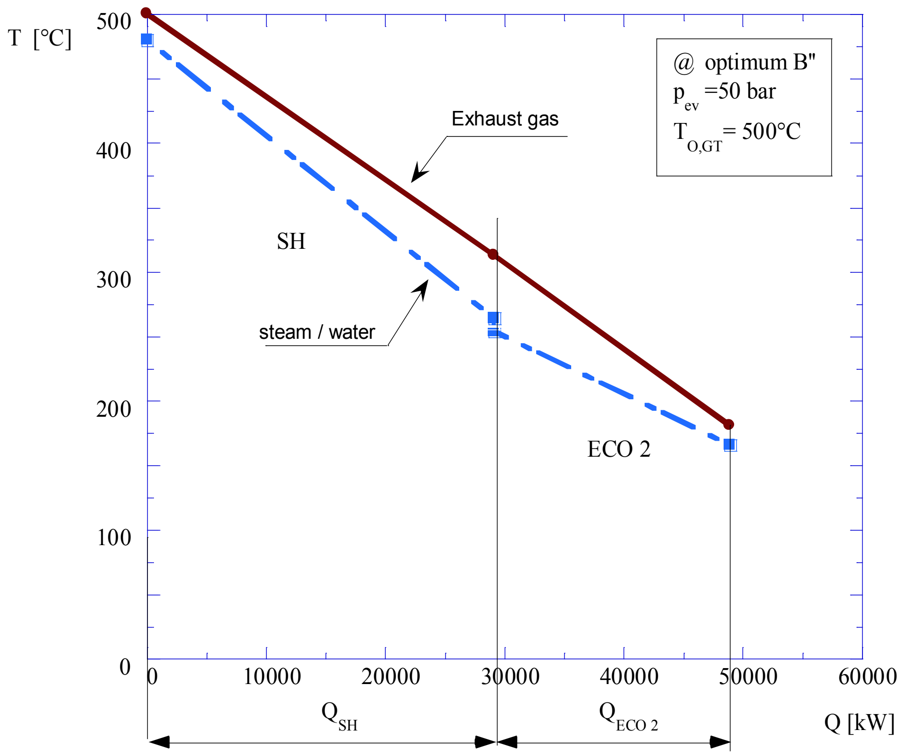

Figure 17 : Layout_2 T-Q diagram for HRSG section.

Figure 18 shows an alternative possibility to share water economization between the two sub-systems, namely Layout_2bis. In the considered configuration, water, out of deaerator, is firstly fed to the ECO2 (placed in the HRSG) where part of the economization is performed to be completed in the ECO1, inside the WTE section.

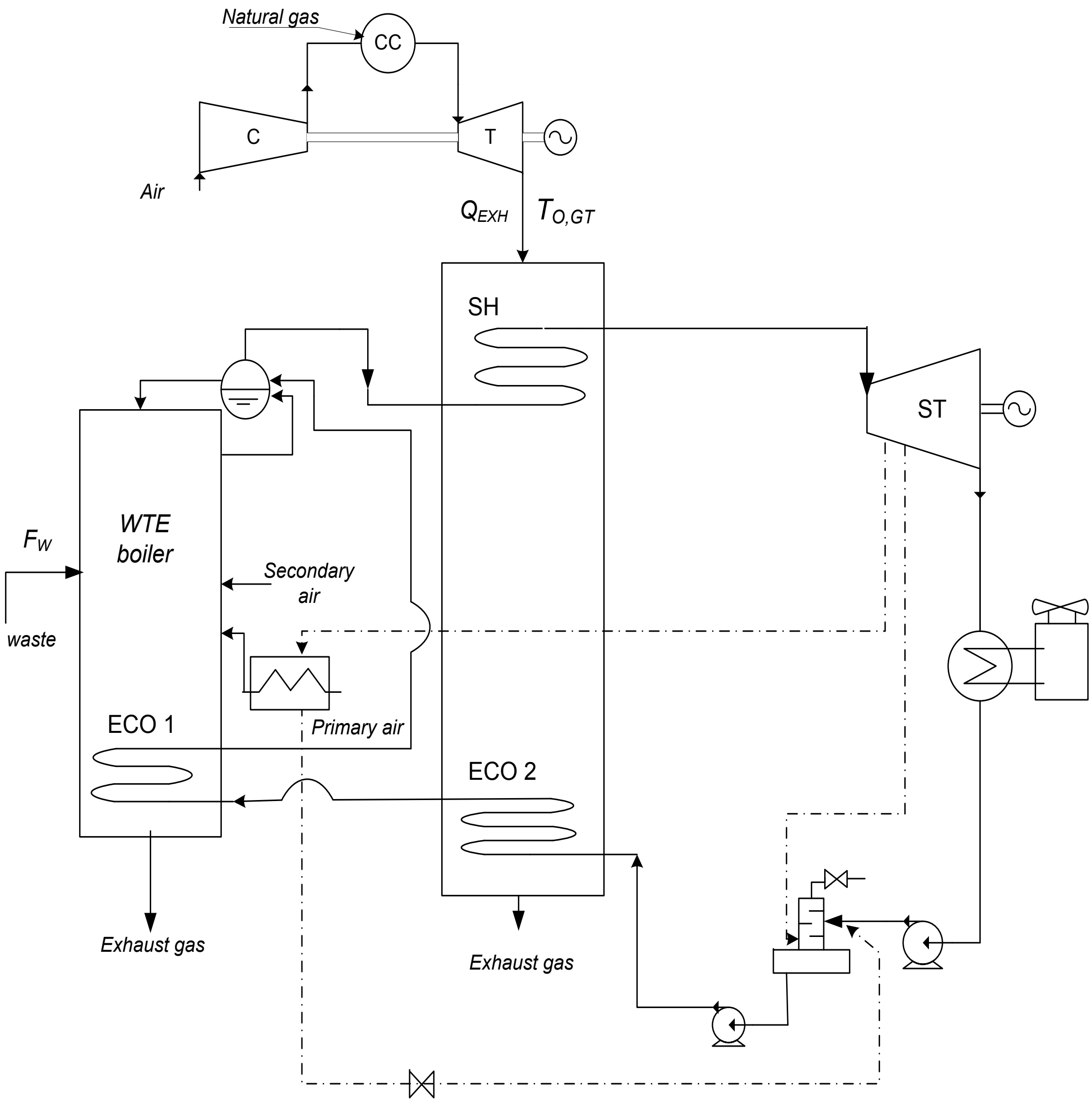

Figure 18 : Layout_2bis

The T-Q diagram relative to Layout_2bis is shown in Figure 19. In this case, on the respect of Layout_2, a lower HRSG discharged temperature is obtained. Comparing steam mass flow rate production for the proposed layouts (Figure 20) it can be observed that Layout_2bis optimum condition (B') is comprised within point B'' (optimum value for Layout_2) and B (optimum value for Layout_1). While, for all the proposed configurations, the maximum amount of generated steam mass flow rate still remains the same.

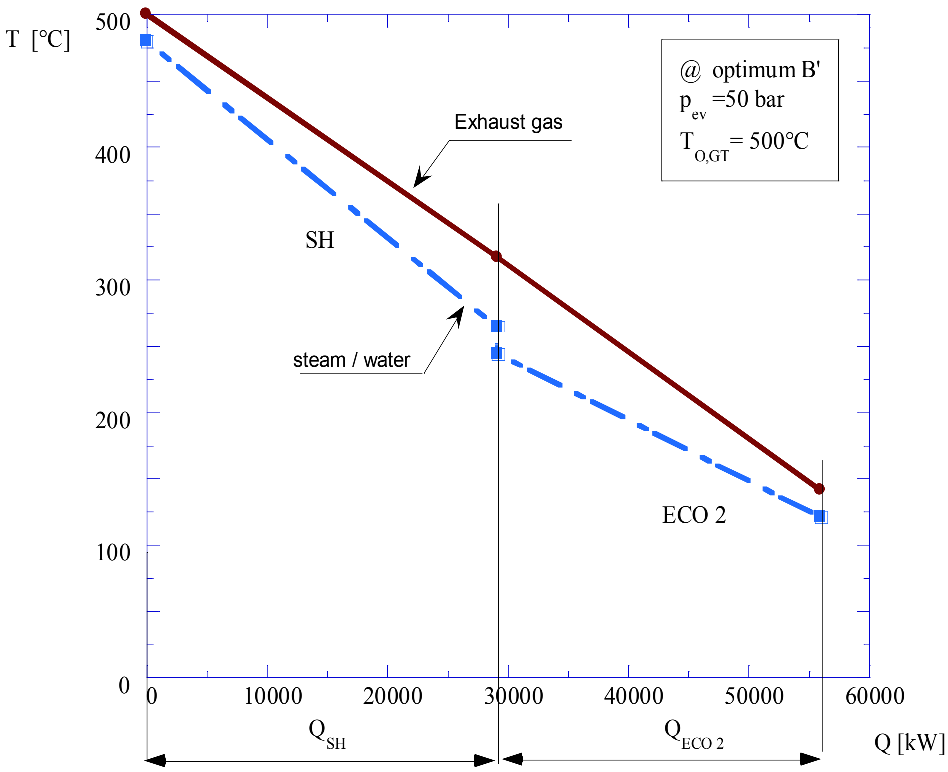

Figure 19 : Layout_2bis T-Q diagram for HRSG section.

Main results of Layout_2 and 2bis are summarized in Table 8. It can be noticed that, in correspondence to optimum conditions, Layout_2 reaches the minimum allowed WTE exhaust gases temperature while HRSG outlet temperature turns out to be considerably higher than the minimum allowed value.

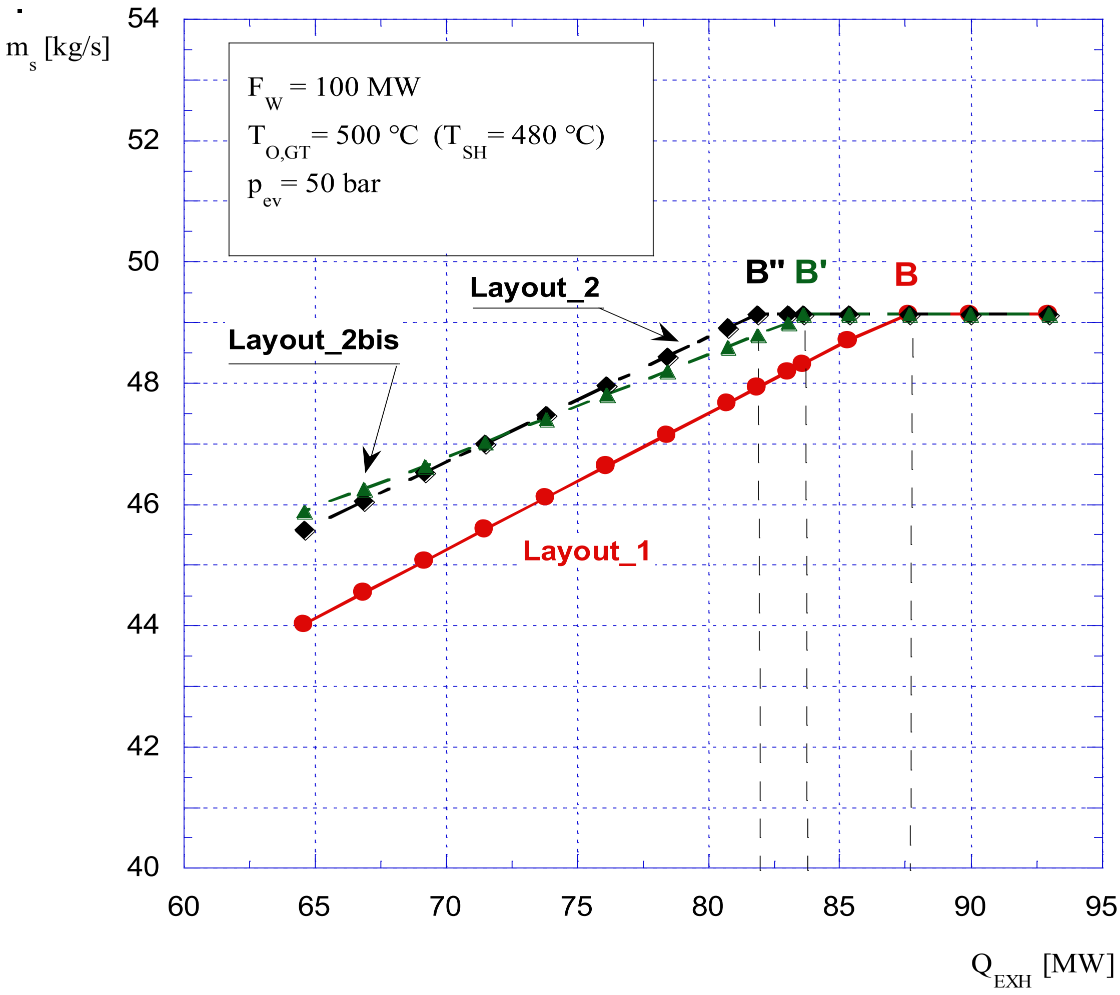

Figure 20 : steam mass flow rate for Layout_1, 2 and 2bis as function of GT discharged thermal power.

Table 8 : Layout_2 and Layout_2bis results

	Layout_2	Layout_2bis
F_W [MW]	100	100
p_{ev} [bar]	50	50
$T_{O,GT}$ [°C]	500	500
T_{SH} [°C]	480	480
m_{exh} [kg/s]	142	145
Q^*_{EXH} [MW]	81.91	83.64
Q^*_{EXH}/F_W [-]	0.82	0.84
$m_{s,max}$ [kg/s]	49.12	49.12
$T_{O,HRSG}$ [°C]	181	141
$T_{O,WTE}$ [°C]	160	246
P_{ST} [MW]	45.76	45.76

LAYOUT_3

In the proposed layout, a parallel configuration between economizer sections is evaluated. Water, out of deaerator is divided into two streams: a fraction (f) goes into ECO 2, the economizer section inside HRSG, while the remaining part is sent to ECO1, inside WTE. Before entering the WTE boiler, a mixer, combining both streams, is present. Water mass flow rate splitting is adjusted in order to have similar mixer inlet temperatures (or economizer outlet temperatures), trying to maximize the exploitation of available thermal power minimizing exhaust gases temperatures.

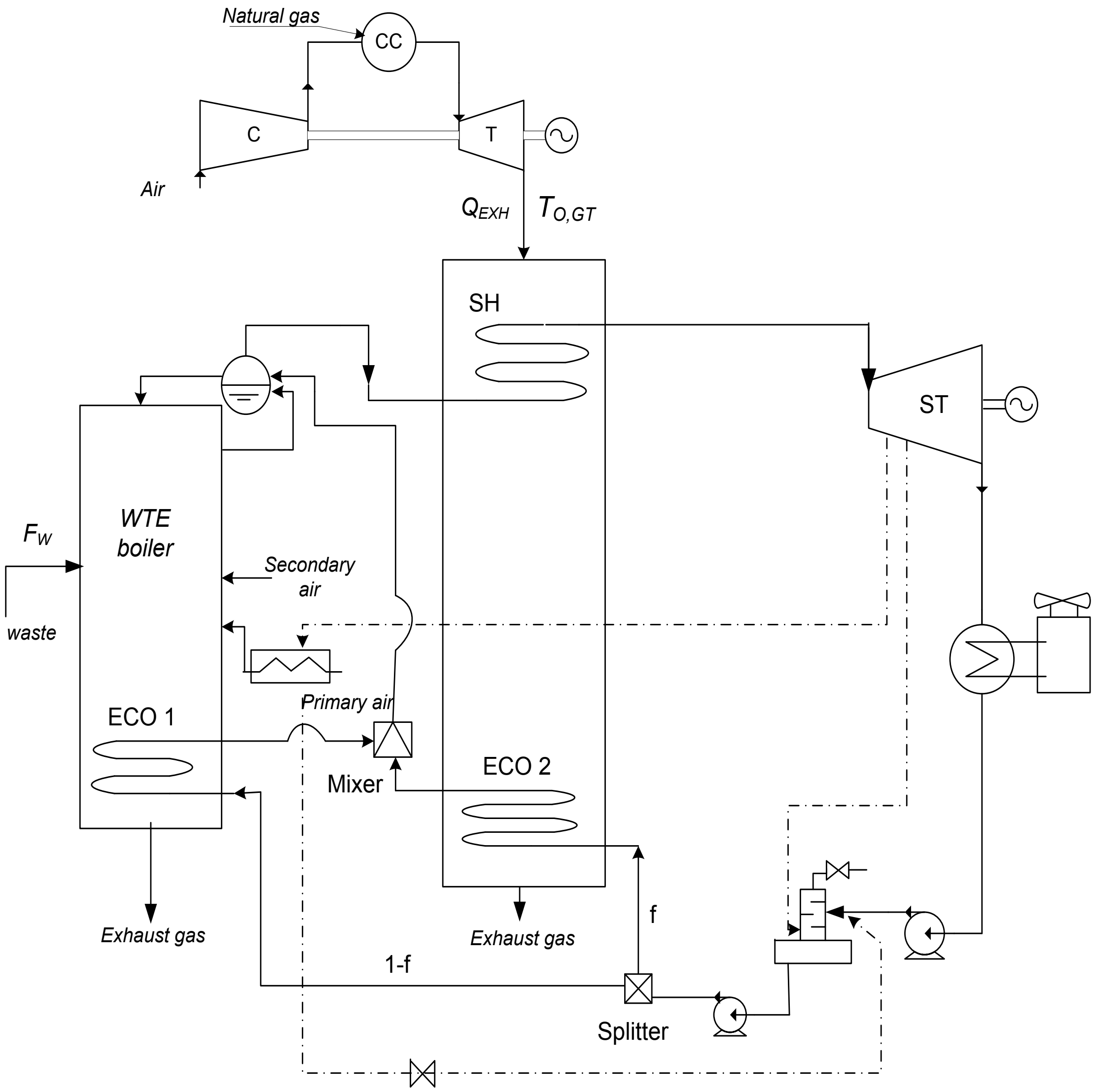

Figure 21 : Layout_3

In correspondence to the optimum condition, for an evaporative pressure equal to 50 bar and a GT outlet temperature of 500 °C, the following water mass flow rate splitting has been found: 70% of the total water mass flow is sent to ECO2 inside HRSG. In correspondence to optimum Q^{*}_{EXH}, the assumed water mass flow rate splitting allows to reach the same water minimum sub-cooling temperature difference of both streams. Below the optimum value (point B''', Figure 22), due to a decrease of Q_{EXH}, a slight decrease of f must occurs, increasing water mass flow rate economized into WTE section.

As highlighted in Figure 22, the proposed configuration allows a considerable decrease of Q^{*}_{EXH} (about 11% of decrease with the respect of Layout_1). Moreover, for $Q_{EXH} < Q^{*}_{EXH}$, Layout_3 achieves the best performance in terms of generated steam mass flow rate.

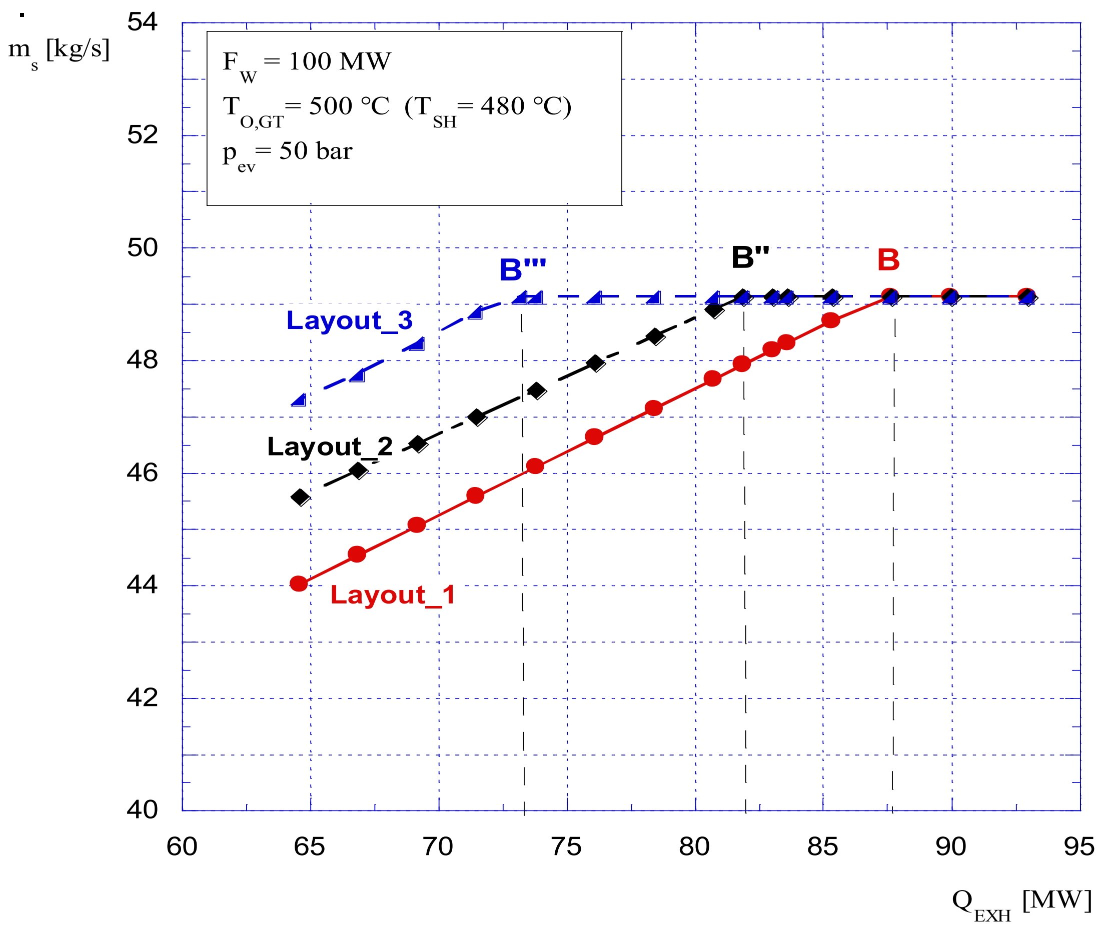

Figure 22 : steam mass flow rate for Layout_1, Layout_2 and Layout_3 as function of GT discharged thermal power.

Focusing on T-Q diagram for Layout_3, Figure 23, a reduction of the distance between the exhaust gas and steam/water lines, with respect to previous layouts can be observed; moreover, water mass flow rate splitting has been also optimized in order to obtain a parallelism between exhaust gas and water lines .

Main results of Layout_3 are summarized in Table 9. It can be noticed that, in correspondence to optimum conditions, WTE exhaust gases temperature is close to reach its minimum value.

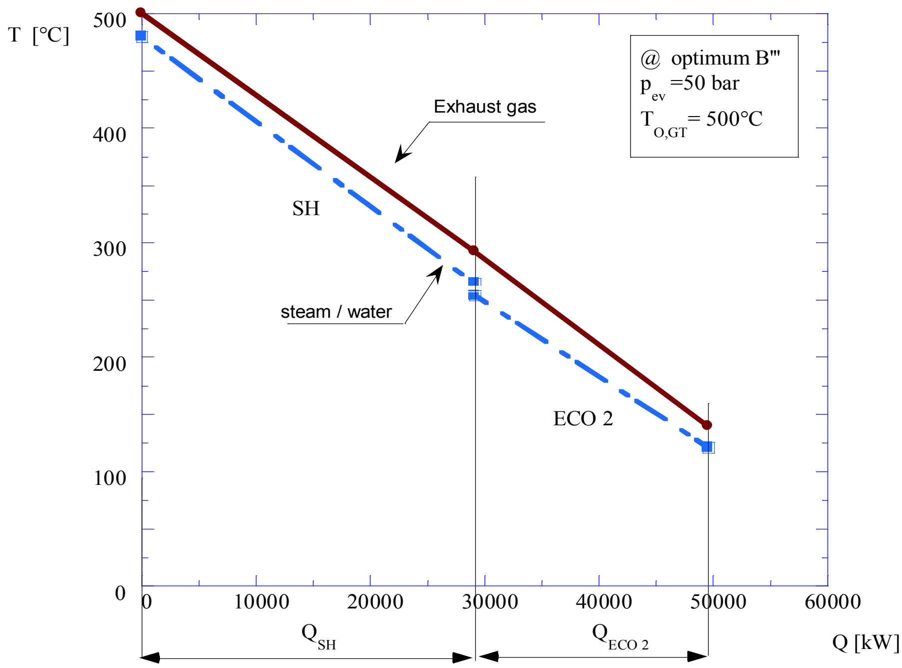

Figure 23 : Layout_3 T-Q diagram for HRSG section.

Table 9 : Layout_3 results

	Layout_3
F_W [MW]	100
p_{ev} [bar]	50
$T_{O,GT}$ [°C]	500
T_{SH} [°C]	480
m_{exh} [kg/s]	128
Q^*_{EXH} [MW]	73.83
Q^*_{EXH}/F_W [-]	0.74
$m_{s,max}$ [kg/s]	49.12
$T_{O,HRSG}$ [°C]	140
$T_{O,WTE}$ [°C]	168
P_{ST} [MW]	45.76

LAYOUT_4

Starting from Layout_2, the possibility to further exploit the HRSG discharged thermal power has been evaluated in next proposed configuration, namely Layout_4. Starting from results of Layout_2, in terms of HRSG outlet temperature, the possibility to preheat waste combustion air has been investigated in this layout (Figure 24). At the exit of economizer section, a heat exchanger has been introduced fed by HRSG discharged thermal power. This solution allows to eliminate the steam turbine bleed necessary for air preheating.

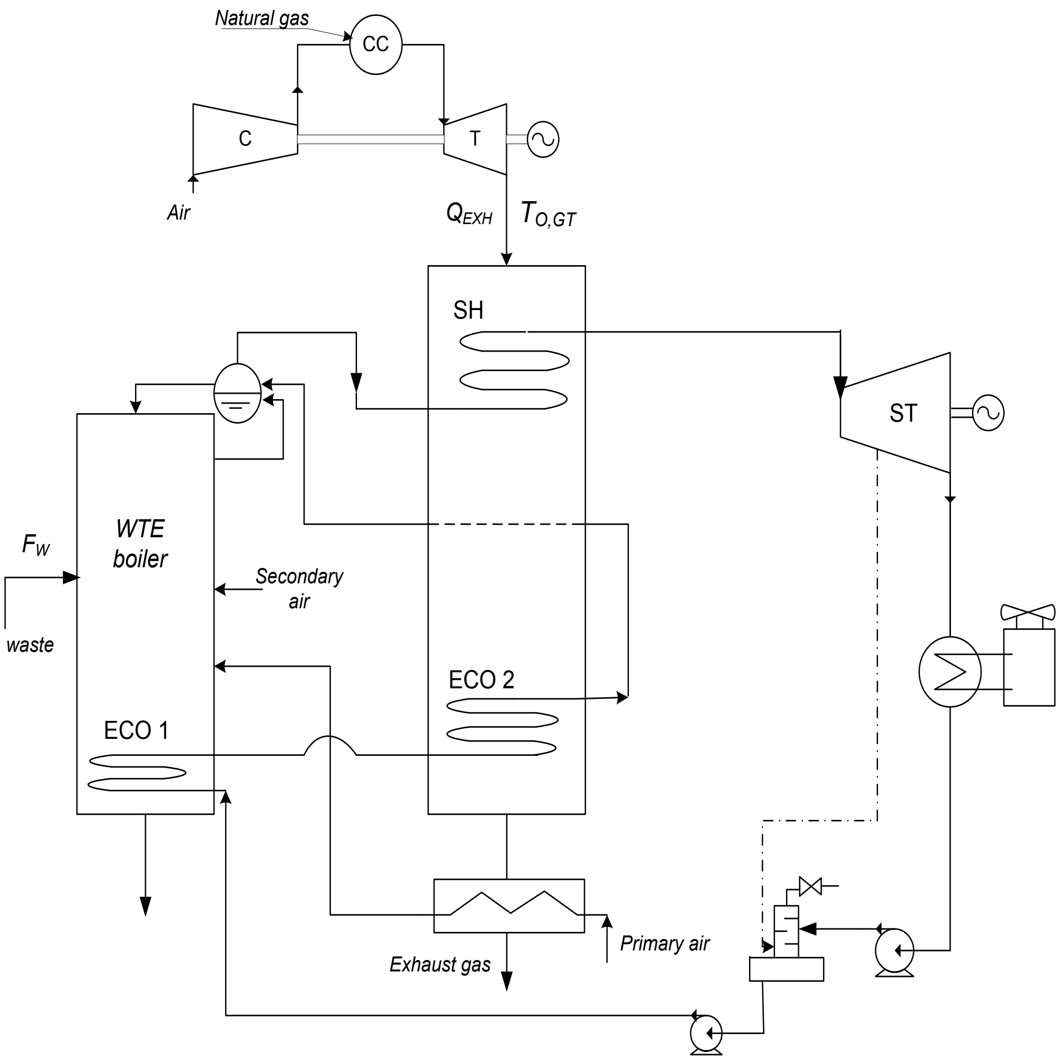

Figure 24 : Layout_4

Layou_4 refers to preheating of only primary combustion air up to a temperature been set equal to 150°C while temperature of secondary combustion air is left equal to 50 C. Preheating of both, primary and secondary combustion air, has been investigated (namely Layout_4_tot_air) where both temperatures (primary and secondary combustion air) have been set equal to 150 °C.

Focusing on Figure 25, an increase in the maximum amount of steam mass flow rate has been observed in both cases, equal to about 1% and 3.5% respectively for Layout_4 and Layout_4_tot_air. For both layouts, an increase in steam mass flow rate for $Q_{EXH} < Q^{*}_{EXH}$ (points B^{IV}) is achieved with reference to Layout_1 and Layout_2. This can be explained considering the increase in combustion air temperature which, for the considered layouts reaches 150 °C with an increase equal to 20 °C on previous configurations (see Table 5), increases the reactants enthalpy therefore increasing the combustion temperature.

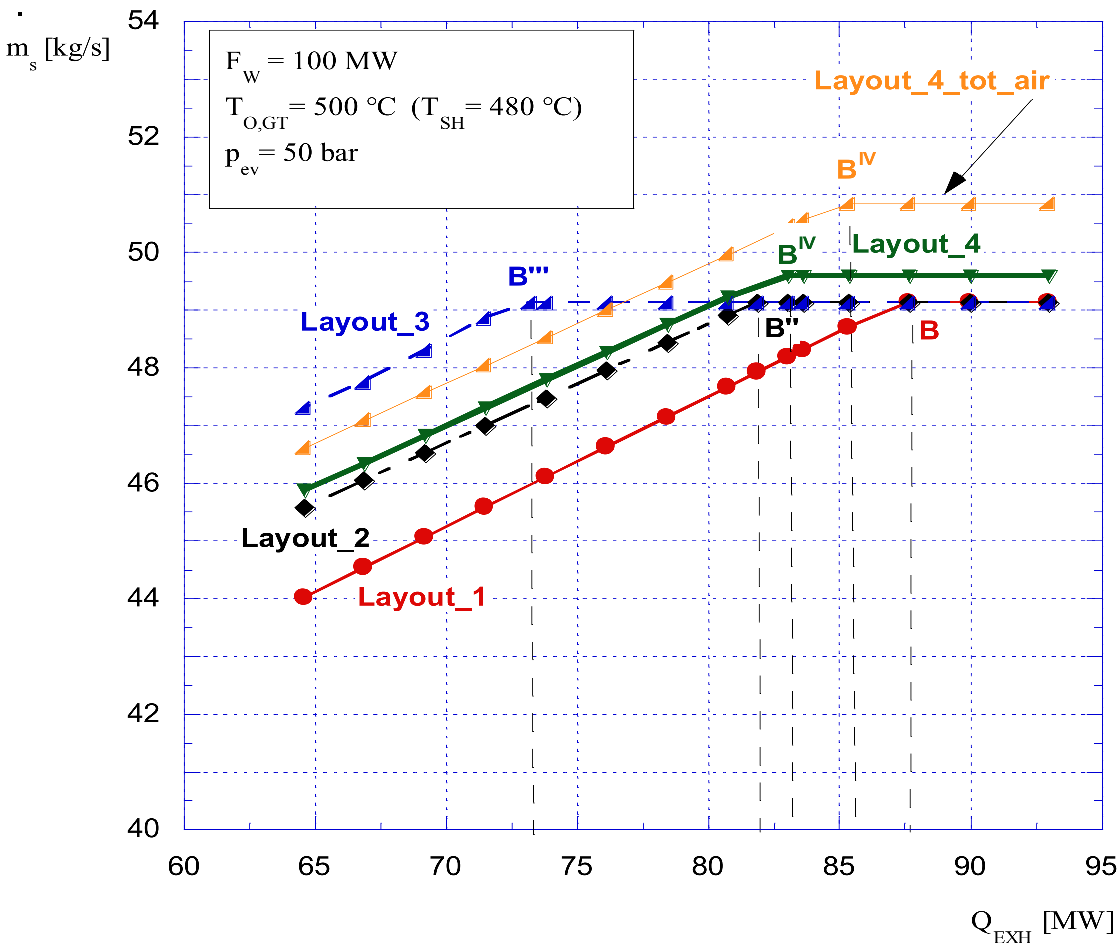

Figure 25 : steam mass flow rate for Layout_1, Layout_2 and Layout_3 as function of GT discharged thermal power.

As shown in Table 10, the increase in maximum steam mass flow rate and the elimination of ST bleed for air preheating allows an increase in ST power output.

Table 10 : Layout_4 and Layout_4_tot_air results

	Layout_4	**Layout_4_tot_air**
F_W [MW]	100	100
p_{ev} [bar]	50	50
$T_{O,GT}$ [°C]	500	500
T_{SH} [°C]	480	480
m_{exh} [kg/s]	144	148
Q^*_{EXH} [MW]	83.06	85.37
Q^*_{EXH}/F_W [-]	0.83	0.85
$m_{s,max}$ [kg/s]	49.57	50.84
$T_{O,HRSG}$ [°C]	158	144
$T_{O,WTE}$ [°C]	160	160
P_{ST} [MW]	46.69	47.88

LAYOUT_5

In the proposed layout (namely Layout_5, shown in Figure 26) the possibility to introduce an integral deaerator inside the HRSG section has been analyzed, thus eliminating ST bleed to feed it. Water out of condenser is send to a pre-economizer (ECO) placed into HRSG feeding the integral deaerator (EVA DEA). Water mass flow rate out of deaerator is then divided into two streams to be economized in parallel. Concerning water splitting between ECO1 and ECO2 the same considerations as for Layout_3 has been maintained.

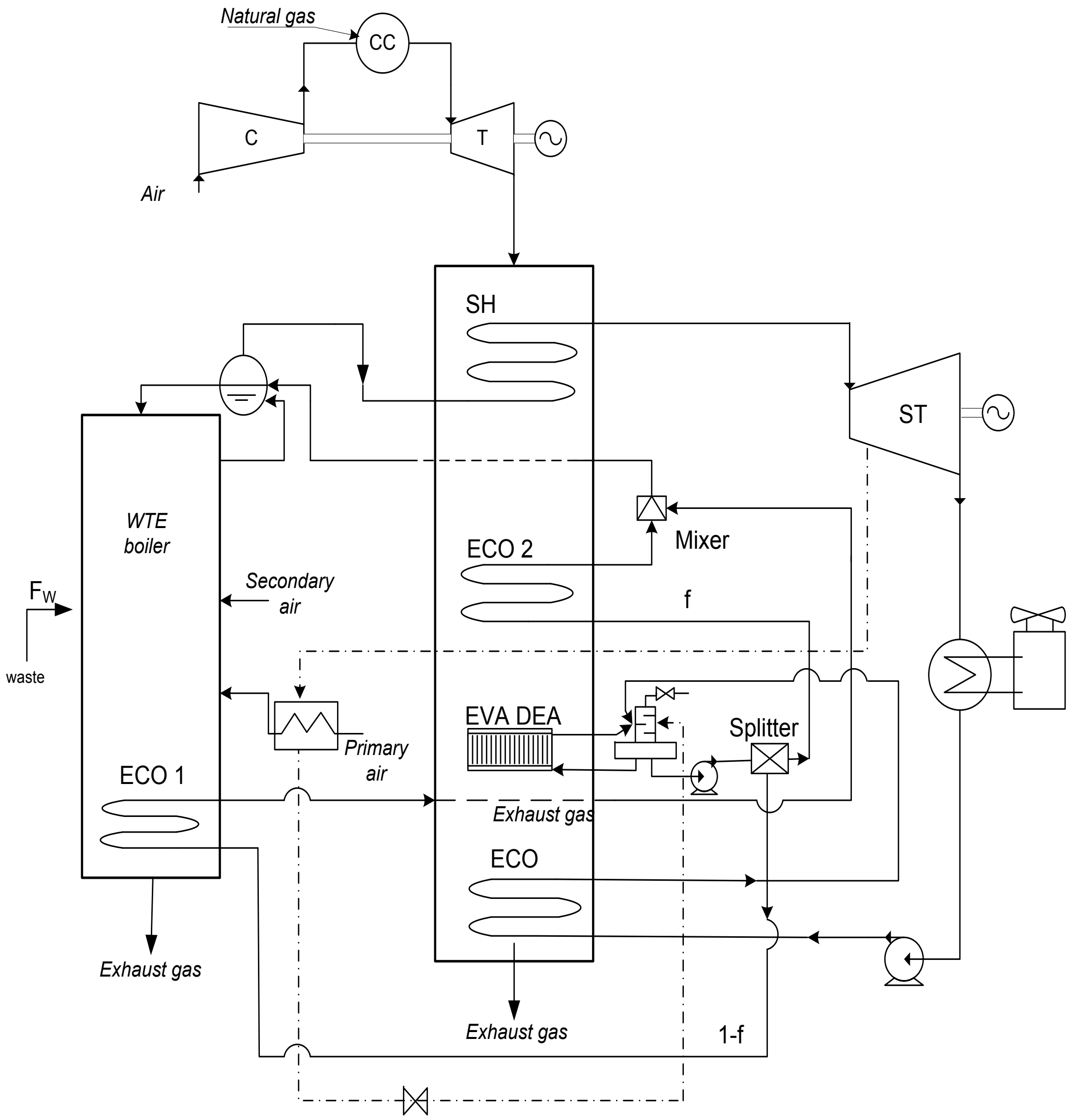

Figure 26 : Layout_5

In Figure 27 the steam mass flow rate produced in Layout_5 as function of GT discharged thermal power is shown. Comparing results of Layout_5 with other configurations, an increase in the optimum value of Q^*_{EXH} (point B^V) is found. Moreover, for $Q_{EXH} < Q^*_{EXH}$, Layout_5 has the lowest steam mass flow rate production. When the integrated system is operating below the optimum condition, water splitting between ECO2 and ECO1 deviates from the condition of point B^V (70% of the total water mass flow to ECO2): a decrease of Q_{EXH}, decreases the water mass flow rate that ECO2 can manage: in correspondence to the lowest considered value of Q_{EXH}, only 28% of the total mass flow rate is fed to ECO2.

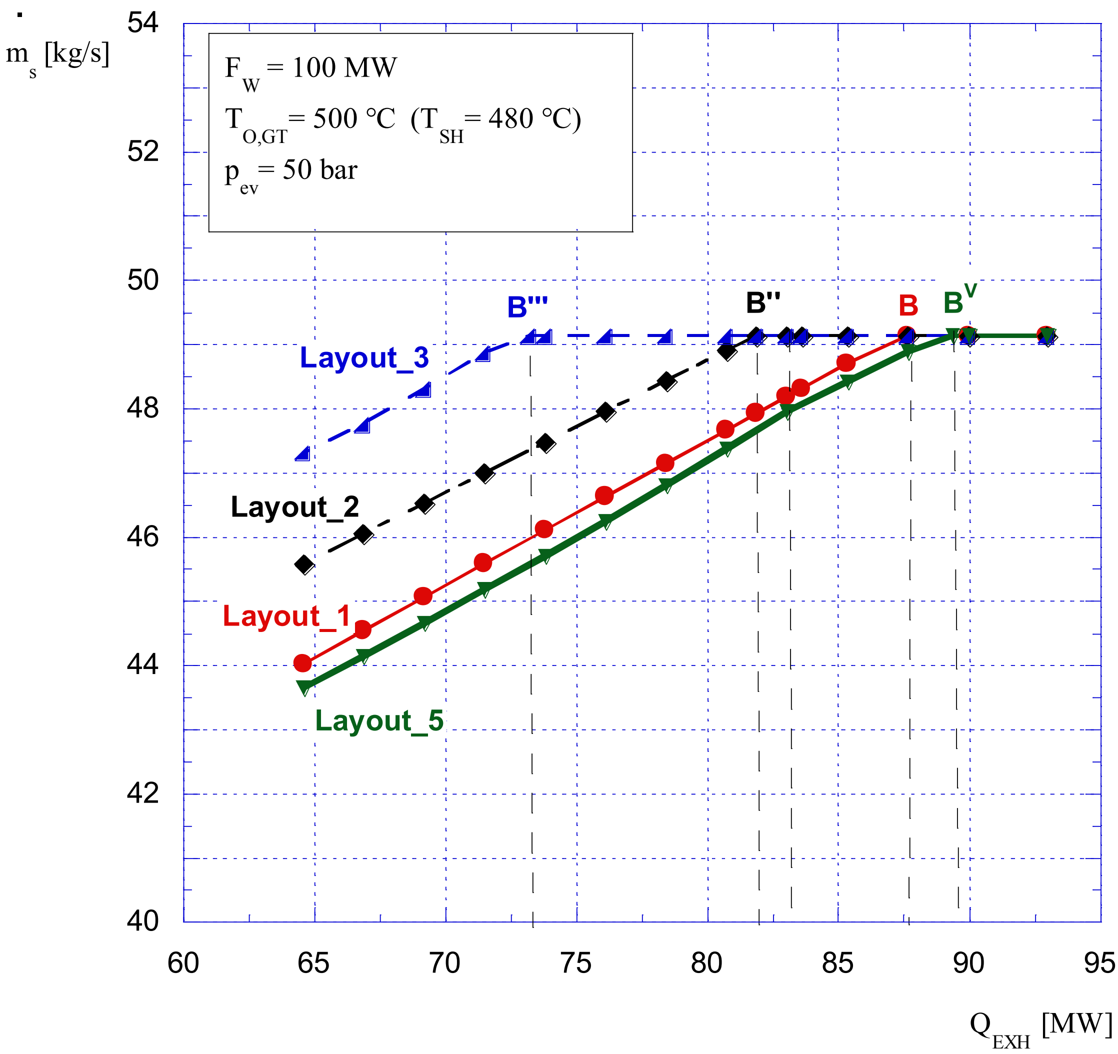

Figure 27 : steam mass flow rate for Layout_1, Layout_2, Layout_3 and Layout_5 as function of GT discharged thermal power.

Focusing on T-Q diagram in Figure 28, the proposed configuration, in correspondence to the optimum plant match condition (point B^V), allows to achieve the lowest HRSG outlet temperature (112 °C). Besides, the presence of the integral deaerator inside HRSG, increases the distance between exhaust gases and steam/water lines. As shown in Table 11, the elimination of the ST bleed to fed the deaerator, allows an increase in ST power output.

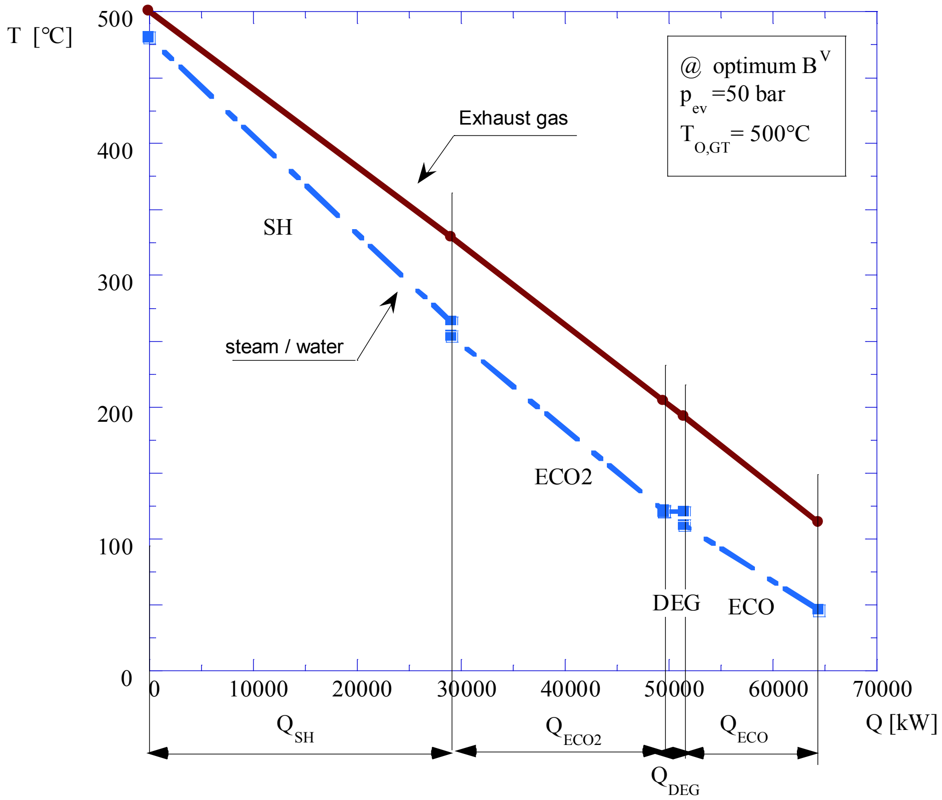

Figure 28 : T-Q diagram for HRSG section for Layout_5.

Table 11 : Layout_5 results

	Layout_5
F_W [MW]	100
p_{ev} [bar]	50
$T_{O,GT}$ [°C]	500
T_{SH} [°C]	480
m_{exh} [kg/s]	155
Q^*_{EXH} [MW]	89.41
Q^*_{EXH}/F_W [-]	0.89
$m_{s,max}$ [kg/s]	49.12
$T_{O,HRSG}$ [°C]	112
$T_{O,WTE}$ [°C]	168
P_{ST} [MW]	47.97

LAYOUT_6

In Layou_6, shown in Figure 29, a steam turbine bleed to preheat feed water before entering the economizer (ECO) is present. Out of deaerator, water economization is firstly performed into WTE (ECO1) to be completed inside HRSG section (ECO 2).

ST bleed for water preheating is set equal to 1.1 bar, this value is chosen not to penalize ST power output and to improve T-Q diagram for HRSG. Even in this layout, an integral deaerator (EVA DEA) is present.

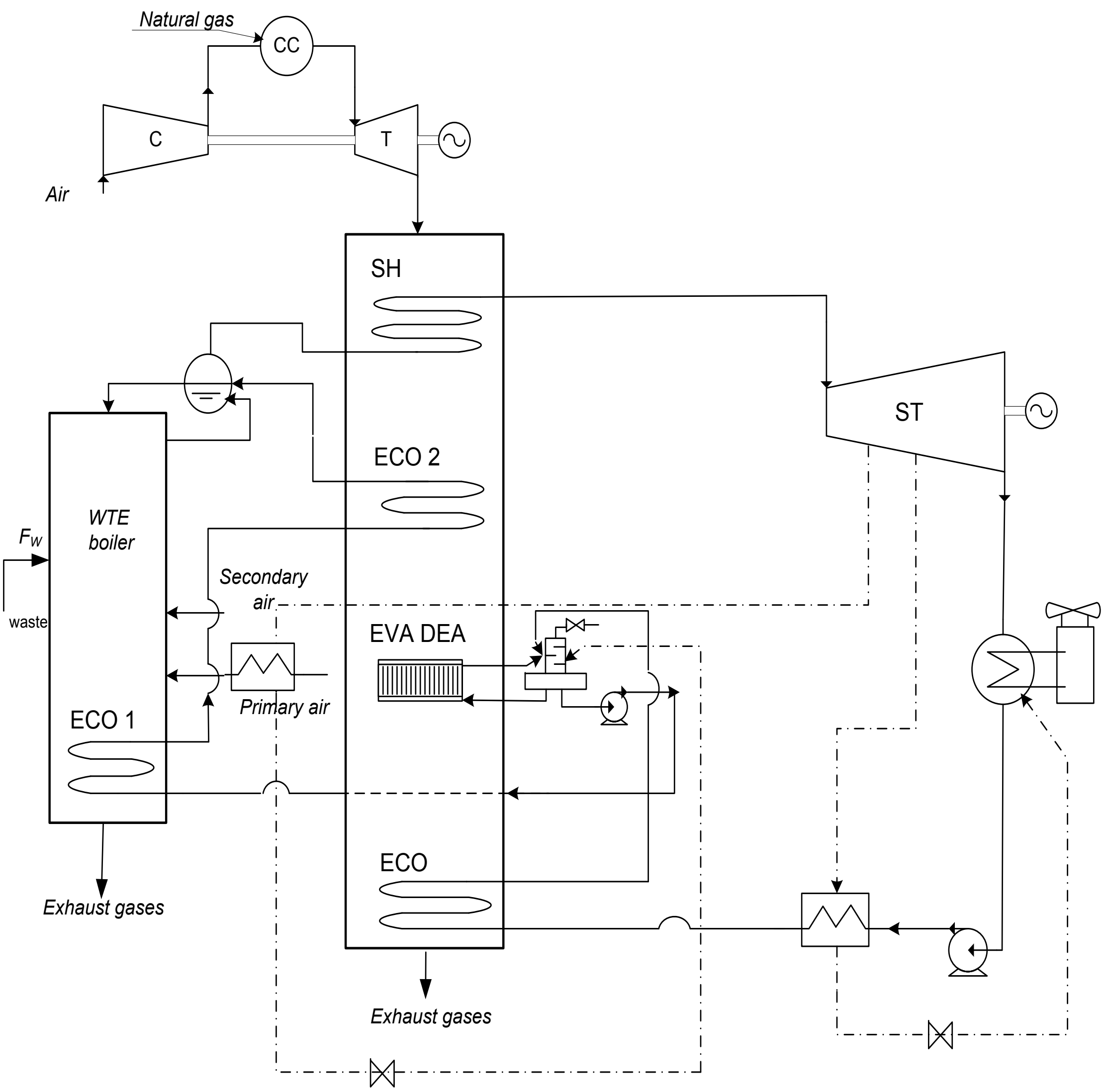

Figure 29 : Layout_6

Starting from a deaerator pressure equal to 2 bar, working pressure has been changed in order to improve HRSG T-Q profile (minimize the distance between exhaust gas and steam/water lines and decrease HRSG outlet temperature); an optimized pressure value equal to 3.5 bar has been found. The configuration obtained with the new value of deaerator pressure refers to Layout_6bis. T-Q diagrams relative to Layout_6 and Layout_6bis are shown respectively in Figure 30 and Figure 31.

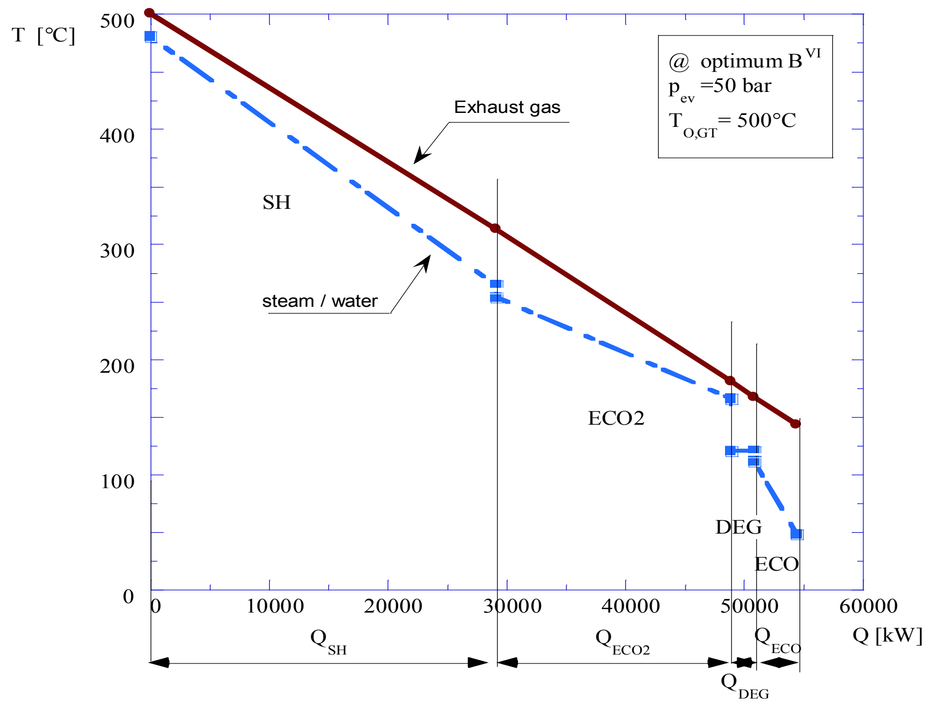

Figure 30 : T-Q diagram for HRSG section for Layout_6.

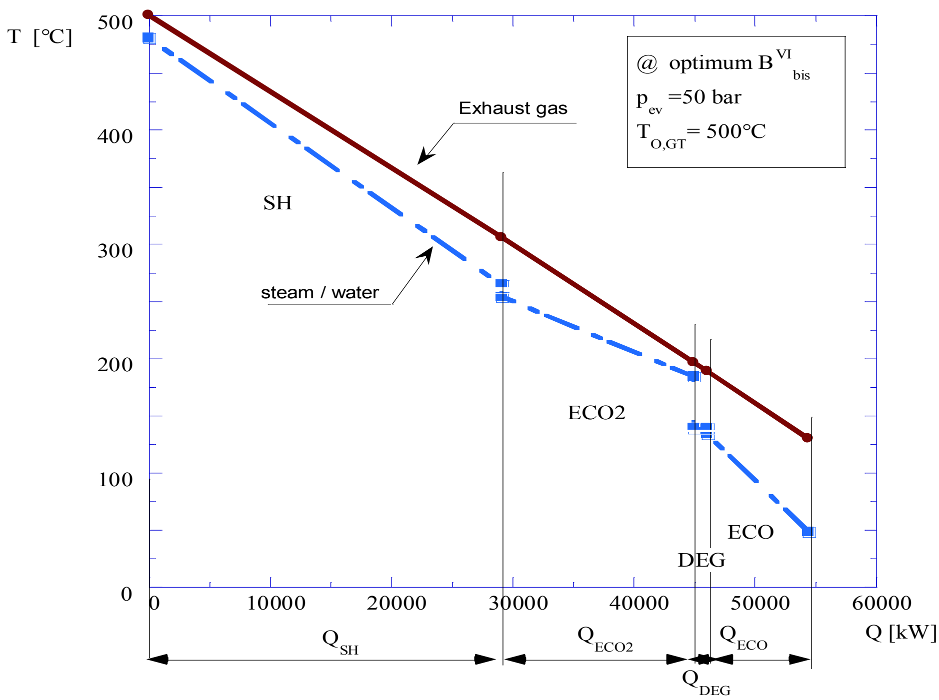

Figure 31 : T-Q diagram for HRSG section for Layout_6bis.

As shown in Figure 32, the improved layout leads to a decrease in Q^*_{EXH} optimum value (point B^{VI} bis < B^{VI}) and to an increase in generated steam mass flow for $Q_{EXH} < Q^*_{EXH}$.

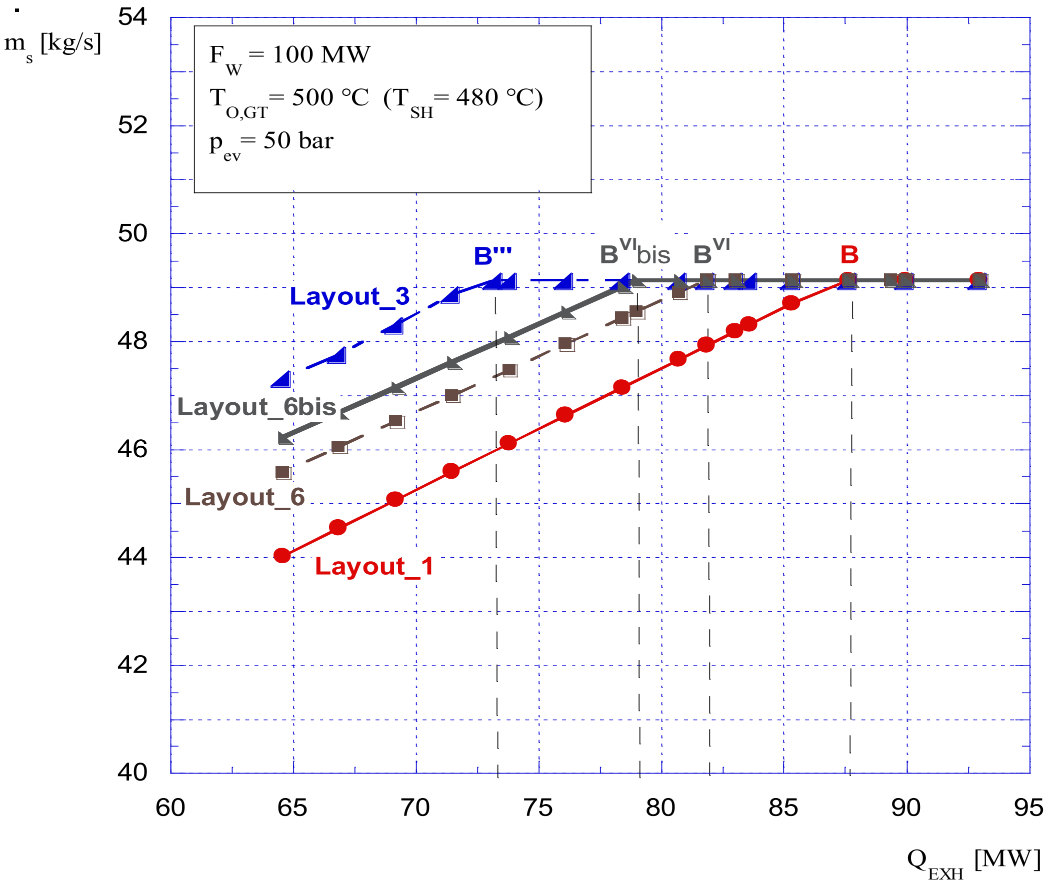

Figure 32 : steam mass flow rate for Layout_1, Layout_3, Layout_6 and Layout_6bis as function of GT discharged thermal power.

Table 12 : Layout_6 and Layout_6bis results

	Layout_6	**Layout_6bis**
F_W [MW]	100	100
p_{ev} [bar]	50	50
$T_{O,GT}$ [°C]	500	500
T_{SH} [°C]	480	480
m_{exh} [kg/s]	142	137
Q^*_{EXH} [MW]	81.91	79.02
Q^*_{EXH}/F_W [-]	0.82	0.79
$m_{s,max}$ [kg/s]	49.12	49.12
$T_{O,HRSG}$ [°C]	143	130
$T_{O,WTE}$ [°C]	160	160
P_{ST} [MW]	46.72	46.72

LAYOUT_7

In the last proposed layout for a one pressure level integrated power plant, namely Layout_7, Figure 33, the possibility to generated a fraction of the total saturated steam in the HRSG section is investigated. The proposed configuration can be interesting, in particular, considering: the system transient behaviour, LHV deviation from the design value and different system start-up times. In Layout_7 an evaporative heat exchanger (EVA 2) is placed into the heat recovery boiler unit thus, a fraction of the total saturated steam is here generated. Water out of condenser, before entering the deareator, is divided into two streams: one part (h) is preheated in the ECO section inside HRSG while, the remaining (1-h), is sent to a heat exchanger fed by a ST bleed. Out of deaerator, water economization is performed in parallel: one part (f) is sent to the WTE economiser (ECO 1) while, the other one (1-f) is sent to another economiser section inside HRSG (ECO 2); economized streams are mixed and then splitted again before entering the evaporators sections. Splitting of water entering the economizer sections is adjusted, in order to exploit all the available heat inside HRSG. Before entering the superheated heat exchanger saturated steam streams are mixer together.

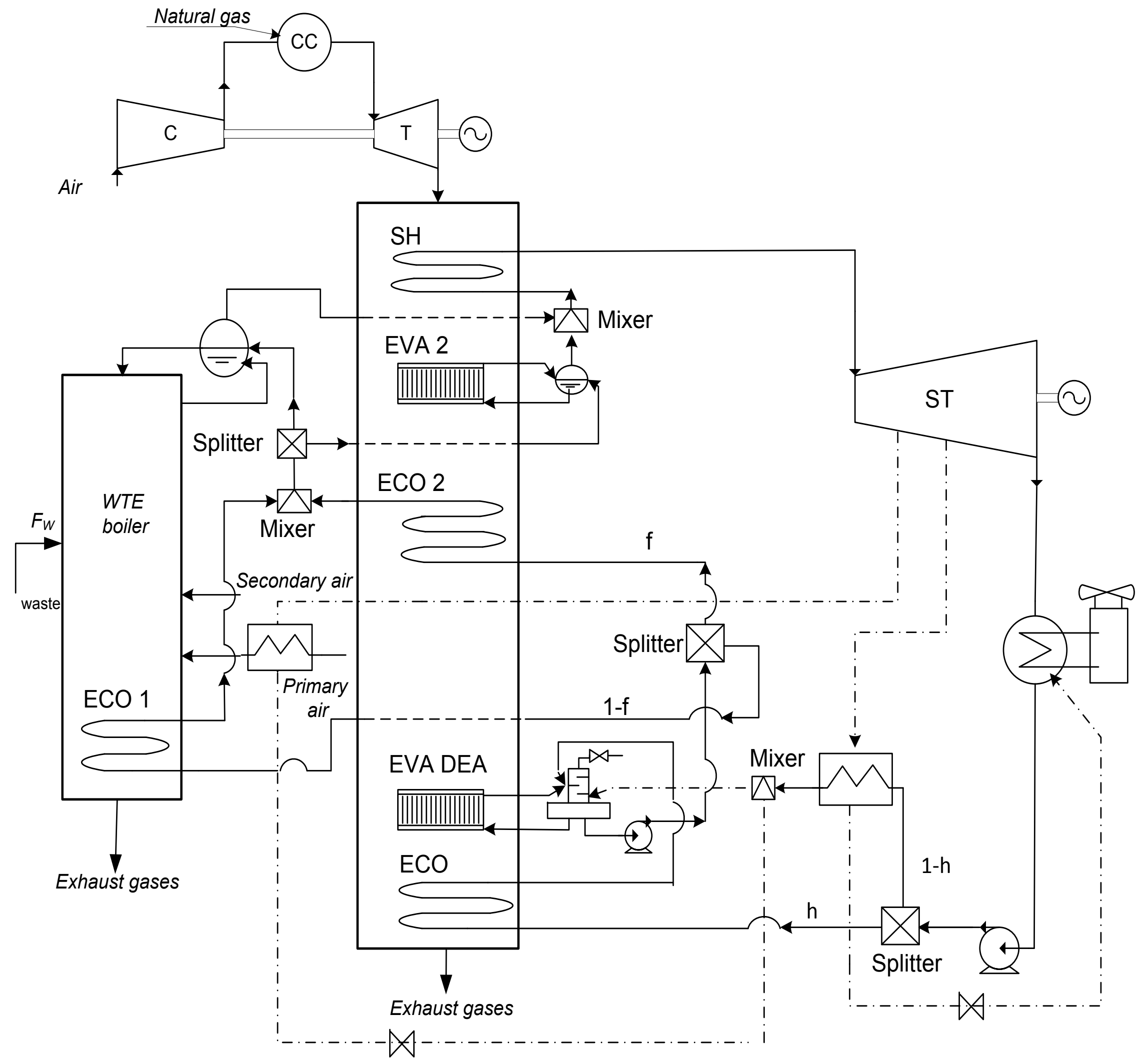

Figure 33 : : Layout_7

On the contrary of previous analyzed configurations, for Layout_7 there is no optimal condition in terms of Q_{EXH}; in fact, increasing the value of Q_{EXH} the total steam mass flow rate (sum of steam generated into WTE boiler and into the HRSG section) always increase. Instead, a minimum value of GT discharged thermal power is required to have steam generation inside the HRSG section. Figure 34 shows steam mass flow rate generated in the WTE and HRSG section plus the total amount as function Q_{EXH}; while, in Figure 35 the total amount of steam mass flow rate generated for Layout_7 is presented as function of GT discharged thermal power. As highlighted in both figures, an increase in Q_{EXH} increases $\dot{m}_{s,HRSG}$, thus increasing the total amount ($\dot{m}_{s,TOT}$) of generated steam.

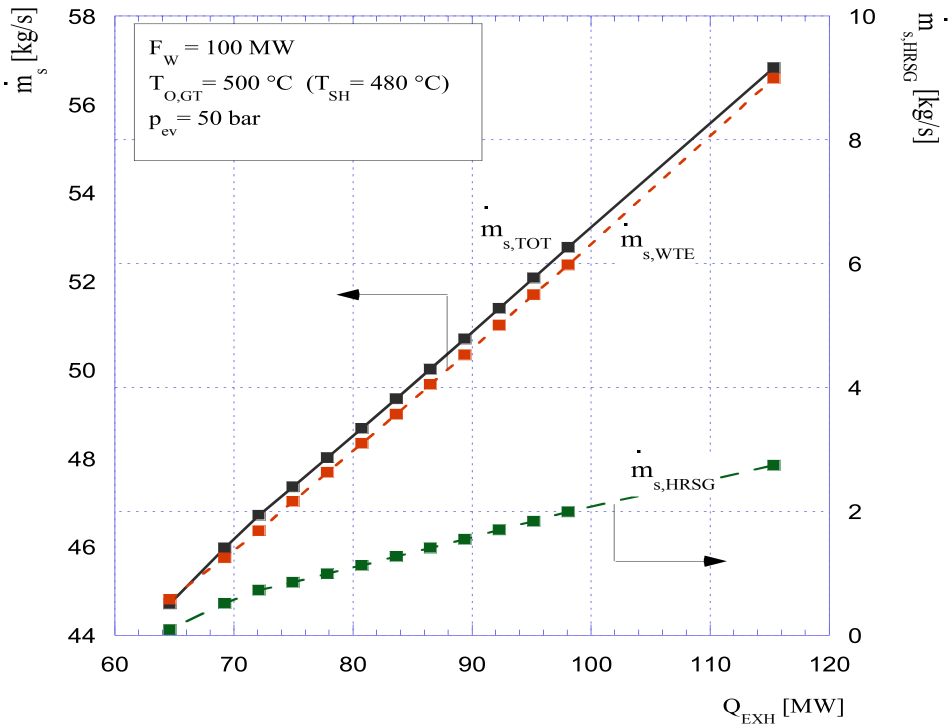

Figure 34 : steam mass flow rate for Layout_7 as function of GT discharged thermal power.

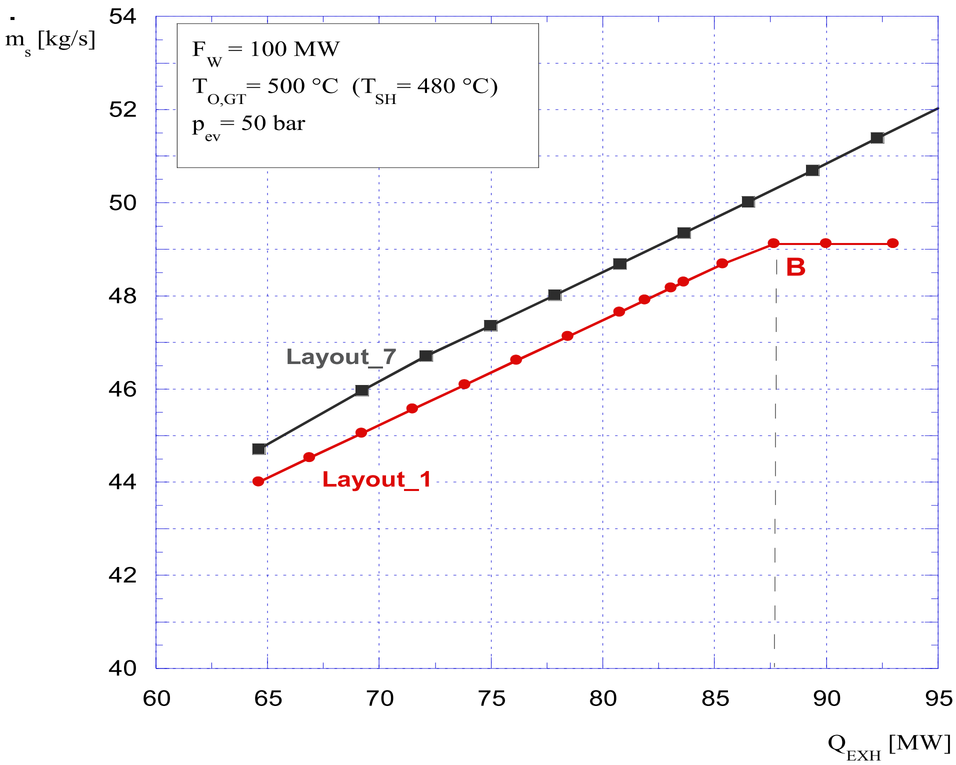

Figure 35 : steam mass flow rate for Layout_1, and Layout_7 as function of GT discharged thermal power.

The proposed configuration, allows to achieve the minimum HRSG outlet temperature (see T-Q diagram of Figure 36). Thus, with the above configuration, the GT discharged thermal power is completely exploited. Moreover, as suggested in Table 13, even WTE outlet temperature has reached its minimum allowed value. T-Q diagram shown in Figure 36 and the numerical results reported in Table 13 refers to Q_{EXH} equal to 89.4 MW; since for Layout_7 it is not possible to identify an optimum plant match condition, the highest value of GT discharged thermal power previously found out (see Layout_5 optimum condition) has been taken as reference.

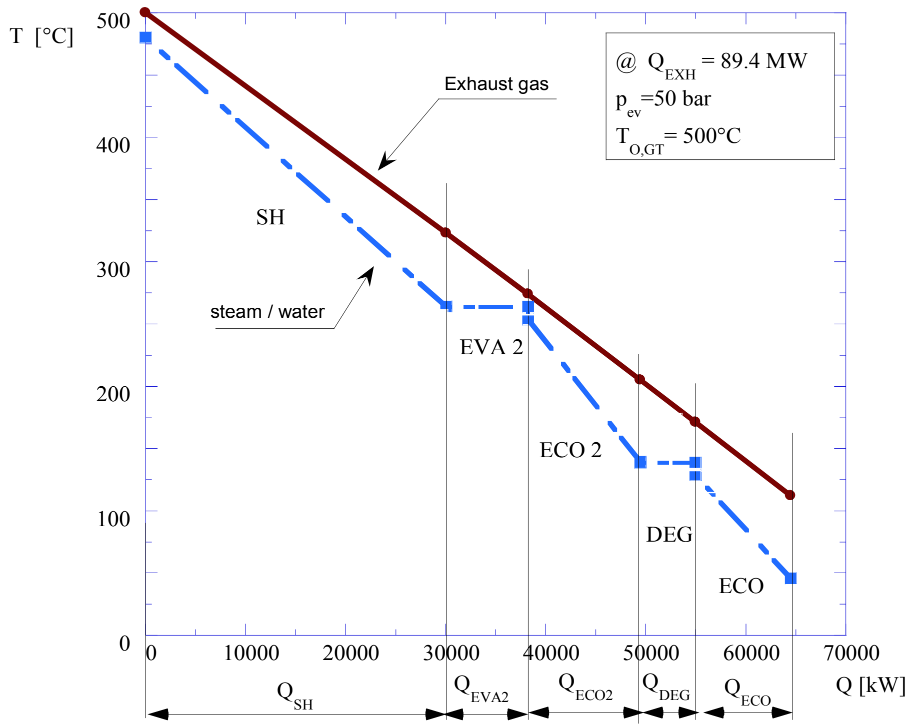

Figure 36 : T-Q diagram for HRSG section for Layout_7.

Table 13 : Layout_7 results

	Layout_7
F_W [MW]	100
p_{ev} [bar]	50
$T_{O,GT}$ [°C]	500
T_{SH} [°C]	480
m_{exh} [kg/s]	155
Q_{EXH} [MW]	89.41
Q_{EXH} /F_W [-]	0.89
$m_{s,max}$ [kg/s]	50.70
$T_{O,HRSG}$ [°C]	112
$T_{O,WTE}$ [°C]	160
P_{ST} [MW]	48.95

5.4. *Comparative results for WTE-TG one pressure level integrated layouts*

Results for all the proposed one pressure level WTE-GT integrated layouts are summarized in Figure 37, Figure 38 and Figure 39 showing respectively, the maximum amount of steam mass flow rate, the ST power output and the ratio between steam to waste mass flow rate, as function of GT discharged thermal power, in correspondence to optimum plant match conditions (points B) identified in previous paragraph.

As previously discussed, for the proposed configurations, Layout_3 reaches the minimum value of Q^{*}_{EXH}. While, layouts 1, 2, 2bis, 5, 6 and 6bis reaches the same maximum steam mass flow rate generation, Layout_4, Layout_4_tot_air, and Layout_7 lead to an increase in the maximum amount of steam that plant could produce. For what concern ST power capacity, Layout_7 gives the best performance, followed by Layout_5 and 4_tot_air.

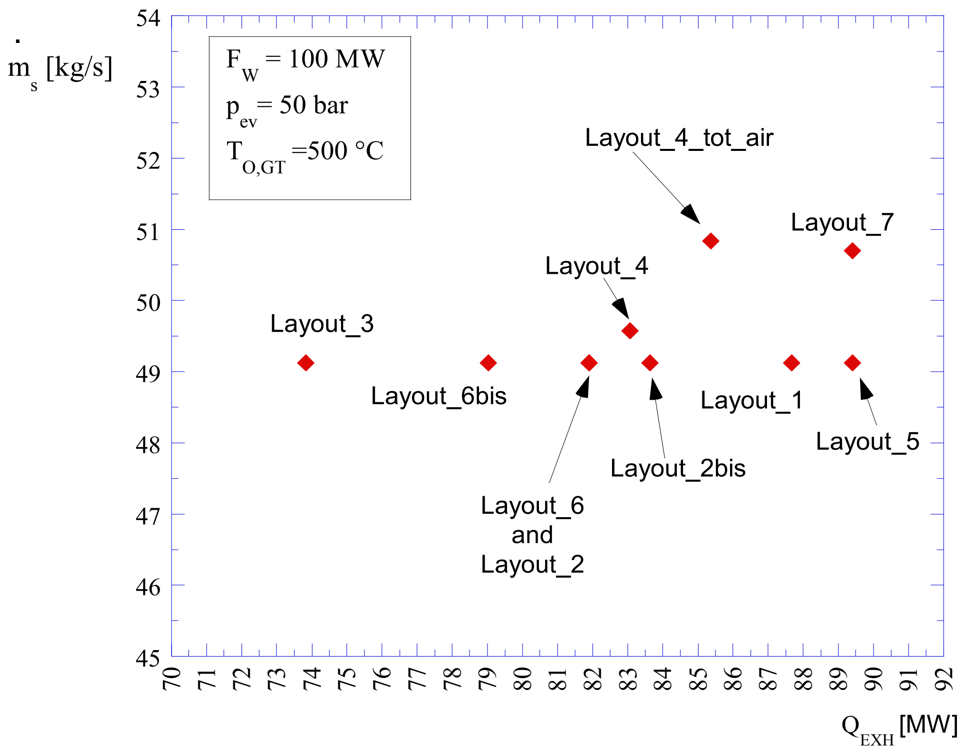

Figure 37 : Maximum steam mass flow rate (optimum condition, points B) as function of GT discharged thermal power.

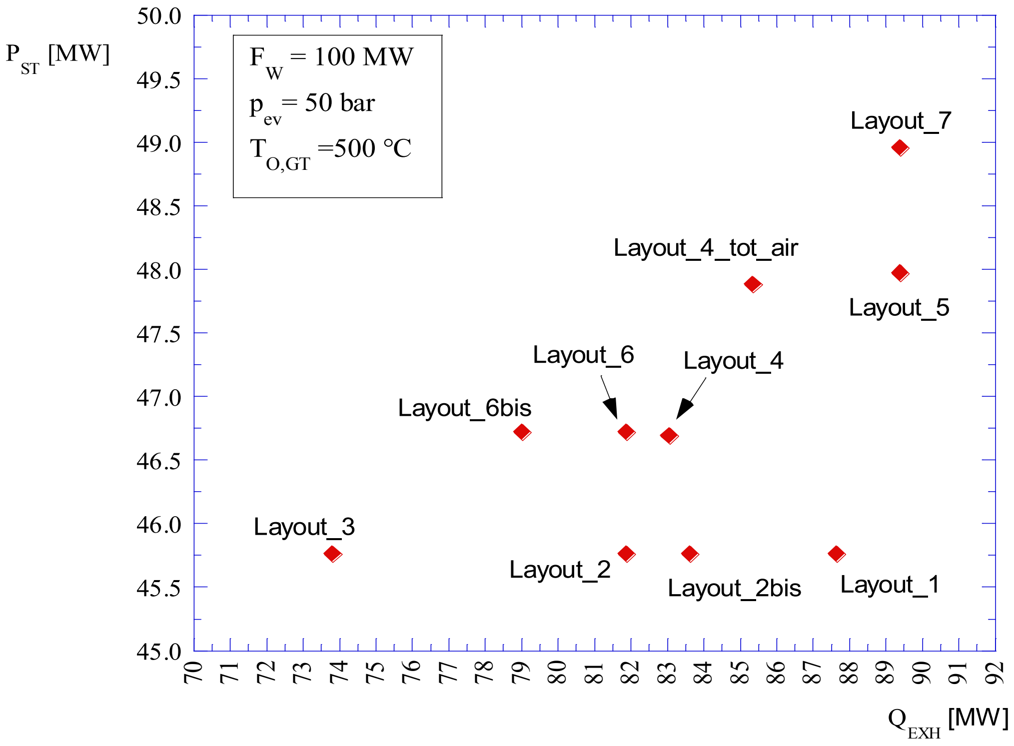

Figure 38 : Steam turbine power output as function of GT discharged thermal power.

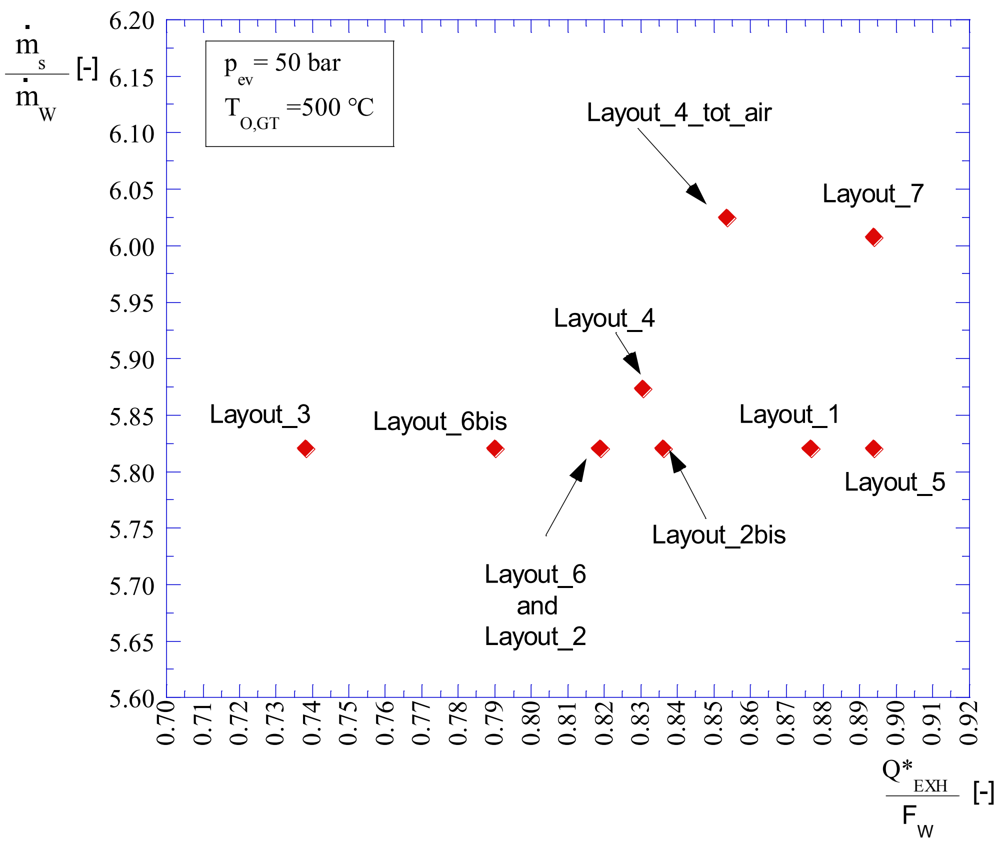

Figure 39 : steam to waste mass flow rate ratio as function of thermal input powers ratio.

The performed analysis shows that the steam mass flow rate produced by the integrated plant, for all the proposed layouts, is higher than that generated with a traditional WTE power plant fed with the same amount of waste. Figure 40 shows increment of steam mass flow rate and steam turbine power output for each investigated layout compared to a traditional WTE fed with the same thermal input power (F_W = 100 MW) and working at the same evaporative pressure and steam superheated temperature, respectively equal to 50 bar and 480 °C. The steam mass flow rate production of a stand alone WTE would be equal to about 32.09 kg/s while the ST capacity would be equal to 29.7 MW, thus, increments range, for the considered cases, from 53% up to 58 % for the steam mass flow and from 54% up to 65% for the ST capacity.

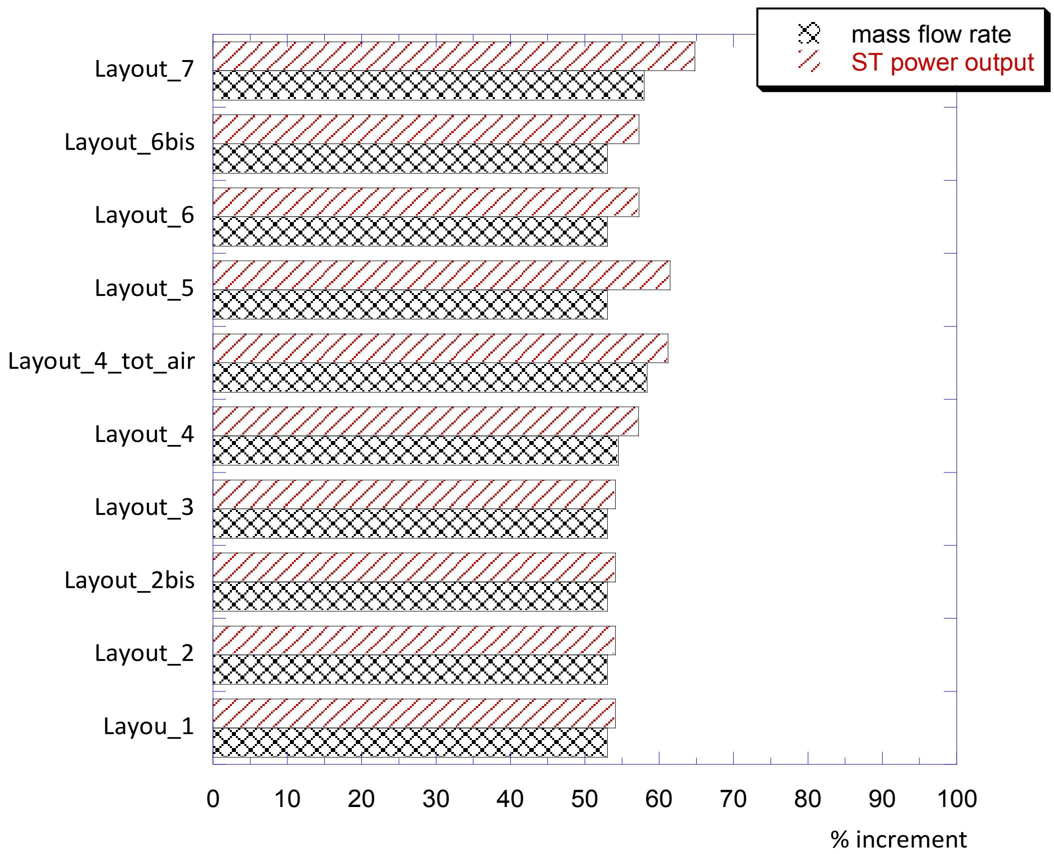

Figure 40 : increments on steam mass flow rate on the respect of a traditional WTE.

References

[1] *Gate Cycle^{TM}* by GE Energy. http://site.ge-energy.com/prod_serv/products/oc/en/opt_diagsw/gatecycle1.htm

[2] Petrov M. P., Martin A. R., Hunyadi L., 2002, "Hybrid Dual-Fuel Combined Cycles: General Performance Analysis", ASME International Joint Power Generation Conference, Phoenix AZ, USA, June 2002.

[3] Lozza G., Turbine a gas e cicli combinati, Progetto Leonardo, 2° Edizione.

6. WTE-GT steam/water side integration: thermodynamic analysis on two pressure levels

Pursuing the same idea of Chapter 5, here the integration between a WTE and a GT is carried out focusing on steam/water side for a two pressure level HRSG. Assumptions and hypothesis made for the thermodynamic study on one pressure level (see Table 3, Table 4 and Table 5 in Chapter 5) have been maintained also in the following analysis on two pressure levels integrated layouts.

The first step of the analysis identifies two reference configuration, namely EVA_LP and EVA_HP, where WTE boiler has the task of producing low and high pressure saturated steam, respectively. A comparative analysis on these reference layouts has been carried out, in order to select the most promising "starting" configuration. After that, different WTE-GT layouts have been proposed and investigated, in order to maximize plant performance and minimize flue gas discharged temperatures.

6.1. Comparing EVA_LP and EVA_HP integrated layouts

Starting from two reference cases, namely EVA_LP and EVA_HP, steam mass flow rate generation is investigated and analyzed. Both proposed configurations have two evaporative sections for low and high pressure saturated steam generation. In EVA_LP, WTE boiler provides low pressure saturated steam, while the generation of high pressure steam occurs inside the HRSG section.

On the contrary, in EVA_HP, low pressure saturated steam is generated inside the HRSG leaving the production of high pressure steam inside the WTE boiler. Schematic layouts for EVA_LP and EVA_HP are presented in Figure 1 and Figure 3, respectively along with the T-Q diagrams relative to the HRSG sections (Figure 2 and Figure 4).

Both layouts have, inside the HRSG section, two economizers, ECO LP and ECO HP, for low and high pressure water heating. Water out of deaerator is pressurized, reaching low evaporation pressure value, and then sent to the ECO LP, to be economized. After that, in EVA_LP, economized water is sent to the WTE boiler, where low pressure saturated steam is generated. Instead, in EVA_HP, low pressure saturated steam is generated inside the HRSG section. A fraction of the water mass flow rate, drawn out from low pressure drum, is pressurized reaching the high evaporation pressure value and further economized inside ECO HP into the HRSG section. High pressure saturated steam is generated inside HRSG or WTE boiler for EVA_LP and EVA_HP, respectively. High pressure saturated steam is superheated, in both layouts, in SH inside the HRSG.

Two steam turbines bodies are present: high pressure (HP ST) and low pressure (LP ST); between ST bodies a header is placed, combining steam out of HP ST and low pressure saturated steam.

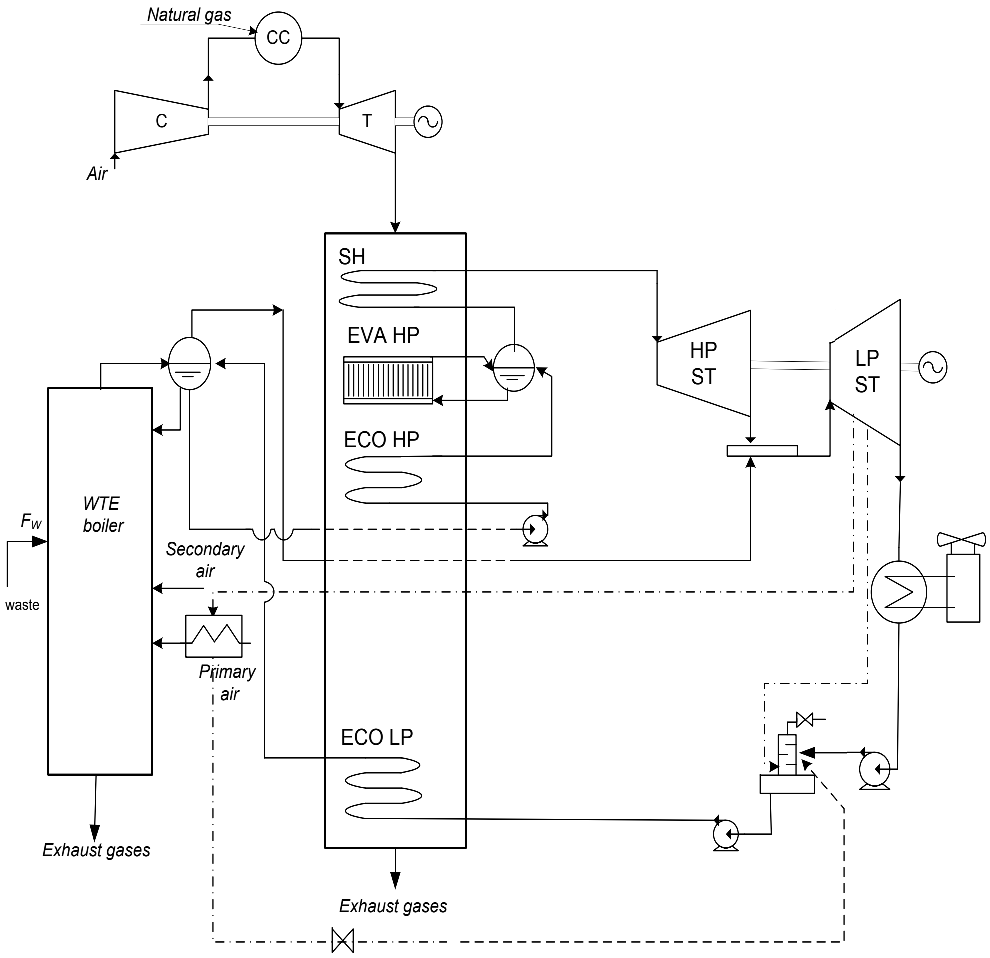

Figure 1 : Layout EVA_LP.

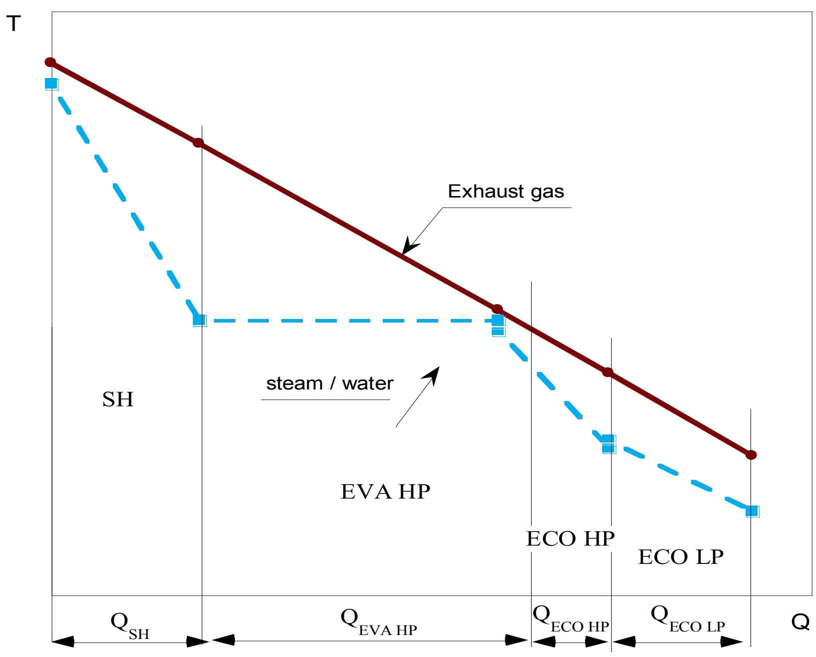

Figure 2 : schematic of T-Q diagram for Layout EVA_LP.

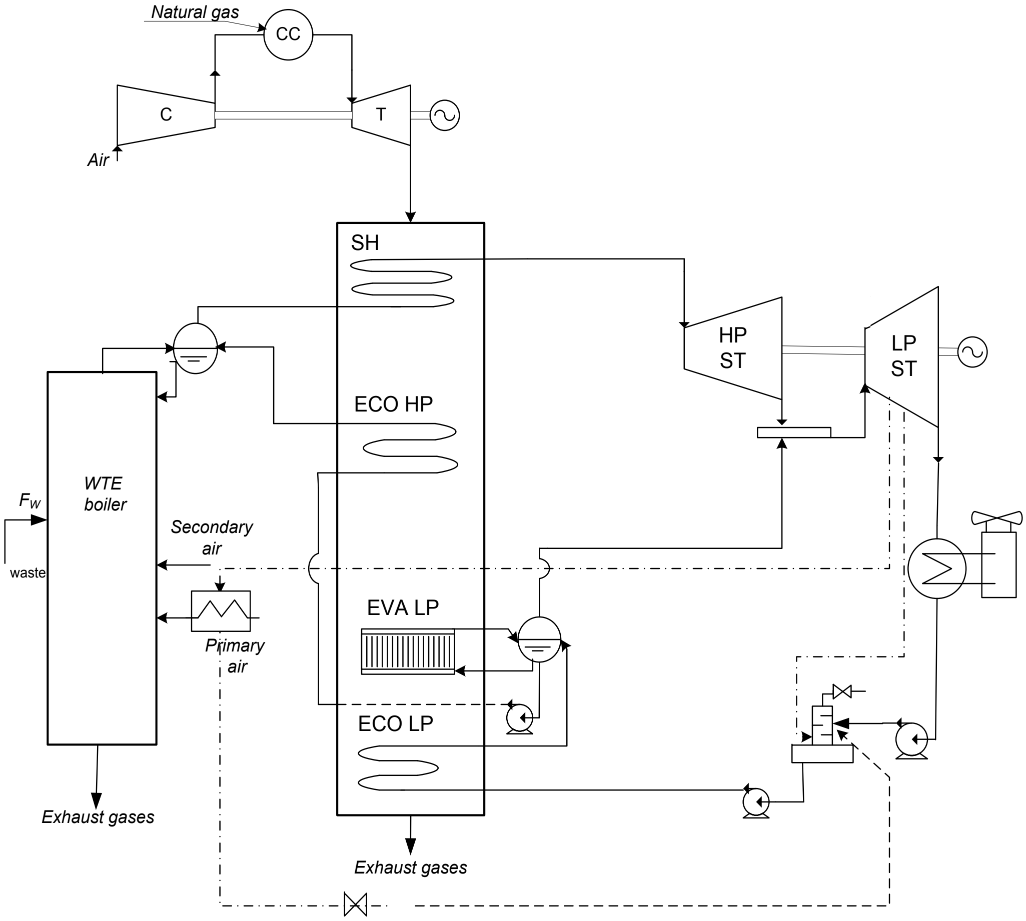

Figure 3 : Layout EVA_HP.

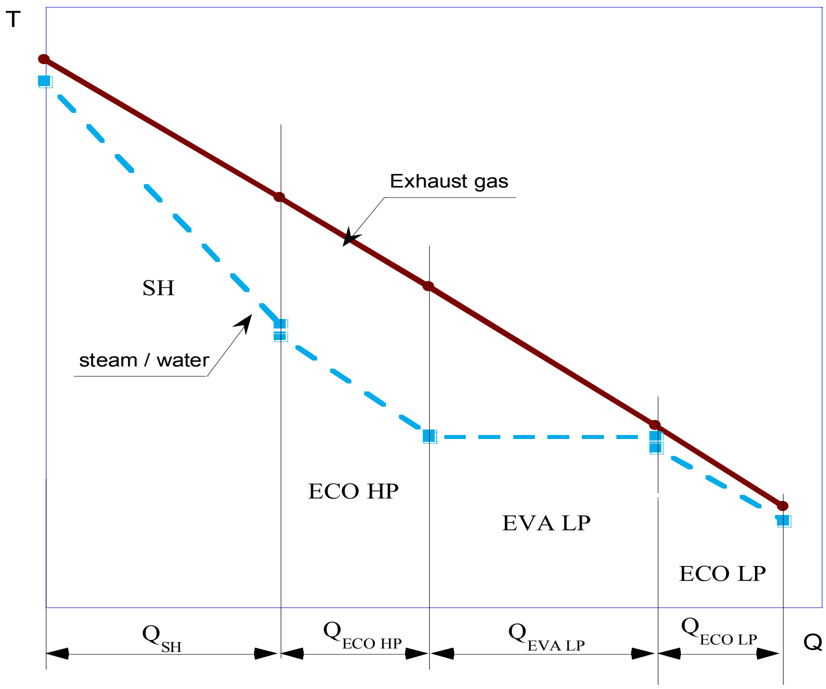

Figure 4 : T-Q diagram for Layout EVA_HP.

A thermodynamic analysis on low and high pressure saturated steam mass flow has been carried out, for both layouts, as function of GT discharged thermal power.

Figure 5 shows the qualitative behaviour of saturated steam mass flow and ST total power output, as function of Q_{EXH}, for layout EVA_LP. The following can be observed:

- There is a minimum value of GT discharged thermal power ($Q_{EXH,\min}$) which allows the production of high pressure saturated steam ($\dot{m}_{S,HP}$). For Q_{EXH} below $Q_{EXH,\min}$ a reduction of T_{SH} or an increase in pinch point temperature difference for EVA HP is required to have generation of high pressure steam. On the contrary, for Q_{EXH} beyond $Q_{EXH,\min}$ high pressure steam mass flow rate increases linearly increasing the GT discharged thermal power.
- Low pressure steam mass flow rate ($\dot{m}_{S,LP}$), generated inside the WTE boiler, increases almost linearly (D→B) reaching its maximum production in correspondence to point B (when the minimum sub-cooling temperature difference in ECO LP is reached) then a constant trend is kept (B→A) regardless the value of Q_{EXH}.
- The total ST power output increases almost linearly increasing Q_{EXH}. Two different trends can be observed: the first one, due to the increase of both low and high pressure steam mass flow rate and a second one (once low pressure steam reaches its maximum value, becoming constant) due to the increase only of high pressure steam.
- For the considered layout there is no optimal condition in terms of Q_{EXH}: ST power output and high pressure steam mass flow increase, increasing GT discharged thermal power. On the contrary, a minimum value of Q_{EXH} can be identified which guarantee high pressure saturated steam generation.

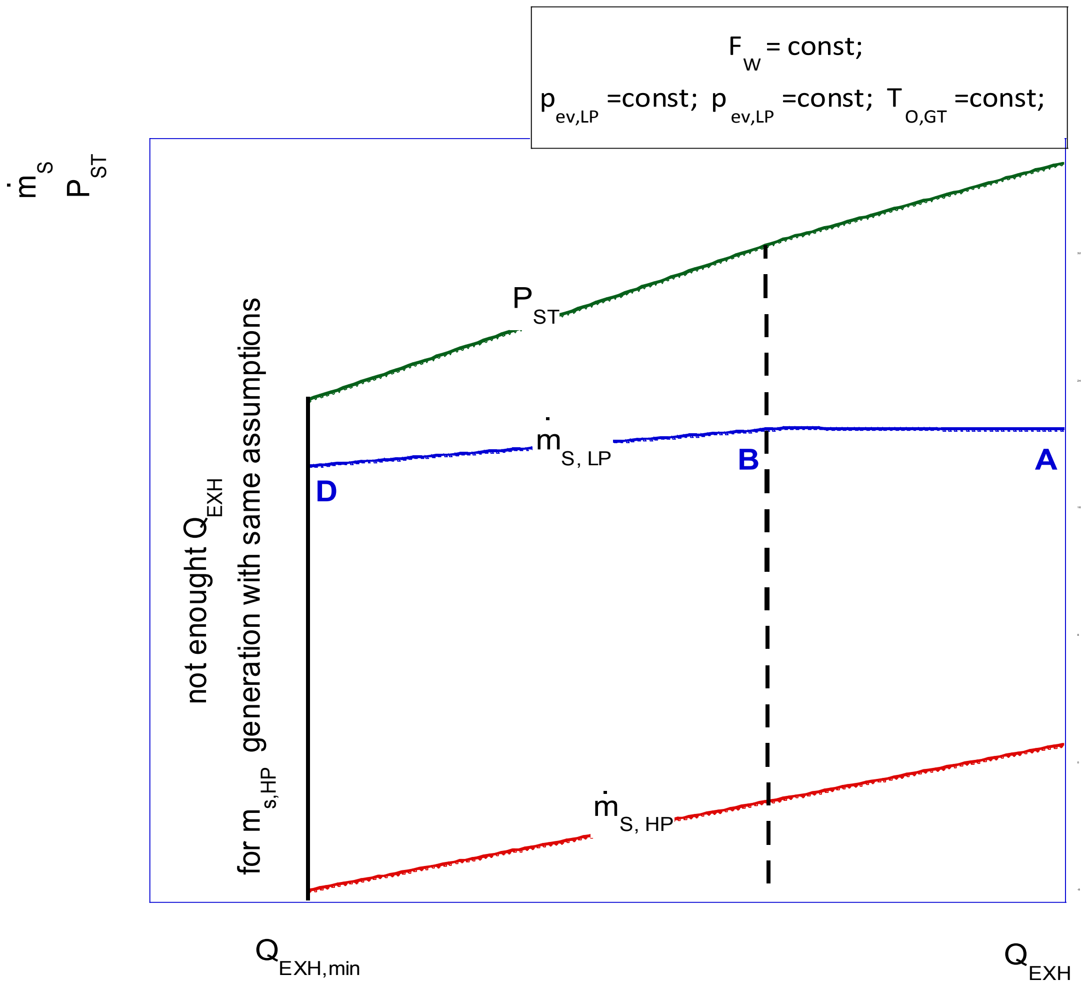

Figure 5 : qualitative behaviour of steam mass flows and ST power output for layout EVA_LP as function of GT discharged thermal power.

Instead, focusing on Figure 6, for layout EVA_HP, where qualitative behaviour of high and low pressure saturated steam and ST total power output are presented, the following observations can be pointed out:

– Also for this configuration there is a minimum value of GT discharged thermal power ($Q_{EXH,\ min}$) which allows the production of low pressure steam mass flow ($\dot{m}_{S,LP}$). For Q_{EXH} under $Q_{EXH,\min}$, to have generation of low pressure saturated steam, two possible strategies can be followed: (i) reducing high pressure steam maximum temperature, T_{SH}, thus decreasing steam cycle efficiency or (ii) reducing high pressure economizer effectiveness, thus decreasing high pressure saturated steam mass flow. On the contrary, for Q_{EXH} beyond $Q_{EXH,\min}$ low pressure steam mass flow rate increases linearly, increasing the GT discharged thermal power.

- For Q_{EXH} over $Q_{EXH,\min}$, high pressure steam mass flow rate ($\dot{m}_{S,HP}$), generated inside the WTE boiler, is constant and at its maximum value, regardless the value of Q_{EXH}.
- The total ST power output increases linearly due to the increase of low pressure steam mass flow rate.
- For the considered layout there is no optimal condition in terms of Q_{EXH}: ST power output and low pressure steam mass flow increase increasing GT discharged thermal power. On the contrary, a minimum value of Q_{EXH} can be identified.

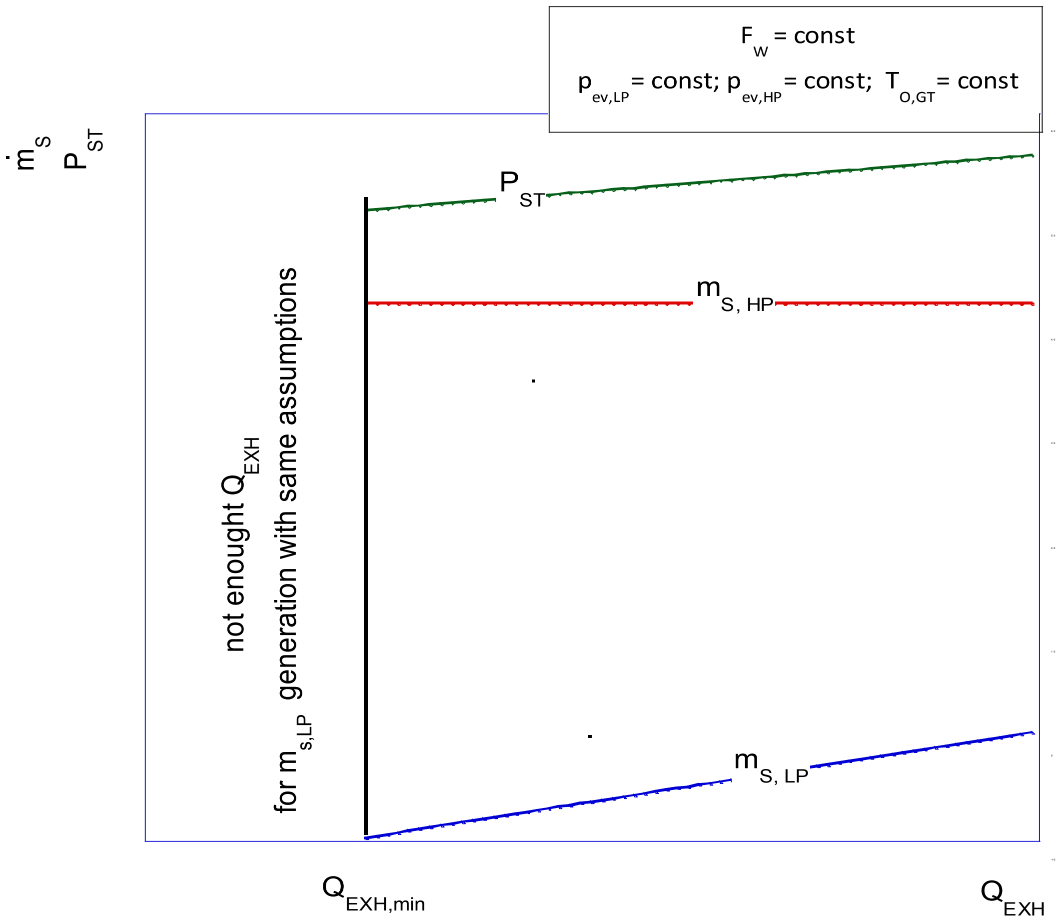

Figure 6 : qualitative behaviour of steam mass flow and ST power output for layout EVA_HP as function of GT discharged thermal power.

In conclusion, for both analyzed layouts, being constant the thermal power introduced with waste, evaporation pressures and superheated temperature, a minimum value of Q_{EXH} is identified. Below $Q_{EXH,\min}$, the thermal power available into HRSG section is not enough to guarantee generation of high or low pressure steam mass flow rate for EVA_LP and EVA-HP, respectively. A reduction of Q_{EXH} under $Q_{EXH,min}$ can be managed with the following strategies: (i) reducing the high pressure steam superheated temperature (or increasing the approach temperature difference) for both layouts; (ii) increasing pinch point difference inside evaporator for EVA_LP; (iii) reducing the evaporative pressure values for both layouts; (iv) reducing high pressure economizer effectiveness (or increasing the minimum sub cooling temperature difference) for EVA_HP.

Figure 7 shows the ST power output, for the investigated layouts, as function of Q_{EXH}; results have been obtained assuming a constant waste input power (F_W =100 MW) and the same steam cycle operative parameters (low and high evaporation pressure and superheated temperature). The power output for all one pressure level integrated layouts analyzed in Chapter 5 is also shown for comparison purpose.

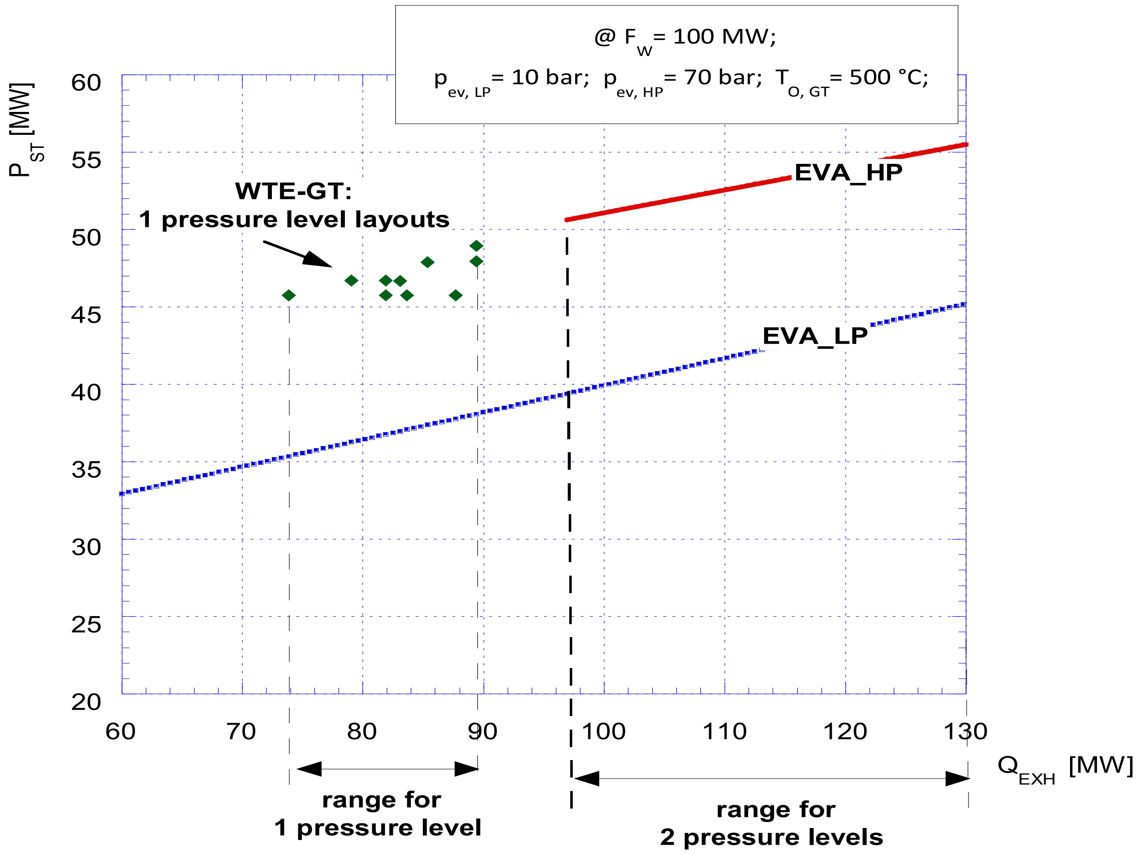

Figure 7 : Comparing ST power output for EVA_LP, EVA_HP as function of GT discharged thermal power.

From the figure it can be observed that:

- being equal the GT discharged thermal power, EVA_HP gives better performance in terms of ST power output on the respect of EVA_LP;
- EVA_LP shows an operative range, in terms of Q_{EXH}, wider than EVA_HP, but in this range its performance are lower than that achieved with one pressure level integrated layouts[1].

Concluding, for the operative range of interest in terms of Q_{EXH}, the starting configuration for WTE-GT two pressure level layouts is where high pressure saturated steam is generated inside the WTE boiler.

[1] Results for WTE-GT one pressure level integrated layouts refer to an evaporation pressure equal to 50 bar and to a steam superheated temperature equal to 480 °C (or an equivalent GT output temperature equal to 500 °C).

6.2. WTE-GT proposed layouts for two pressure levels HRSG

LAYOUT_9:

A first modified two pressure level integrated layout, namely Layot_9, is shown in Figure 8. With respect to EVA_HP layout, here low pressure economizer is shared between WTE (ECO LP1) and HRSG (ECO LP2) sections. Water, out of deaerator is divided into two streams: a fraction (f) goes into ECO LP2, the economizer section inside HRSG, while the remaining part (1-f) is sent to ECO LP1, inside WTE. Before entering low pressure drum, a mixer, combining both streams, is present. Water mass flow rate splitting is adjusted in order to maximize exploitation of both thermal power available into WTE and HRSG sections. Moreover, water mass flow rate distribution is adjusted in order to have similar mixer inlet temperatures (or economizer outlet temperatures).

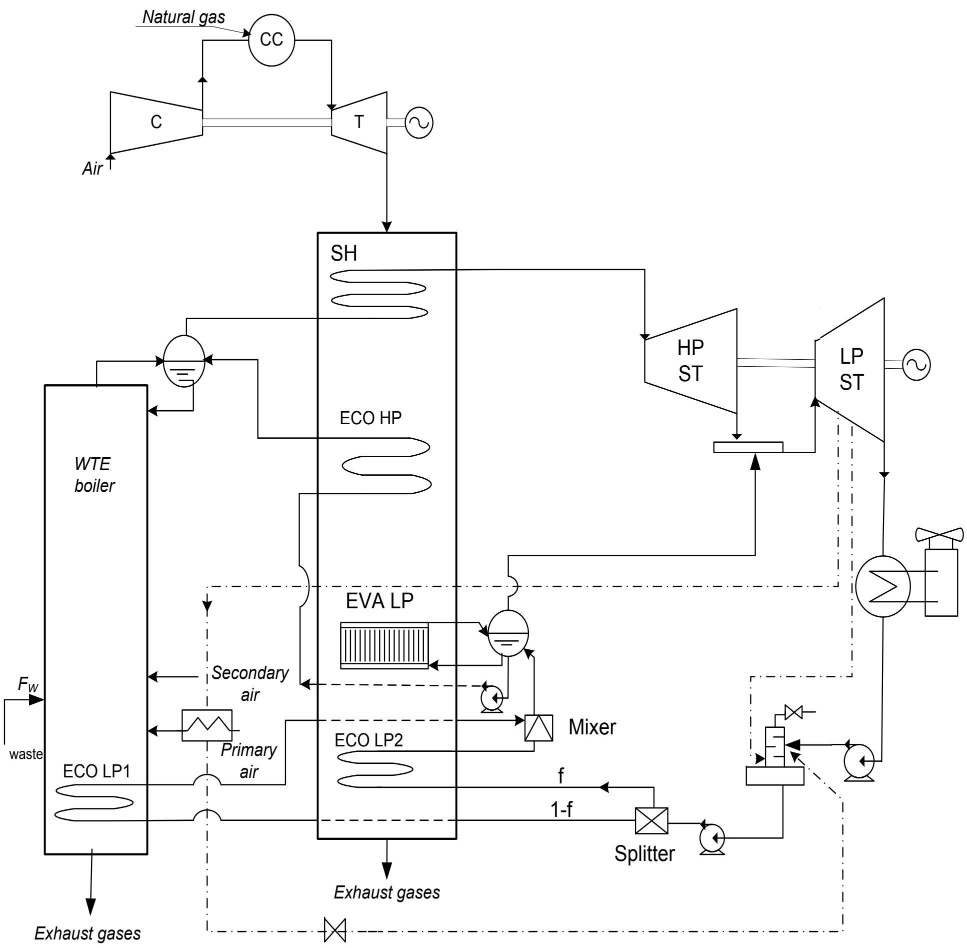

Figure 8 : schematic of Layout_9.

Optimal water splitting has been found in correspondence to f equal to 0.5 thus, water mass flow rate is equally divided between WTE and HRSG economisers.
No change occurs in steam mass flow rate (Figure 10) and steam turbine power output (Figure 9) for a fixed value of Q_{EXH}, in comparison with EVA_HP layout. Figure 10 clearly identifies a minimum value of GT discharged thermal power ($Q_{EXH,\min}$) which allows generation of low pressure steam mass flow keeping T_{SH} and ECO HP sub-cooling temperature difference at their maximum and minimum, respectively; below $Q_{EXH,\min}$, the available heat is not enough for low pressure evaporation heat exchanger.

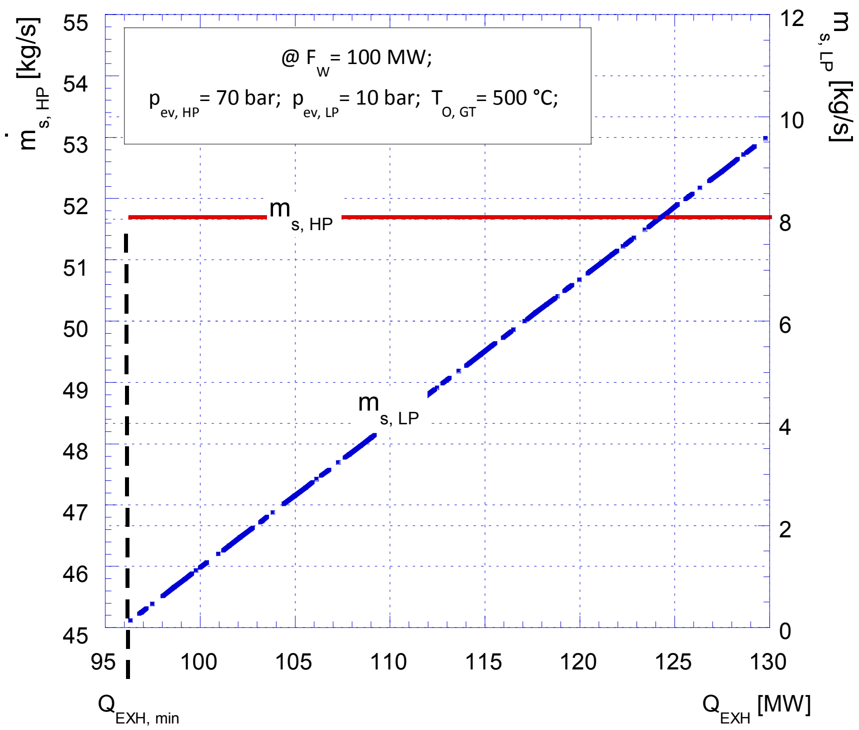

Figure 9 : low and high pressure steam mass flow rate, for Layout_9, as function of GT discharged thermal power.

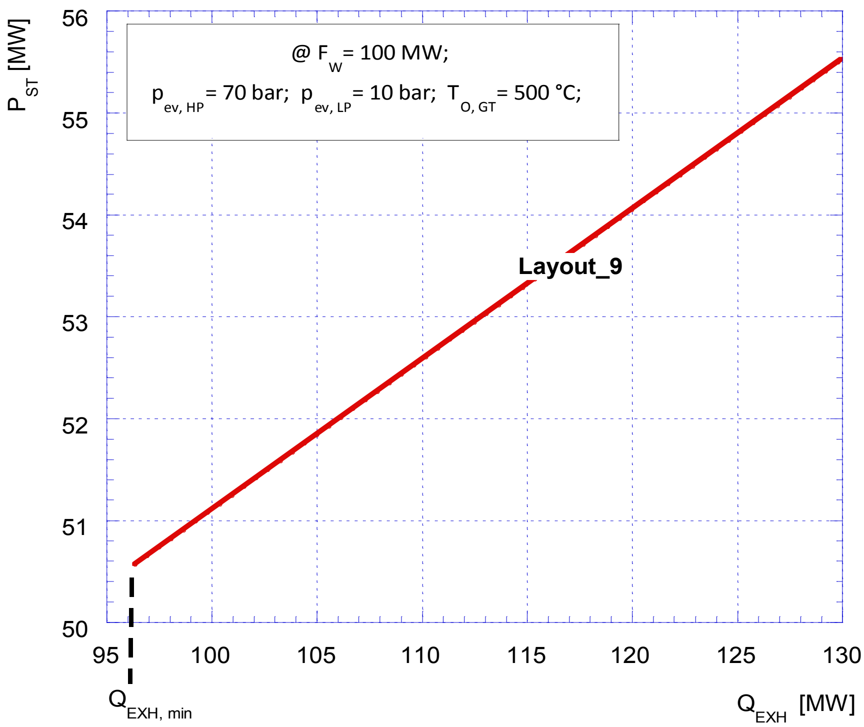

Figure 10 : ST power output for Layout_9 as function of GT discharged thermal power.

In Figure 11 the T-Q diagram relative to Layout_9 is shown. Diagram refers to a Q_{EXH} value equal to 100 MW, high and low evaporative pressure equal to 70 bar and 10 bar, respectively.

The proposed configuration allows to reduce, with reference to EVA_HP layout, WTE flue gas outlet temperature, while HRSG outlet temperature increases. Temperature of flue gas out of WTE boiler reaches 230 °C while, out of HRSG, exhaust gas temperature is equal to about 159 °C. As highlighted in HRSG T-Q diagram, most of the available GT discharged heat is used for SH and ECO HP, penalizing low pressure evaporator.

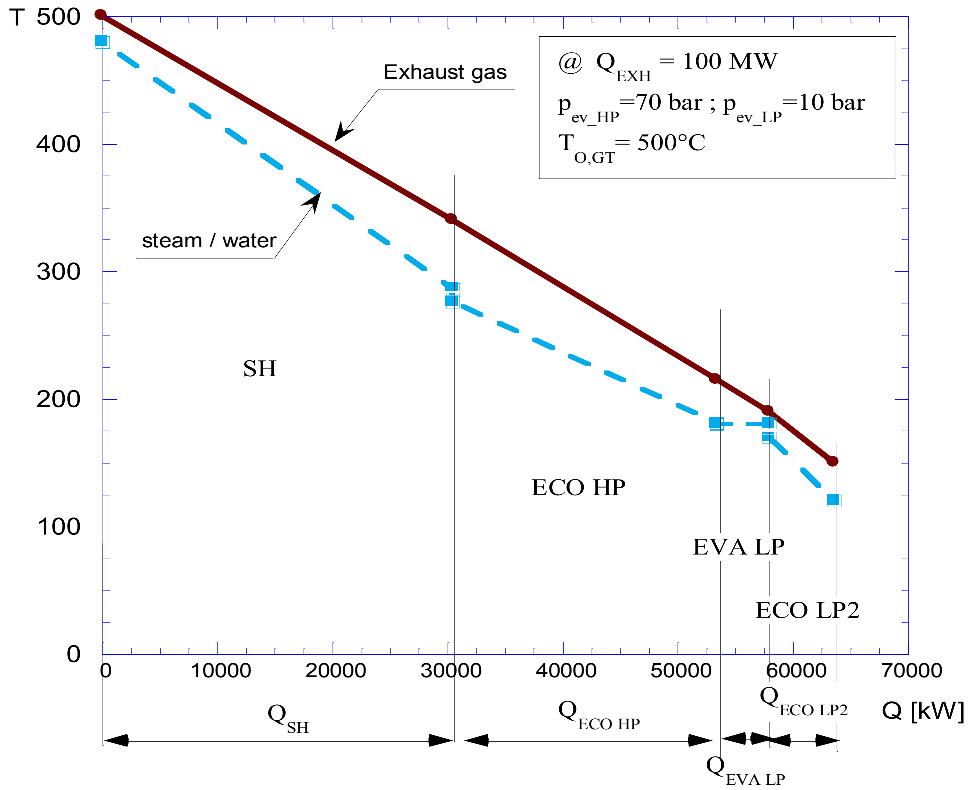

Figure 11 : T-Q diagram for Layout_9.

LAYOUT_10

In Layout_10, shown in Figure 12, the possibility to share high pressure water economization is investigated. Low pressure water economization is completely performed into the HRSG section (ECO LP), while high pressure economization is divided between WTE (ECO HP1) and HRSG (ECO HP2) with a configuration of heat exchangers in series.

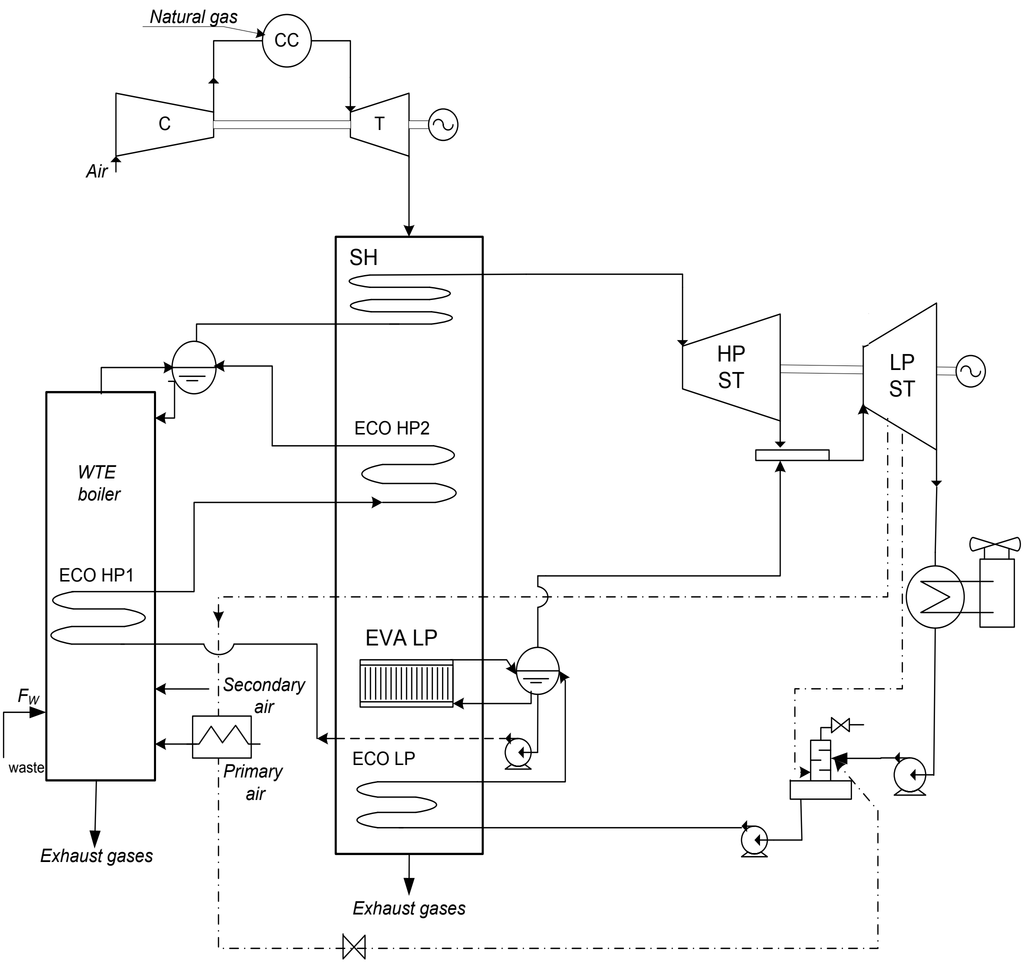

Figure 12 : schematic of Layout_10.

Figure 13 shows HP and LP steam mass flow rate as function of GT discharged heat; as highlighted in figure, in comparison with Layout_9, a considerable decrease in $Q_{EXH,\min}$ has been found. This can be explained considering that water temperature at ECO HP2 inlet is higher than previous layout thus, less heat is required for ECO HP2, making it available for low pressure evaporator heat exchanger. High pressure saturated steam shows two different trends, depending on Q_{EXH} : a linear increasing trend and a subsequent constant trend. For Q_{EXH} lower than 87.7 MW, water out of ECO HP2 can not reach the minimum sub cooling temperature difference (as economizer effectiveness has been limited) thus, the amount of saturated steam generated is lower than the maximum amount that the WTE boiler could produce. Once reached the minimum sub cooling temperature difference, $\dot{m}_{s,HP}$ turns out to be constant, equal to its maximum value. Decreasing Q_{EXH} below $Q_{EXH,\min}$ there is not enough thermal power to superheat all the amount of high pressure steam, keeping T_{SH} at its maximum value.

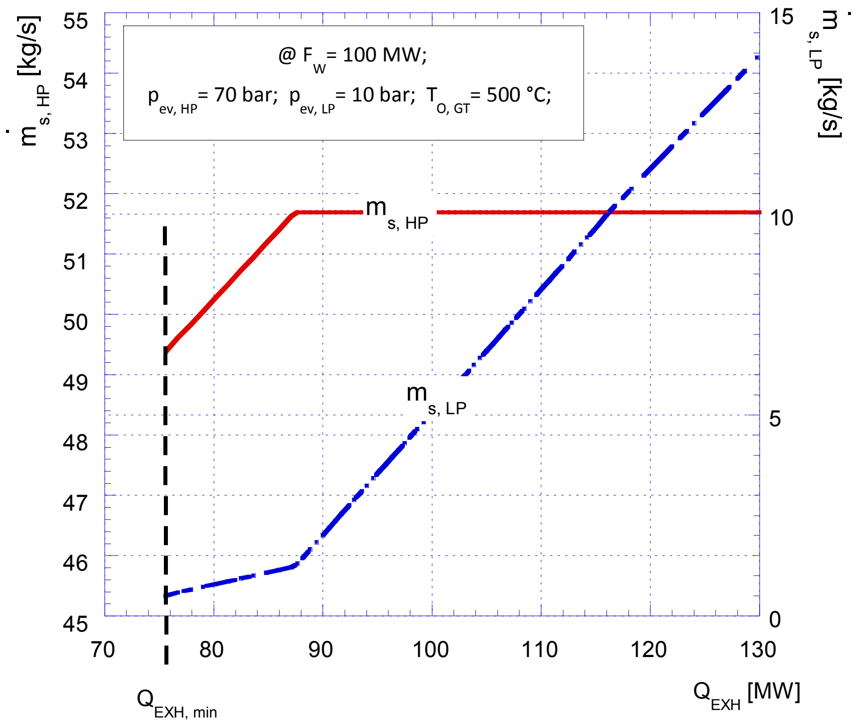

Figure 13 : low and high pressure steam mass flow rate, for Layout_10, as function of GT discharged thermal power.

Steam turbine power output behaviour is presented in Figure 14 as function of Q_{EXH}. For a given value of Q_{EXH}, the increase in low pressure steam mass flow leads to an increase in ST power output in case of Layout_10, in comparison with Layout_9.

In Figure 15 the T-Q diagram relative to Layout_10 is shown. The proposed configuration allows a considerable reduction in WTE exhaust gas temperature which reaches about 193 °C. Also a reduction in HRSG outlet temperature is achieved (equal to 127 °C) decreasing the distance between exhaust gas line and

ECO LP. In this configuration, in comparison with Layout_9, being equal available heat for low pressure evaporator increases.

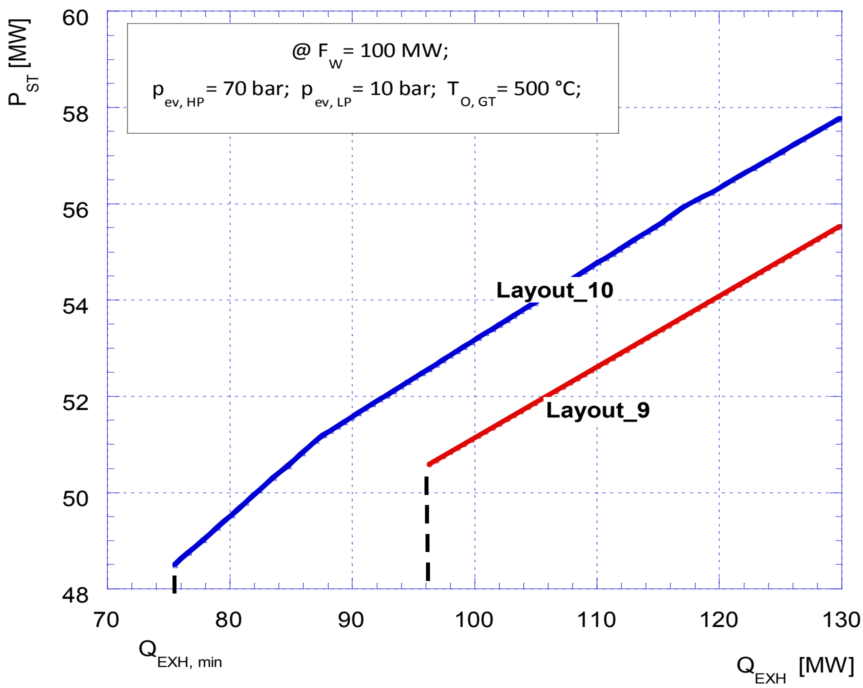

Figure 14 : ST power output for Layout_10 and Layout_9 as function of GT discharged thermal power.

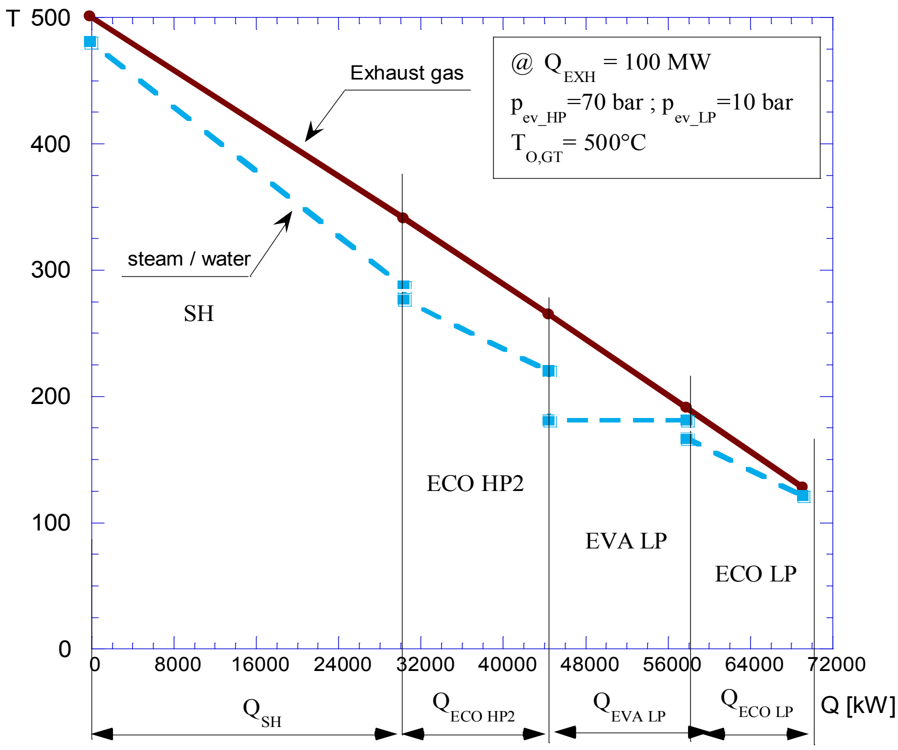

Figure 15 : T-Q diagram for Layout_10.

LAYOUT_11:

The possibility to share both low and high pressure water economization is investigated in Layout_11, shown in Figure 16. As for Layout_9, low pressure water economization is performed in parallel between WTE (ECO LP1) and HRSG (ECO LP2) sections: water out of deaerator is split into two streams and mixed before entering low pressure drum. Instead, as for Layout_10, high pressure water economization is shared between WTE (ECO HP1) and HRSG (ECO HP2) through a series configuration.

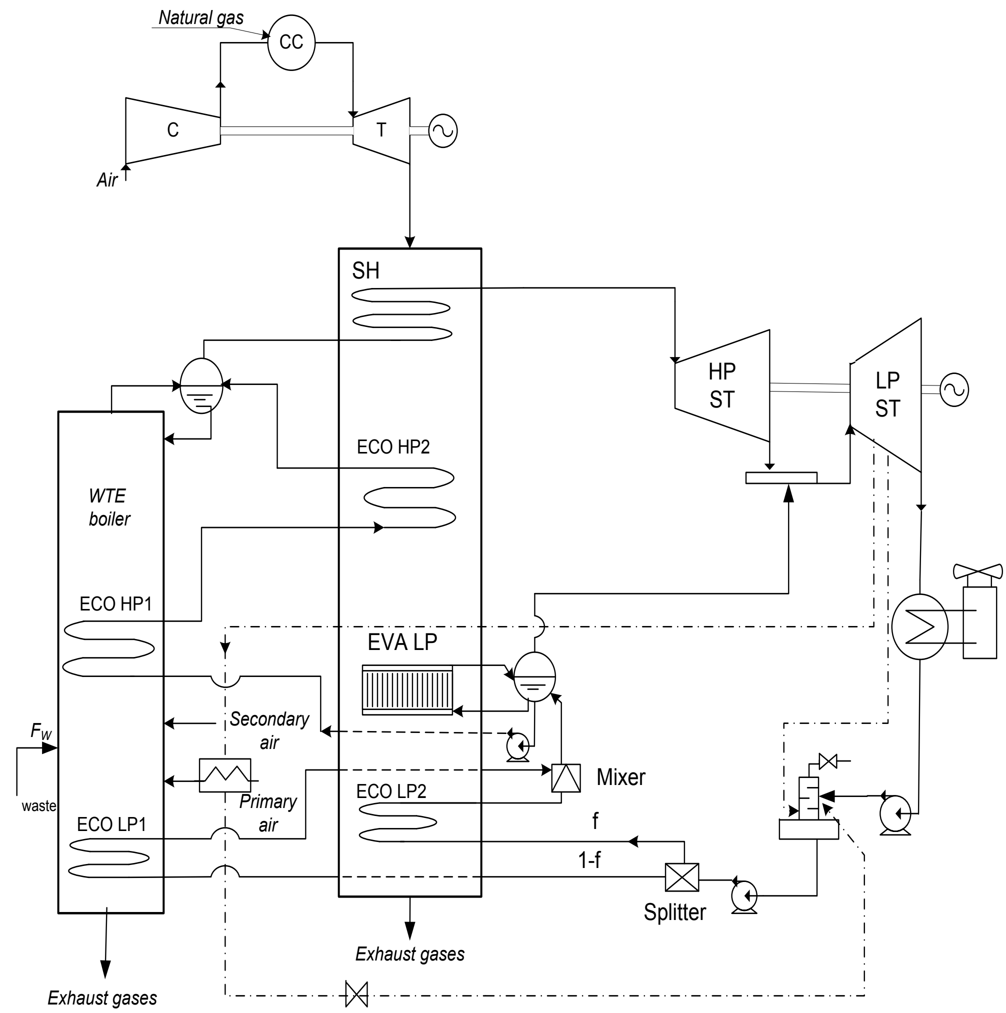

Figure 16 : schematic of Layout_11.

Figure 17 and Figure 18 show steam mass flow rate and ST power output for Layout_11. High pressure steam behaviour can be explained with the same considerations made for Layout_10. Comparing ST powers output of Layout_10 and

Layout_11, small differences can be highlighted; this is due to similar layouts behaviour for what concerns low pressure saturated steam generation.

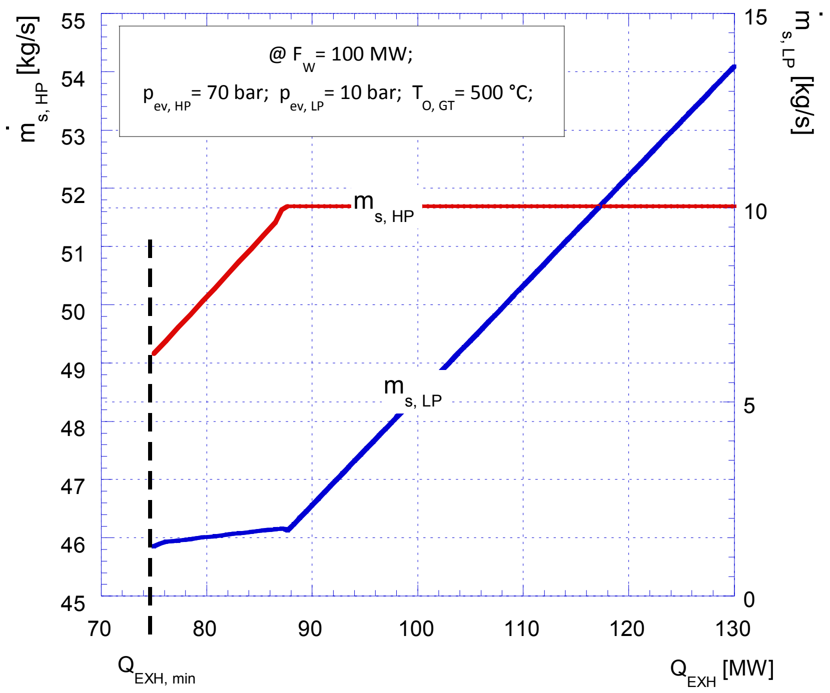

Figure 17 : low and high pressure steam mass flow rate, for Layout_11, as function of GT discharged thermal power.

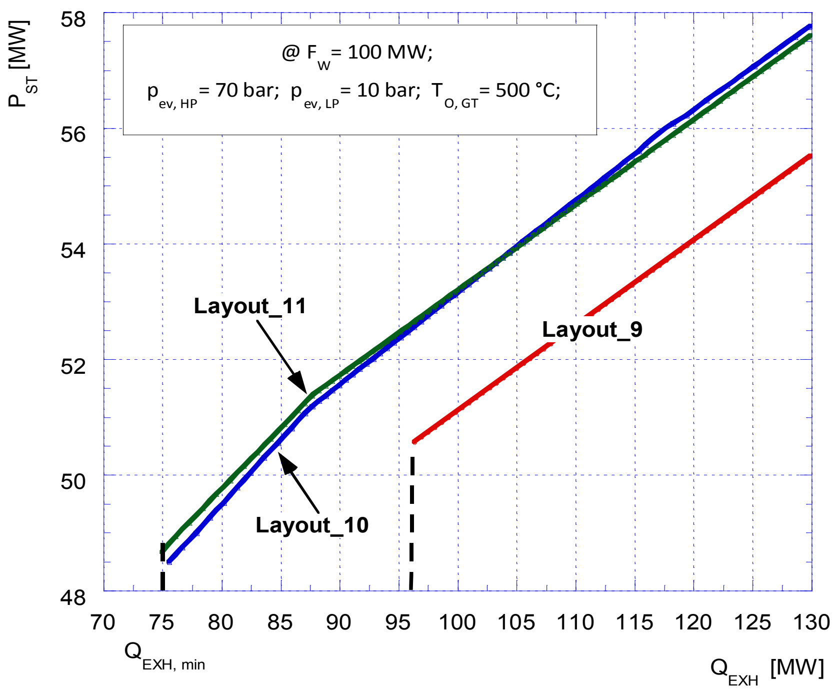

Figure 18 : ST power output for Layout_11, Layout_10 and Layout_9 as function of GT discharged thermal power.

In Figure 19 the T-Q diagram relative to Layout_11 is shown. Proposed configuration allows to reduce necessary heat for high pressure economizer, making it available for EVA LP: in comparison with Layout_9, being equal Q_{EXH} and evaporative pressure values, an increase in low pressure steam mass flow rate is reached in this configuration. WTE and HRSG exhaust gas temperature are equal to 171 °C and 136 °C, respectively.

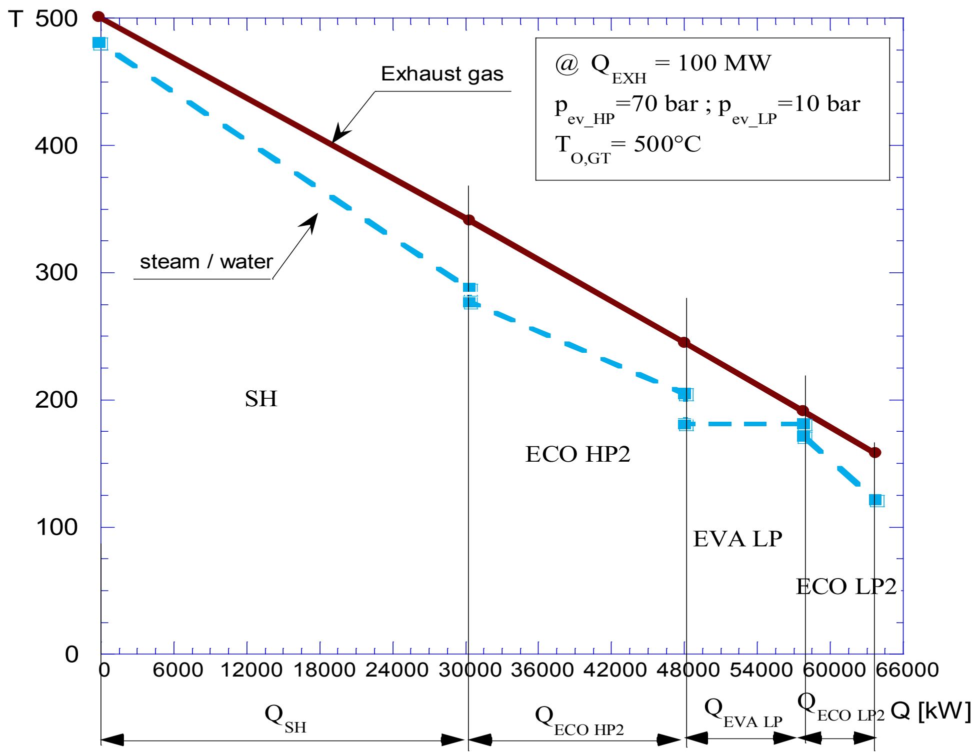

Figure 19 : T-Q diagram for Layout_11.

LAYOUT_12:

Differently to Layout_11, in Layout_12, shown in Figure 20, both high and low pressure economization is shared, through a series, between WTE and HRSG: water out of deaerator is firstly sent to ECP LP1 inside WTE boiler to complete the process in ECO LP2, inside HRSG. In Figure 17 and Figure 18 steam mass flow and ST power output are shown as function of GT discharged thermal power. It is evident, from Figure 18, that performance of Layout_12 are almost coincident with performance of Layout_11.

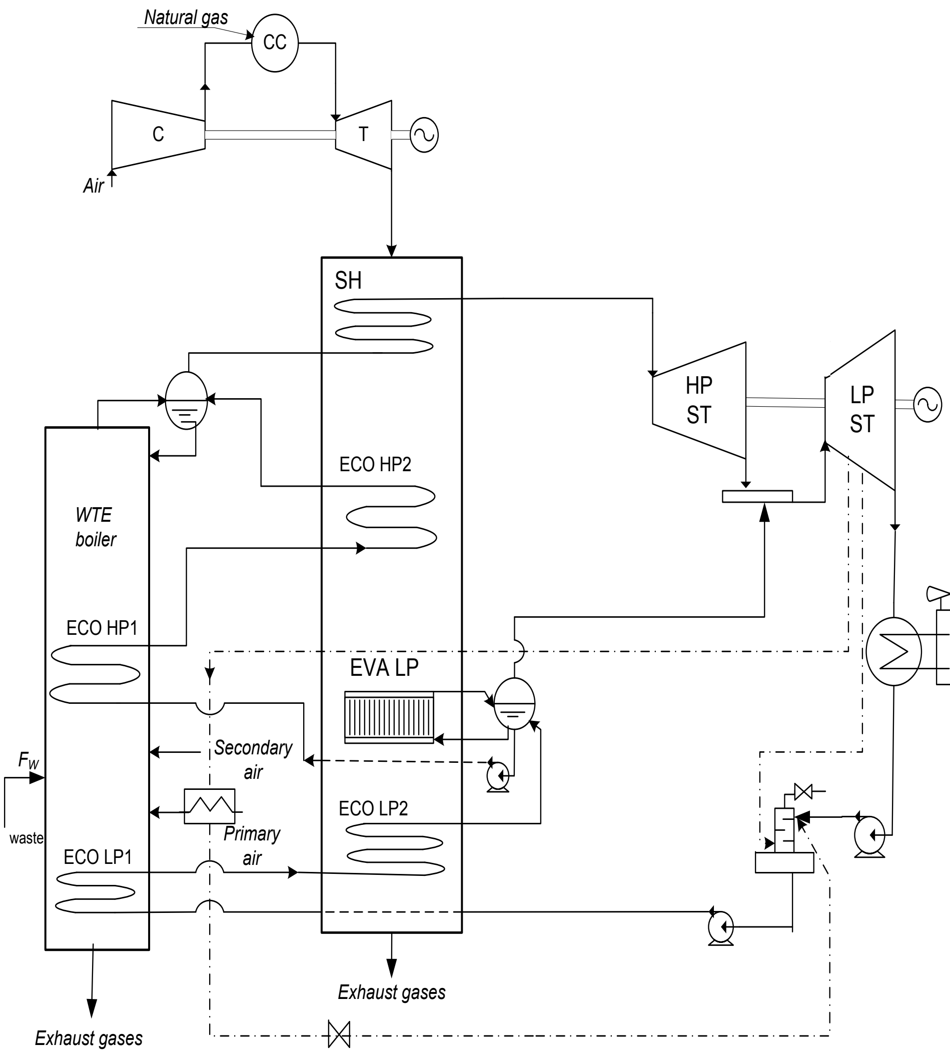

Figure 20 : schematic of Layout_12.

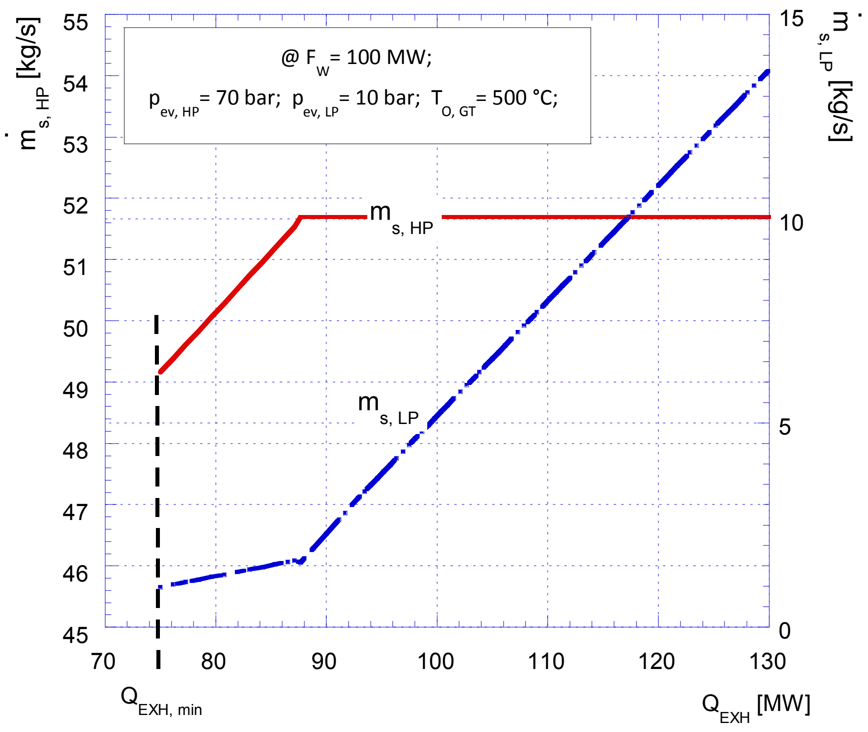

Figure 21 : low and high pressure steam mass flow rate, for Layout_12, as function of GT discharged thermal power.

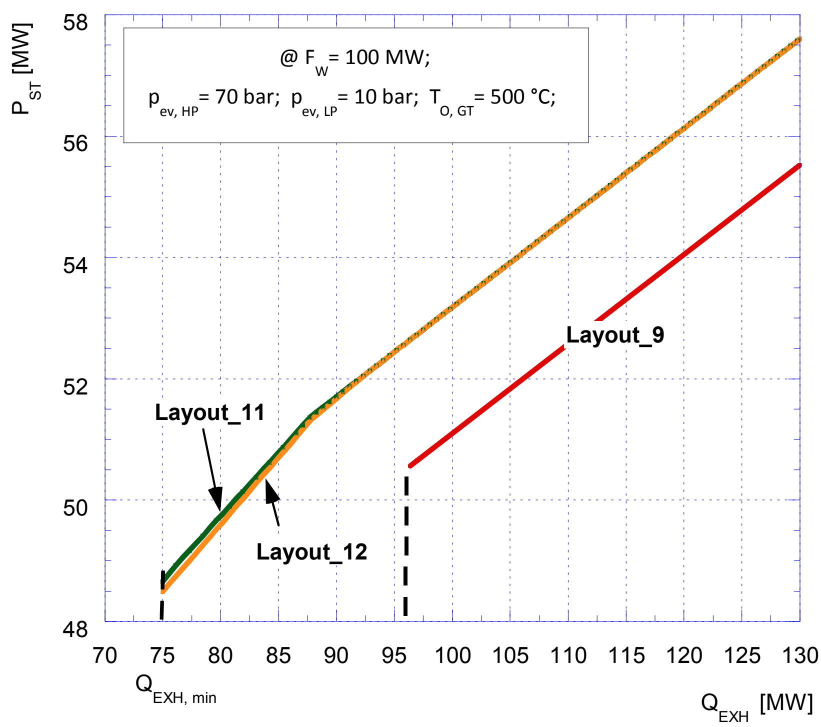

Figure 22 : ST power output for Layout_12 as function of GT discharged thermal power.

In Figure 23 the T-Q diagram relative to Layout_12 is shown. With reference to previous layouts, the investigated configuration allows WTE outlet temperature to reach its minimum value (160 °C), making all the available heat from WTE completely exploited, while HRSG discharged temperature, equal to about 141 °C, is still higher than the minimum allowed.

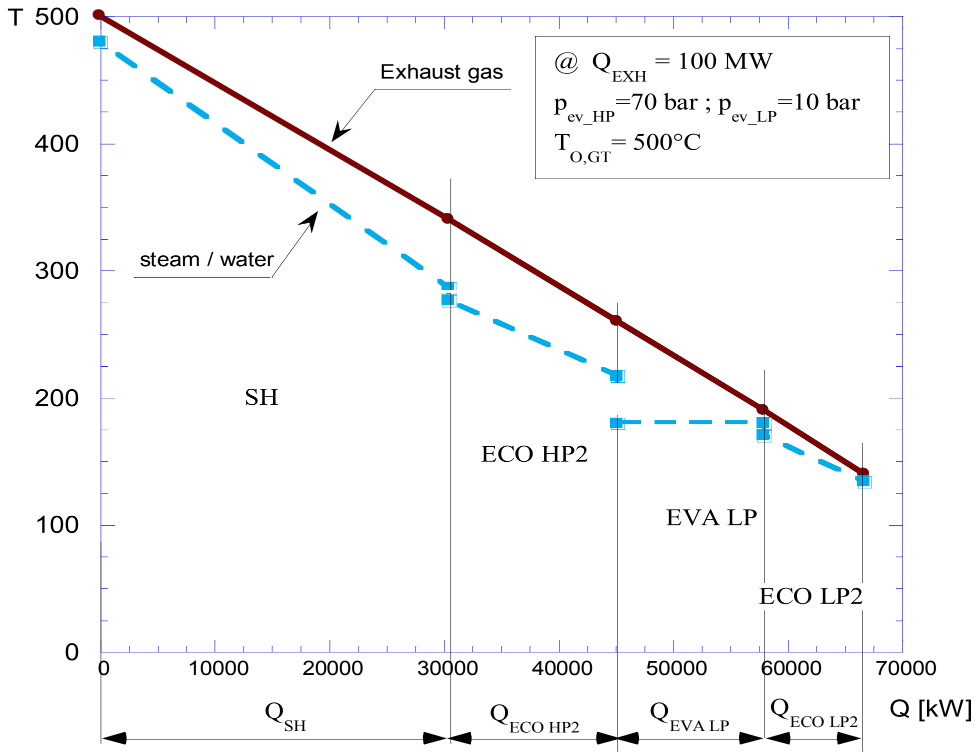

Figure 23 : T-Q diagram for Layout_12.

Changing low pressure economization sequence (first in the HRSG section, then in WTE) will result in no change in plant performance (same ST power output and saturated steam productions). On the contrary, a considerable decrease in HRSG exhaust gas (equal to about 127 °C) is reached; WTE outlet temperature would be close to its minimum, being equal to 186 °C.

Starting from Layout_12, the possibility to introduce a low pressure superheater has been investigated; Low pressure steam superheated temperature has been selected equal to HP ST outlet temperature, thus superheated steam entering the LP ST have an inlet temperature equal to 244 °C.

Being equal GT discharged heat, a decrease in low pressure saturated steam mass flow rate, on the respect of Layout_12 can be observed (Figure 24). Increasing Q_{EXH}, the decrease in low pressure steam mass flow rate grows; this can be explained

considering that, by increasing the total duty required for SH LP, less heat is available for EVA LP.
On the contrary, slight benefits in terms of ST power output have been found. In fact, limited gain in ST power output, due to low pressure superheated temperature, turns out to be hampered by mass flow decrease. As a conclusion, the introduction of low pressure superheater does not lead to additional and relevant benefits in terms of both, low pressure saturated steam mass flow rate and steam turbine power output.

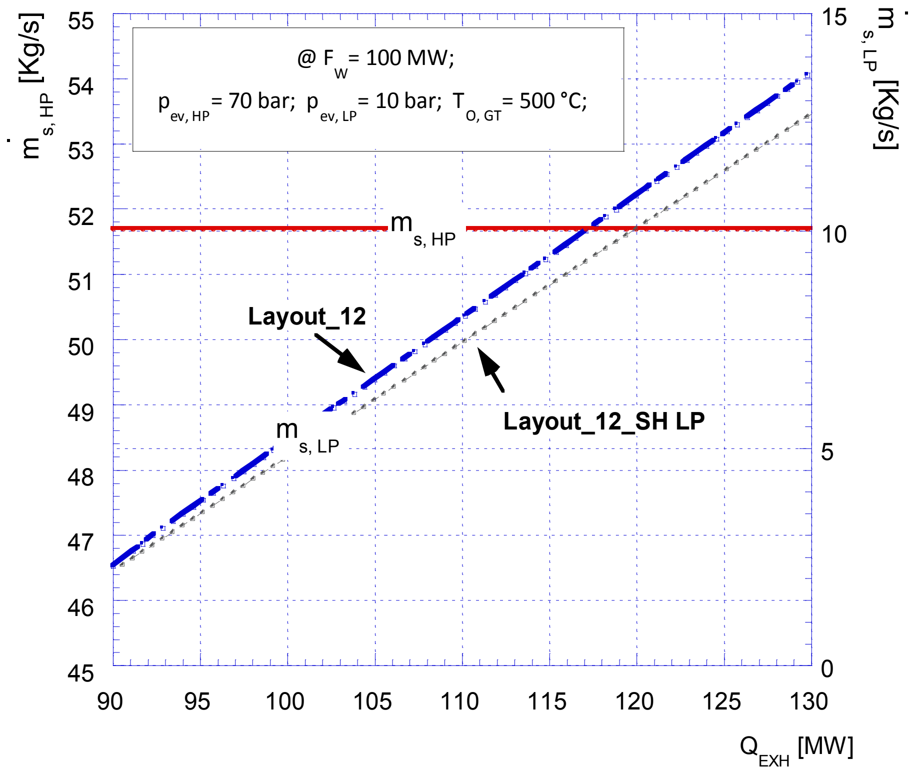

Figure 24 : low and high pressure steam mass flow rate, for Layout_12_SH LP and Layout_12, as function of GT discharged thermal power.

LAYOUT_13:

In Layout_13 (Figure 25), both high and low pressure economization is shared between WTE and HRSG. With reference to previous layouts, in the proposed configuration, an integral deaerator is introduced inside HRSG thus, eliminating the ST bleed. Water out of condenser is divided into two streams: a fraction (1-h) sent to a heat exchanger for water preheating fed by a ST bleed, the remaining (h) is heated up inside HRSG section in ECO PRE DEA.

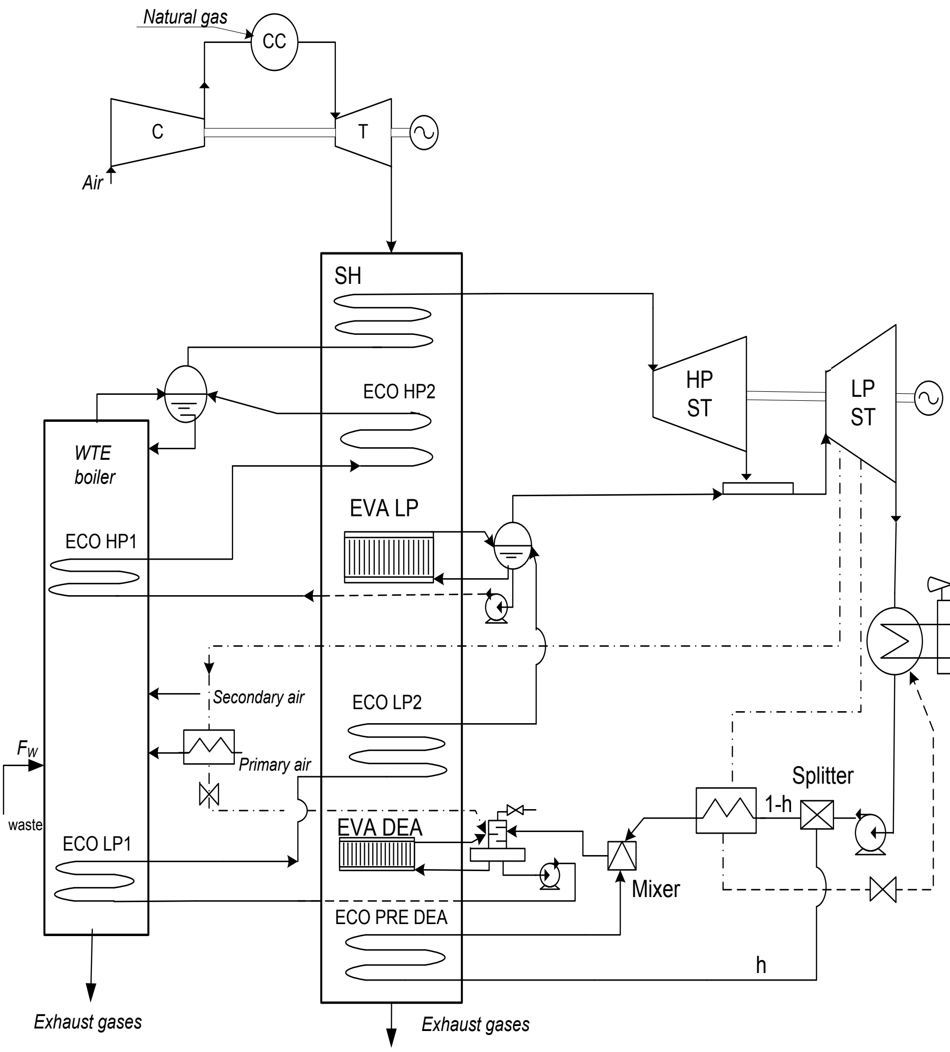

Figure 25 : schematic of Layout_13.

Figure 26 and Figure 27 show steam mass flow rates and ST power output as function of GT discharged thermal power. An interesting increase of P_{ST} for Layout_13 in comparison with Layout_10 and Layout_9 can be observed, in Figure 27, for fixed Q_{EXH} .

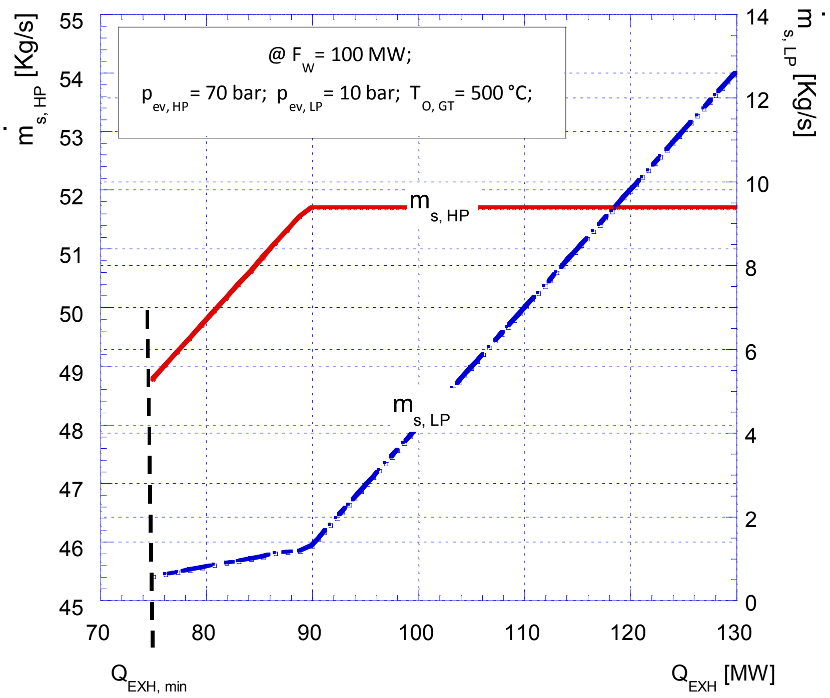

Figure 26 : low and high pressure steam mass flow rate, for Layout_13, as function of GT discharged thermal power.

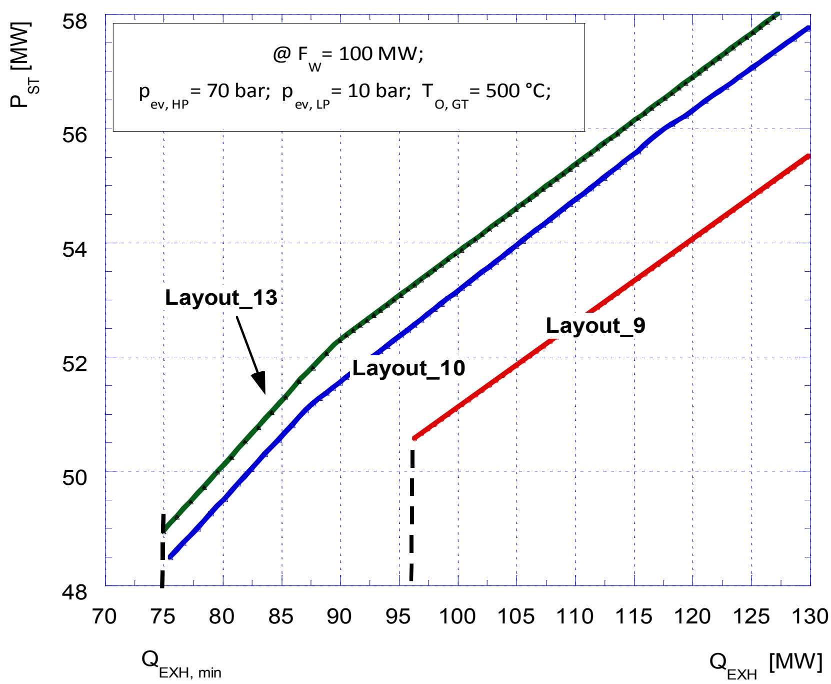

Figure 27 : ST power output for Layout_13 as function of GT discharged thermal power.

The T-Q diagram relative to Layout_13 is shown in Figure 28. Optimum condition for water mass flow rate division has been found when h (the fraction of the total water mass flow going into ECO PRE DEA) is equal to 0.2. The proposed configuration allows to minimize both WTE and HRSG discharged heat, minimizing outlet temperature values equal to 160 °C and 114 °C, respectively for WTE and HRSG sections.

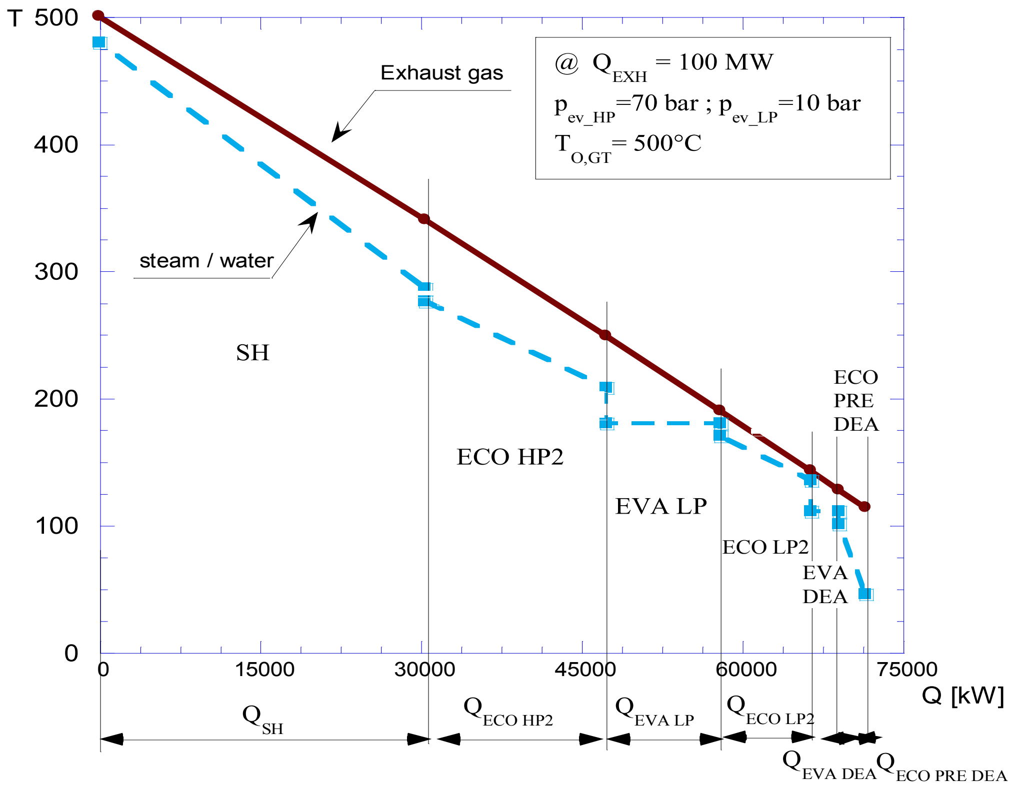

Figure 28 : T-Q diagram for Layout_13.

LAYOUT_14:

In the last proposed layout, namely Layout_14 in Figure 29, the possibility to reheat steam out of HP ST has been investigated; therefore, Layout_14 differs from the previous one for the presence of RH heat exchanger into the recovery boiler. Steam out of high pressure ST and low pressure saturated steam are combined through a header, mixed and then heated up to a temperature equal to 360 °C. Reheating temperature has been chosen, in lines with general assumptions, setting the approach temperature difference (see Chapter 5, Table 5). Water out of condenser has been splitted, depending on Q_{EXH}, in order to minimize HRSG outlet temperature and ST bleed mass flow rate: ratio between water sent to ECO PRE DEG to the total mass flow rate (h) is within the range 0.1 – 0.4, depending on Q_{EXH} value.

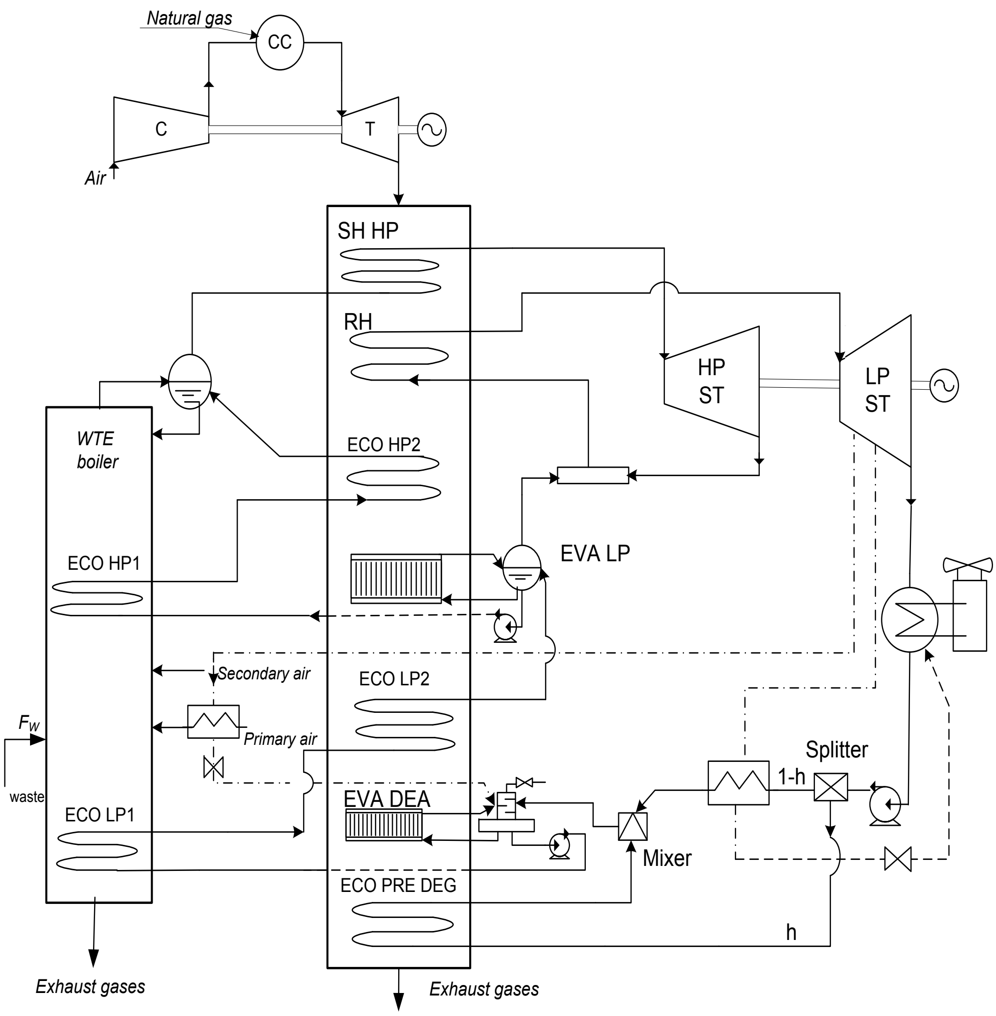

Figure 29 : schematic of Layout_14

Low and high pressure steam mass flow rates and ST power output are shown in Figure 30 and Figure 31, as function of GT discharged thermal power. For Q_{EXH} lower than 79 MW, both low and high pressure economisers can not reach minimum sub cooling temperature difference, thus effectiveness has been selected, as input, for both heat exchangers. For Q_{EXH} higher than 79 MW minimum sub cooling condition is reached in ECO LP2 while, only for Q_{EXH} higher than 109 MW, minimum sub cooling is reached in ECO HP2 thus allowing the generation of the maximum $\dot{m}_{s,HP}$.

Focusing on ST power, for Q_{EXH} greater than 92 MW, Layout_14 shows the best performance compared to all the proposed two pressure level integrated WTE-GT layouts. For lower value, the contribution of reheat on ST power output is limited by the lower amount of high pressure steam mass flow rate, if compared to Layout_13.

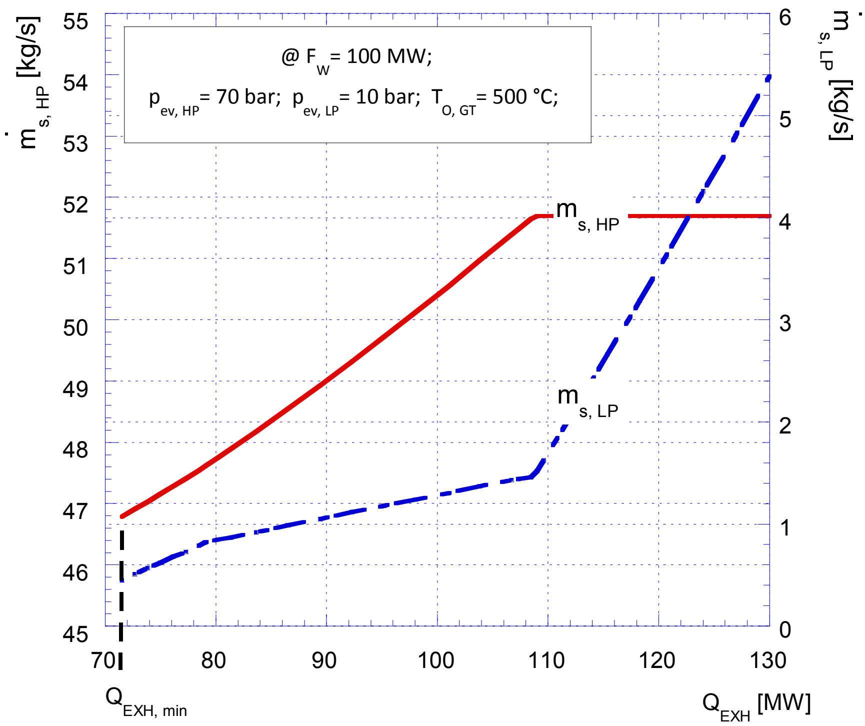

Figure 30 : low and high pressure steam mass flow rate, for Layout_14, as function of GT discharged thermal power.

Figure 32 shows the T-Q diagram relative to Layout_14. Proposed configuration allows a good exploitation of the available GT discharged heat minimizing the HRSG outlet temperature and the difference between gas and steam/water lines.

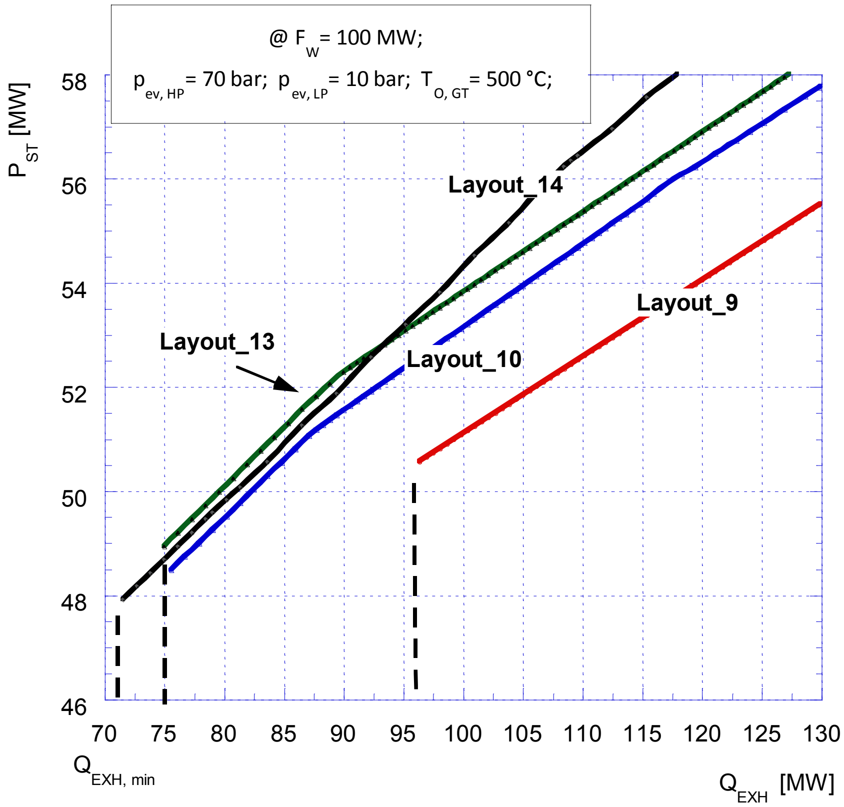

Figure 31 : ST power output for Layout_14 as function of GT discharged thermal power.

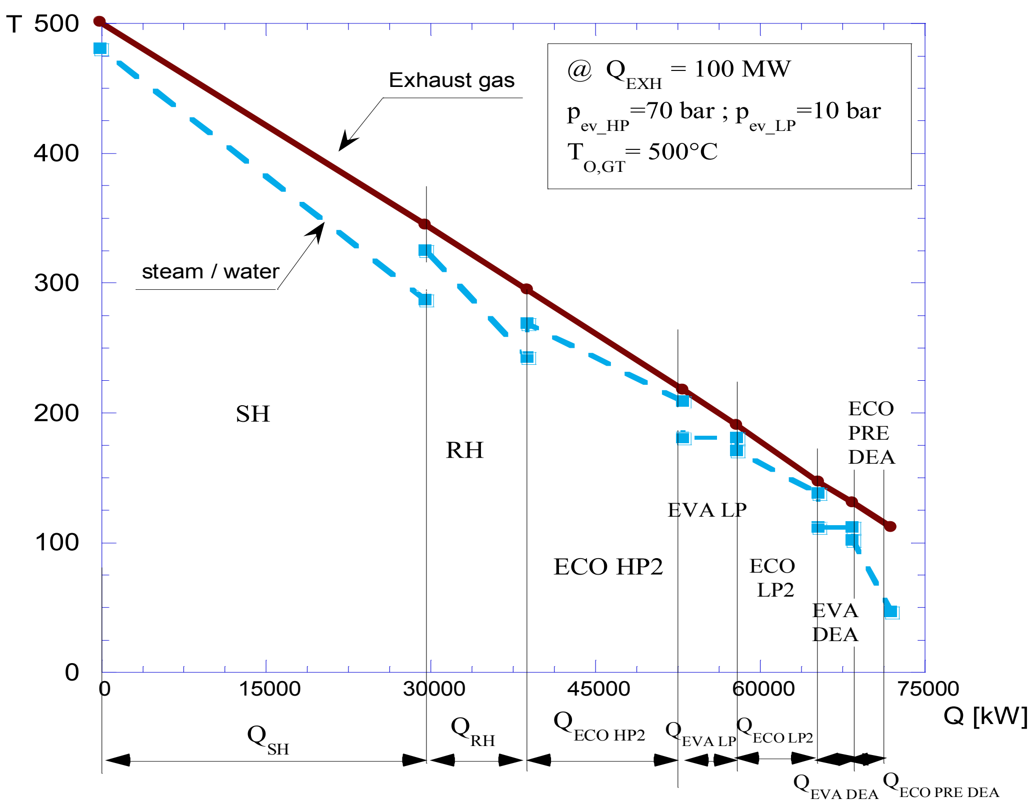

Figure 32 : T-Q diagram for Layout_14.

6.3. *Comparative results for WTE-TG two pressure level integrated layouts*

Results for all the proposed two pressure level WTE-GT integrated layouts are summarized in Figure 33 showing ST power output versus the ratio between GT discharged thermal power and thermal input with waste. In the range of Q_{EXH} / F_W between 0.75 and 0.90 performance of the proposed layouts, except for layout_9, are almost coincident: no significant difference in terms of ST power output can be found. On the contrary, when Q_{EXH} / F_W increases over 0.90 layouts results differentiate much more. In particular, when GT discharged heat equals or exceeds the thermal power input with waste, Layout_14 gives the best performance in terms of ST power output. It must be pointed out that, when Q_{EXH} / F_W ranges between 0.75 and 0.90, low pressure saturated steam mass flow rate generated is significantly lower than high pressure steam mass flow rate thus, two pressure levels integrated layouts are about to degenerate in one pressure level integrated configuration.

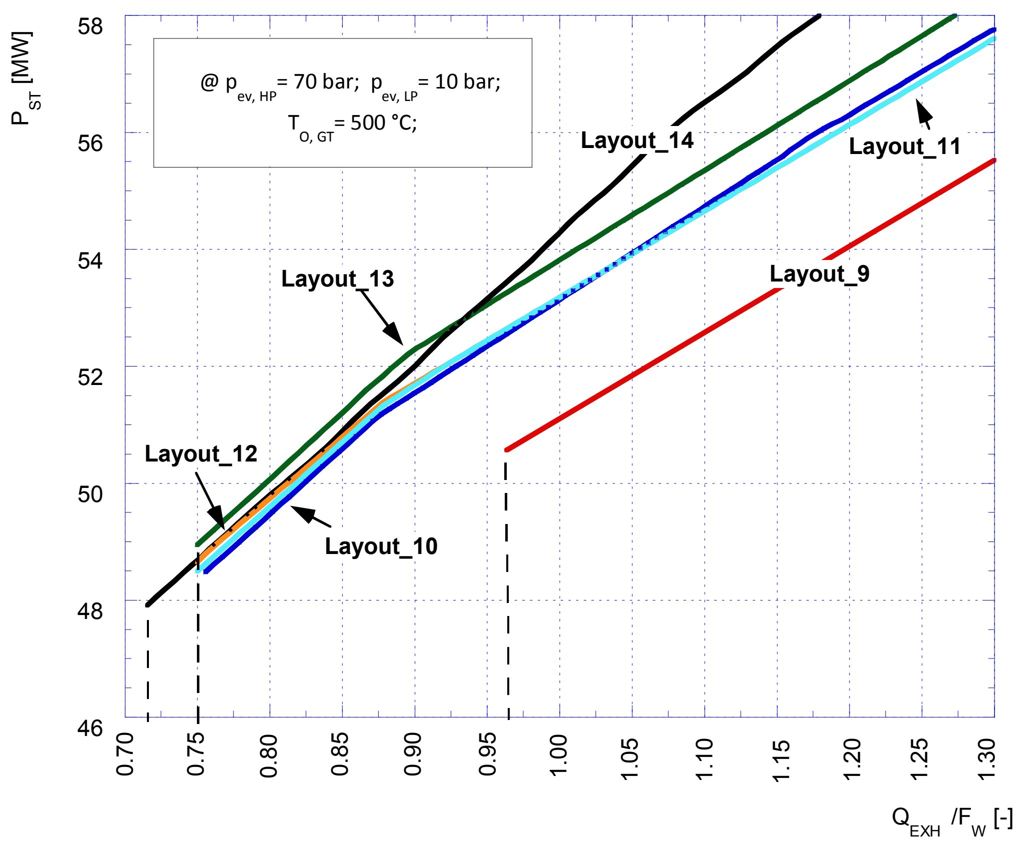

Figure 33 : ST power output for two pressure level integrated layouts as function of Q_{EXH} / F_W .

References

[1] Petrov M. P., Martin A. R., Hunyadi L., 2002, "Hybrid Dual-Fuel Combined Cycles: General Performance Analysis", ASME International Joint Power Generation Conference, Phoenix AZ, USA, June 2002.

7. WTE-GT windbox integration: thermodynamic analysis

This chapter focuses on WTE-GT power plant configurations concerning hot and cold windbox integration where GT exhaust, with or without pre-cooling, are supplied to WTE boiler and used as preheated combustion air. In fact, as described in Chapter 4, high temperature GT exhaust gases makes it well-suited for integration with an MSW incinerator. The gas turbine exhaust typically contains vol. 14% ÷ vo. 16% oxygen, compared to vol. 21% in the fresh air; therefore, in order to provide the same amount of oxygen to the boiler, a considerable increase in flow entering the MSW combustor is required from the gas turbine compared to fresh combustion air.
Considering the GT exhaust gases composition taken as reference (Chapter 5, Table 4, oxygen content equal to vol. 15.31%) and oxygen content assumed in WTE dry exhaust gas (equal to vol. 7% see Chapter 5, Table 5) the following relations can be used to quickly estimate GT exhaust flow rate in order to replace fresh air:

$$\frac{\left(\%O_{2,\mathrm{air}} - \%O_{2,\mathrm{O,WTE}}\right)}{\left(\%O_{2,\mathrm{O,GT}} - \%O_{2,\mathrm{O,WTE}}\right)} \approx \frac{(0.21-0.07)}{(0.15-0.07)} \approx 1.69 \qquad (1)$$

Where $\%O_{2,air}$, $\%O_{2,O,WTE}$ and $\%O_{2,O,GT}$ are volumetric (or molar) oxygen content in fresh air, WTE and GT exhaust gases, respectively. Equation (1) has been derived assumed a constant waste composition and the same oxygen content participating in the reaction. From the above relation it is possible to estimate that about 70% of increase in flow rate is expected if GT exhaust replace fresh combustion air.
Advantages of hot and cold windbox WTE-GT integration can be summarized as follows:

- elimination of WTE combustion air preheating thanks to GT exhaust gas higher temperature;
- reduction of the total exhaust gas mass flow rate and thermal power discharged in comparison with WTE and GT stand alone systems;

- reduction of the environmental impact compared to separate systems: GT exhaust gas would be subject to the same WTE exhaust cleaning treatments;
- reduction of the water fraction in WTE exhaust gases that lower the acid dew point allowing a decrease in WTE minimum allowed temperature: the increase in mass flow rate on the respect of fresh combustion air, for the same water content into the waste, decreases the water fraction in the WTE exhaust.

On the other side, considering the combustion of waste, the exhaust flow should have enough pressure to pass through the waste layer on the grates. This can be accomplished either by the use of air blowers or by expansion to a pressure above the atmospheric level. Moreover, when the gas turbine flow is too large for a given WTE boiler, part of the flow can be bypassed to a stack, or to the convection section of the boiler [1].
Performance of the hot windbox scheme can be further improved, as investigated in [2], for example if external superheating is applied. In this case, the integrated cycle can have both side advantages. The steam superheater, located in the gas turbine exhaust duct is not exposed to the corrosive gases, can achieve the same level as that in the HRSG but has a simpler design and a much smaller surface area than a heat recovery boiler (see Chapter 4, paragraph 4.4).

In the next paragraph, proposed WTE-GT windbox integrated layouts are presented and discussed. The analysis on cold windbox layouts, where GT exhaust before entering the WTE boiler are cooled down by exchanging heat to superheat steam and/or to economize water, is performed keeping constant the thermal power input with waste and the GT outlet temperature for both conditions:

- GT exhaust mass flow rate is calculated in order to replace the fresh combustion air.
- GT exhaust mass flow rate is calculated in order to superheat, at maximum temperature, all of the steam that the WTE boiler generates; in this case being GT exhaust flow rate greater than that required only for waste combustion a bypass, after the heat exchange section, is introduced.

7.1. Hot windbox WTE-GT integrated plant

The investigated hot windbox integrated WTE-GT layout, namely Layout_1hw, is shown in Figure 1. Gas turbine exhaust gas fed the WTE boiler to replace combustion air. Main results of the hot windbox layout are summarized in Table 1 for both cases: GT exhaust replacing all the combustion air and GT exhaust replacing only the primary air. For comparison purpose, results of a WTE stand alone power plant fed with the same amount of waste (F_W) and having the same steam cycle parameters (evaporative and condensing pressure, steam superheated temperature etc.) are also shown.

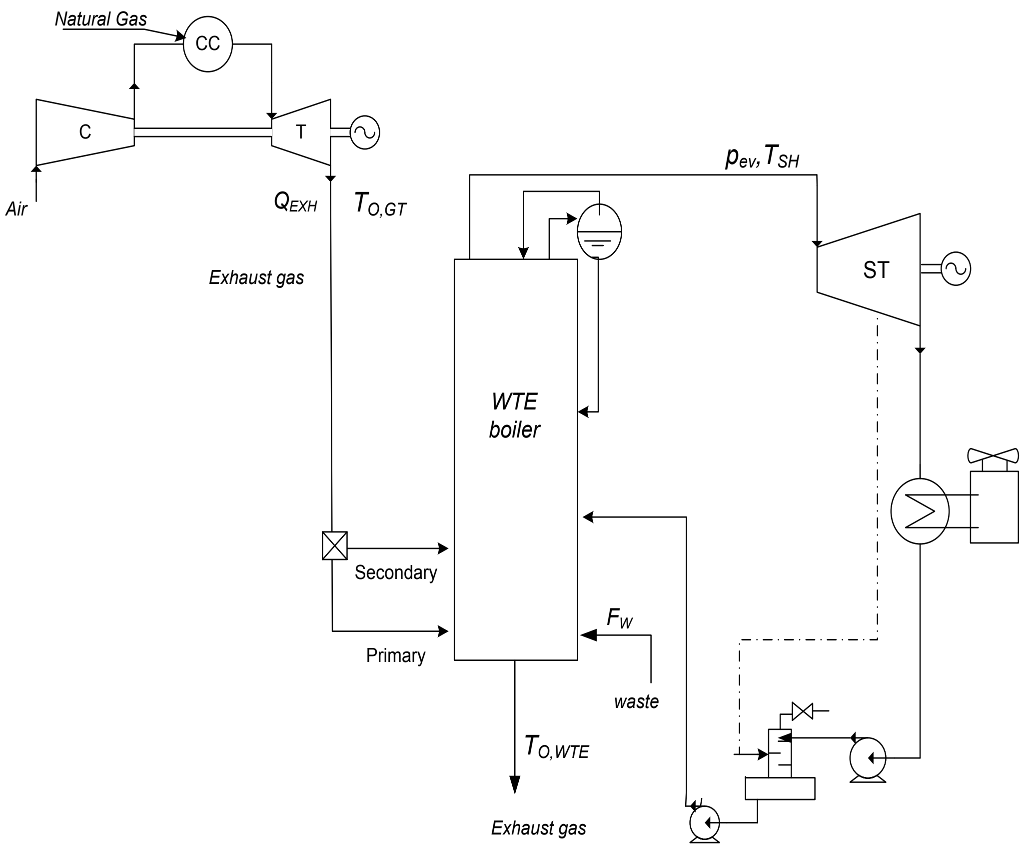

Figure 1 : Layout_hw1

For the investigated layout, an increase in mass flow rate equal to about 55% on fresh air combustion has been found, if GT exhausted fed WTE boiler replacing both, primary and secondary combustion air. The reactants enthalpy increase leads to a corresponding increase in steam mass flow rate generated equal to 41%. For what concern waste combustion, even an increase of about 100 °C in the adiabatic flame temperature has been observed with Layout_1hw compared to WTE stand alone. As a consequence of steam mass flow increase and ST bleed elimination for air preheating integrated plant power output shows an increase equal to 42%. Narrow gains in steam mass flow rate and ST power output (respectively equal to 16% and 17%) have been obtained, if only primary air is replaced with GT exhaust gases.

Table 1 : Layout_hw1 main results.

	WTE stand alone	**Layout_1hw**	**Layout_1hw (only for primary air)**
F_W [MW]	100	100	100
m_{air} [kg/s] (primary, secondary)	55.28 (33.76; 21.52)	-	32.85 (-; 32.85)
p_{ev} [bar]	50	50	50
T_{SH} [°C]	480	480	480
m_s [kg/s]	32.09	45.17	37.22
$m_{GT,exh}$ [kg/s]	-	85.49	34.64
$m_{O,WTE}$ [kg/s]	73.44	92.63	74.64
Q^*_{EXH} [MW]	-	49.31	19.98
Q^*_{EXH}/F_W [-]	-	0.49	0.20
$T_{O,GT}$ [°C]	-	500	500
$T_{O,WTE}$ [°C]	160	160	160
P_{ST} [MW]	29.70	42.19	34.77
T_{comb} [°C]	1155	1261	1272

7.2. *Cold windbox WTE-GT integrated plant*

A variant to the hot windbox repowering approach includes a HRSG to reduce the temperature of GT exhaust transferring superheater from the WTE to HRSG, the most problematic component for what concern high temperature corrosion. In this integrated scheme, known as cold windbox, GT exhaust is first cooled down to a lower temperature level (by various options of, for example, supplying heat for parallel steam generation or feedwater preheating) and then fed to the WTE boiler. Cold windbox arrangements allow for advantages typical of both, gas side and steam/water side integration.
The investigated configurations, namely Layout_cw1, Layout_cw2 and Layout_cw3 are shown in Figure 2, Figure 4 and Figure 6, respectively.
In Layout_cw1 external superheating is applied. In this case steam superheater is located in the gas turbine exhaust duct; GT exhaust enter the WTE boiler with a lower temperature than the hot windbox arrangement of Layout_hw1. T-Q diagram relative to SH exchanger is shown in Figure 3.

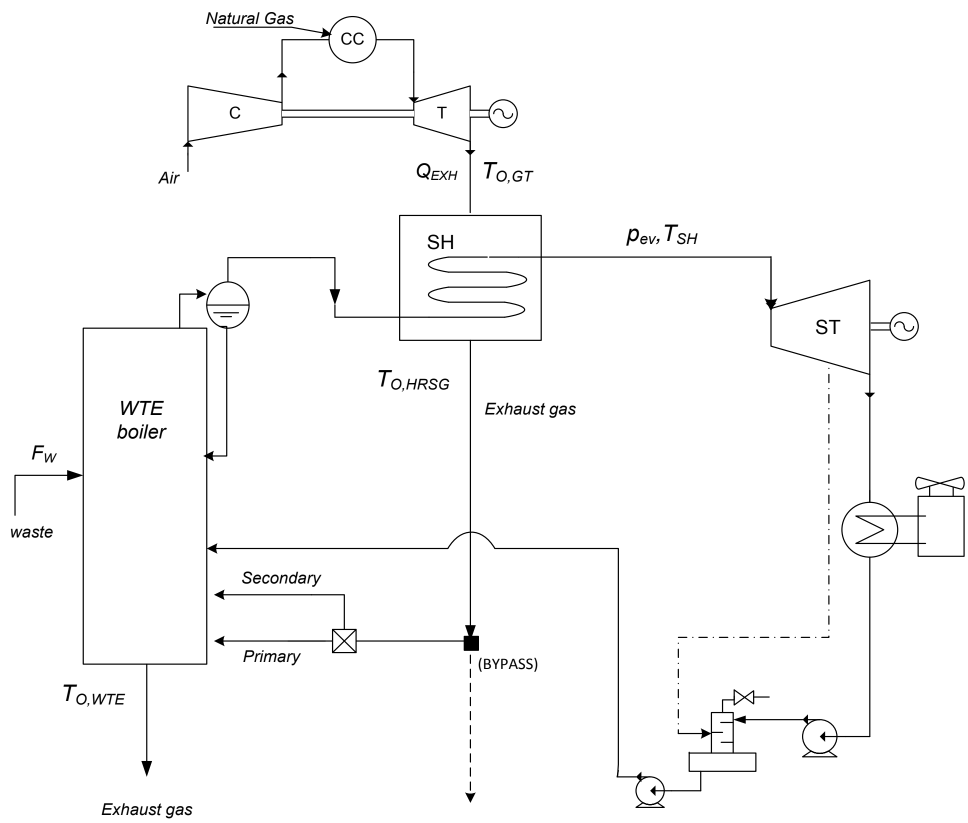

Figure 2 : Layout_cw1

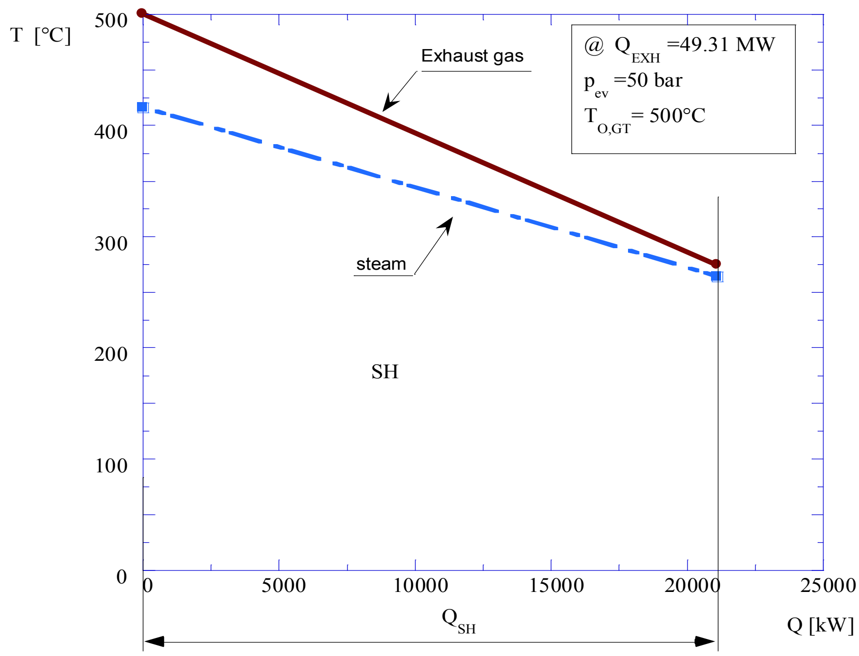

Figure 3 : Layout_cw1 T-Q diagram for SH.

In Layout_cw2 and Layout_cw3, beside the external superheated, and economizer section is placed into GT exhaust path before entering the WTE boiler. Layout_cw2 water out of deaerator, is firstly fed to the ECO1, placed in the WTE convective pass, where part of the economization is performed to be complete in the ECO2, placed in the GT exhaust path.

Instead, for Layout_cw3, a parallel configuration between economizer sections is evaluated. Water stream at deaerator outlet is divided into two streams: a fraction (f) goes into ECO 2, fed by GT exhaust, while the remaining part is sent to ECO1, inside WTE. Before entering the WTE drum, a mixer, combining both streams, is present. Water mass flow rate splitting is adjusted in order to have similar mixer inlet temperatures (or economizer outlet temperatures) and to maximize exploitation of thermal power available into WTE section, thus minimizing exhaust gas temperature. Optimum condition for parallel economisers has been found for f equal to 0.25. The T-Q diagrams relative to Layout_cw2 and Layout_cw3 are shown in Figure 5 and Figure 7, respectively.

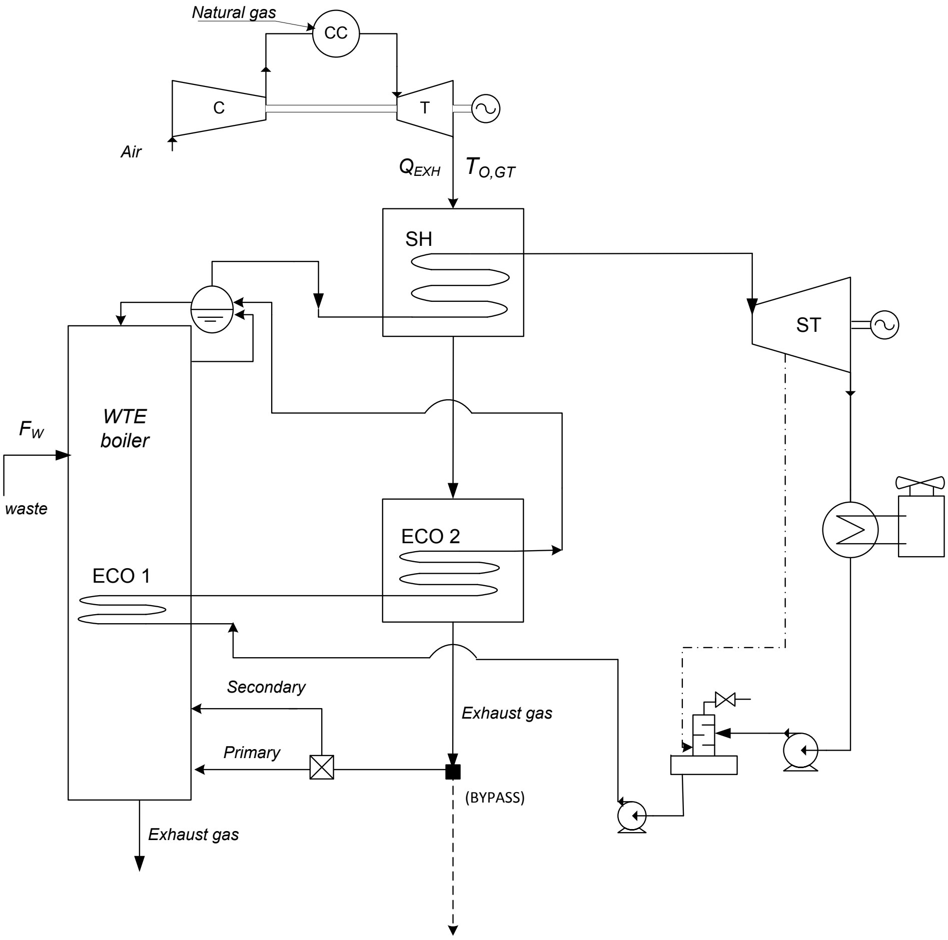

Figure 4 : Layout_cw2

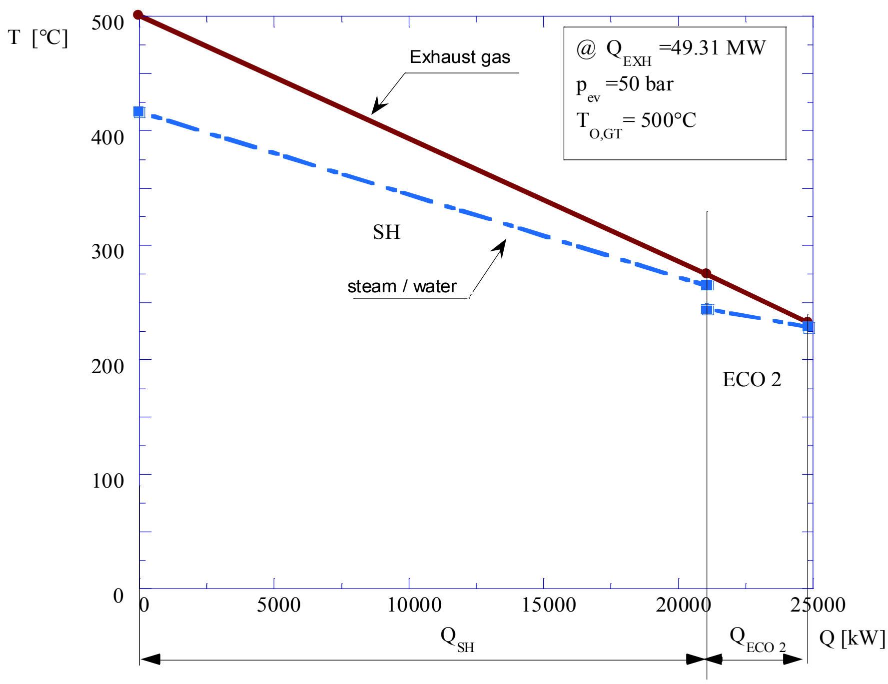

Figure 5 : Layout_cw2 T-Q diagram for SH and ECO 2 sections.

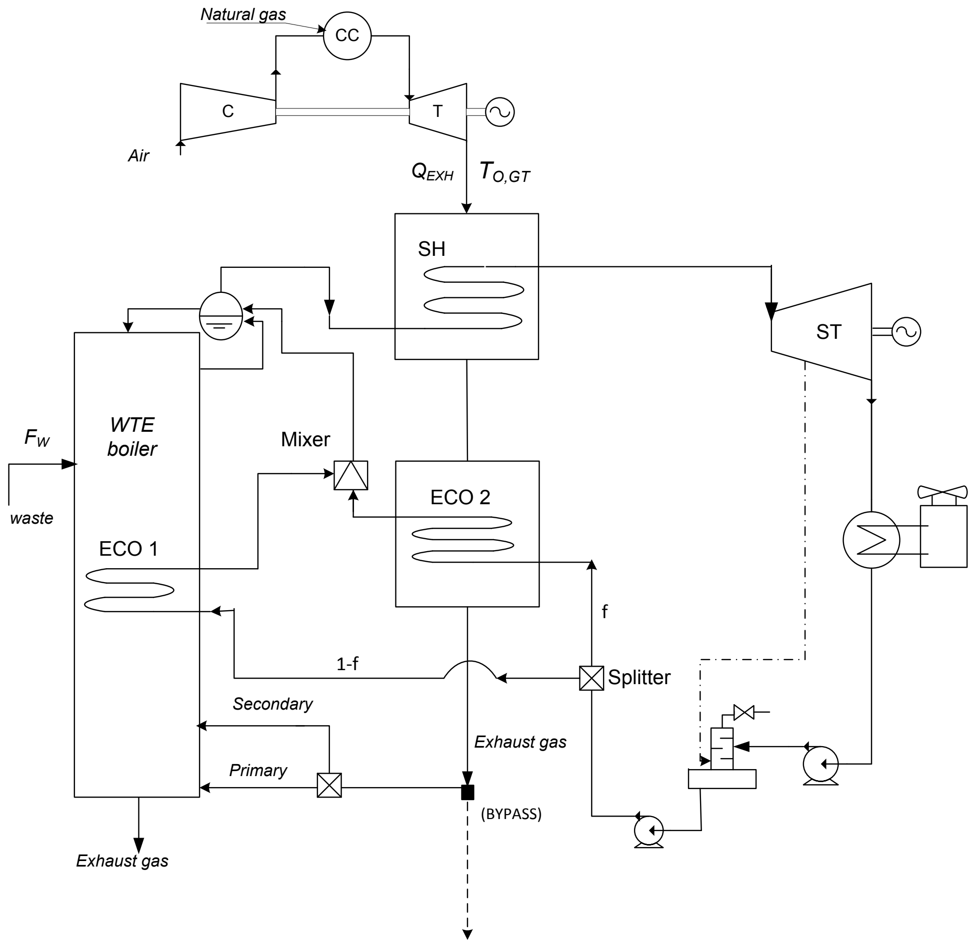

Figure 6 : Layout_cw3

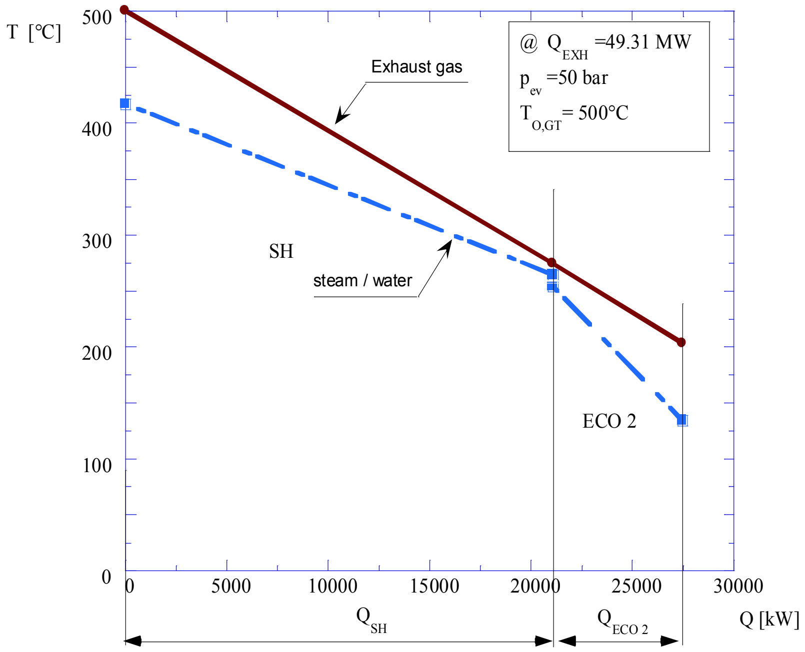

Figure 7 : Layout_cw3 T-Q diagram for SH and ECO 2 sections.

As highlighted from layouts results reported in Table 2, an increase in steam mass flow rate equal to about 48%, compared to a WTE stand alone, has been found. As a consequence of steam mass flow rate increase and limited effectiveness of superheater heat exchanger, GT exhaust mass flow rate necessary to replace fresh combustion air is not enough to guarantee maximum steam superheated temperature. Thus, approach temperature difference is greater than 20 °C.

To guarantee the same steam cycle maximum temperature (480 °C) an increase in GT exhaust heat must be provided; therefore, having assumed a constant GT discharged temperature, and increase in GT exhaust mass flow must occur. Results for the investigated layouts, leading to achieve the maximum steam superheated temperature, are reported in Table 3.

In these cases, layouts modifications must include bypass ducts at the inlet of the WTE boiler for admitting only the right amounts of GT exhaust gasses mass flow rate to replace combustion air.

Table 2 : Layout_cw1, Layout_cw2 and Layout_cw3 main results.

	WTE stand alone	Layout_cw1	Layout_cw2	Layout_cw3
F_W [MW]	100	100	100	100
m_{air} [kg/s] (primary, secondary)	55.28 (33.76, 21.52)	-	-	-
p_{ev} [bar]	50	50	50	50
T_{SH} [°C]	480	416	416	417
m_s [kg/s]	32.09	47.64	47.62	47.38
$m_{GT,exh}$ [kg/s]	-	85.49	85.49	85.49
$m_{O,WTE}$ [kg/s]	73.44	92.63	92.63	92.63
Q^*_{EXH} [MW]	-	49.31	49.31	49.31
Q^*_{EXH}/F_W [-]	-	0.49	0.49	0.49
$T_{O,HRSG}$ [°C]	-	274	232	203
$T_{O,WTE}$ [°C]	160	160	160	165
P_{ST} [MW]	29.70	41.20	41.19	41.03
T_{comb} [°C]	1155	1093	1063	1042

Table 3 : Layout_cw1, Layout_cw2 and Layout_cw3 main results to achieved maximum steam temperature.

	WTE stand alone	**Layout_cw1**	**Layout_cw2**	**Layout_cw3**
F_W [MW]	100	100	100	100
m_{air} [kg/s] (primary, secondary)	55.28 (33.76, 21.52)	-	-	-
p_{ev} [bar]	50	50	50	50
T_{SH} [°C]	480	480	480	480
m_s [kg/s]	32.09	48.00	49.12	48.62
$m_{GT,exh}$ [kg/s]	-	120	125	120
$m_{O,WTE}$ [kg/s]	73.44	92.63	92.63	92.63
Q^*_{EXH} [MW]	-	69.22	72.10	69.22
Q^*_{EXH}/F_W [-]	-	0.69	0.72	0.69
$T_{O,HRSG}$ [°C]	-	283	235	226
$T_{O,WTE}$ [°C]	160	160	160	161
P_{ST} [MW]	29.70	44.85	45.88	45.42
T_{comb} [°C]	1155	1100	1065	1059

7.3. Comparative results for hot and cold windbox WTE-GT layouts

Results for all the proposed hot and cold windbox WTE-GT integrated layouts are summarized in Figure 8 and in Figure 9 showing respectively, the amount of steam mass flow rate generated and ST power output as function of the ratio between GT discharged heat and waste input heat. In correspondence to Q_{EXH} /F_W equal to zero, performance of the reference stand alone WTE working with the same steam cycle parameters are shown for comparison purpose.
Significant increase, for both hot and cold windbox integrated layouts have been found in comparison to the reference WTE power plant, both in terms of steam mass flow rate and ST power output. Moreover, plant performance increases increasing the value of Q_{EXH} . Finally, the operative range in terms of thermal input power ratio, between 0.20 and 0.72, for integrated windbox layouts is lower than optimum range for steam/water side integrated WTE-GT one pressure level (see Chapter 5, paragraph 5.4) while achieved performance are comparable.

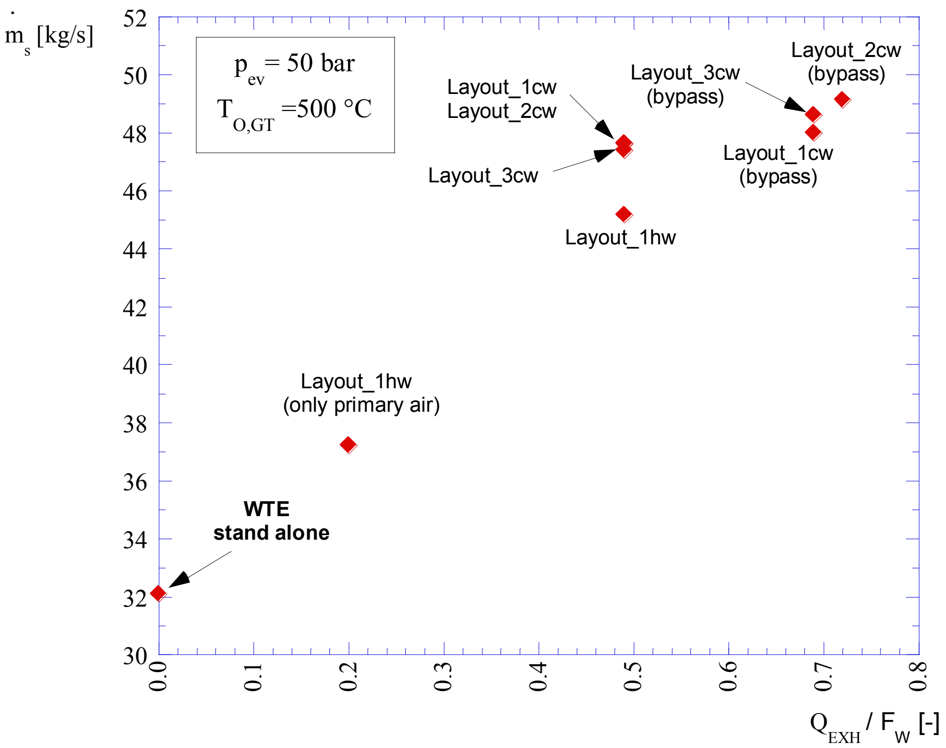

Figure 8 : steam mass flow rate for windbox layouts versus the ratio between GT discharged thermal power and waste input.

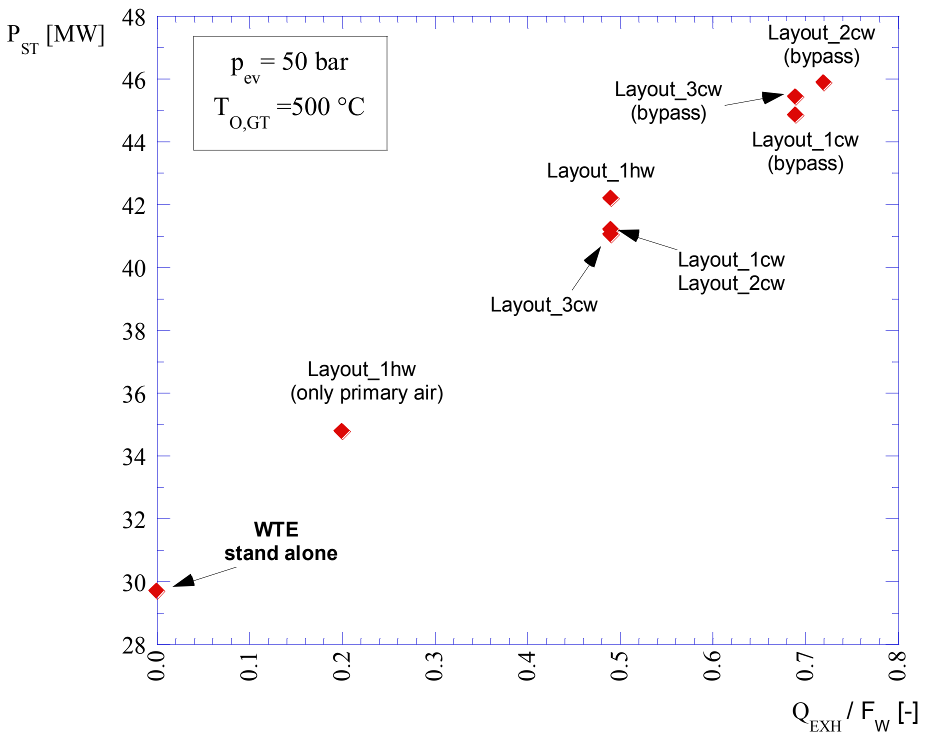

Figure 9 : Steam turbine power output for windbox layouts versus the ratio between GT discharged thermal power and waste input.

References

[1] De Carli , 2004, "*Fattibilità tecnico - economica, inclusi gli aspetti autorizzativi, dell'integrazione inceneritore - ciclo combinato, con riferimento sia all'utilizzo di turbine a gas che di cicli a vapore"*. Technical report, CESI, Ricerca di Sistema, 2004.

[2] Korobitsyn, M. A.; Jellema, P.; Hirs, G. G., 1999, "Possibilities for Gas Turbine and Waste Incinerator Integration", Energy, Vol.24, Issue 9, September 1999, pp. 783-793.

[3] S. Consonni, "Combined Cycles For High Performance, Low Cost, Low Environmental Impact Waste-To Energy". Systems. Proceedings of ASME TURBOEXPO 2000, May 8-11, 2000, Munich Germany, 2000-GT-24.

8. WTE-GT performance indexes definition

In this chapter, the issue of conversion efficiency indexes definition for Multi-Source (MS) energy systems is discussed. An MS system may be defined as a system with more than one input and one, useful, output. Before dealing with specific performance evaluation indexes for a WTE-GT integrated system, a general discussion on performance indicators for MS system is carried out. The conventional first law efficiency, normally used to assess the performance of a single-source power plant, is found to be unsuitable for a multi-input energy system. Thus, several performance indicators, specifically developed, to take into account a system receiving different sources as input and producing useful energy output are presented in this chapther[1].

8.1. Multi-Source Energy System

An energy system receiving "n" energy input streams (F_i), producing useful energy output (U) and rejecting non-useful heat (Q_{NU}) is shown in Figure 1 as a block diagram.
The control surface, surrounding the energy system, can be applied to single components constituting the power plant or to the whole energy system, to evaluate respectively the sub-system or power plant efficiency.

[1] It has to be pointed out that the discussion here presented is restricted to energy systems in which flows are steady.

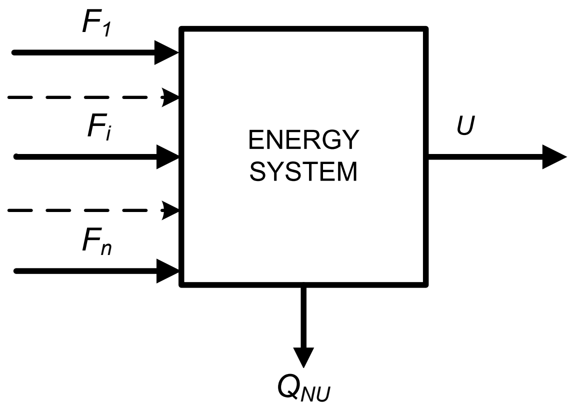

Figure 1 : Basic MS energy system.

Whatever the type of input source or useful output (electrical, mechanical or thermal energy) to the energy system, the energy balance (First law) equation must be satisfied:

$$\sum_{i=1}^{n} F_i = U - Q_{NU} \qquad (1)$$

where F_i represents the total energy contained in the i-th input sources; the expression of energy input can be written, for each input source, as the product of the mass flow and the total energetic input as follows.

$$F_i = m_i \cdot (LHV_i + H_i) \qquad (2)$$

Where LHV_i represents the Lower Heating Value per unit of mass flow; H_i represents the total enthalpy of the input source including in general, not only the static enthalpy, but also the kinetic and geodetic contributes of the inlet fluid; so, the following expression can be written for H_i:

$$H_i = h_i + \frac{u_i^2}{2} + gz_i \qquad (3)$$

The energy system input streams can be multiple and different type of sources as:

- primary fuels, directly available in nature (e.g. natural gas, coal, oil, biomass, etc.) or "energy vectors" obtained with primary energy consuming process, not directly available in nature (e.g. pure oxygen, hydrogen etc.); in this second case, LHV_i must take into account the energetic cost to generate the energy vector.

- heat transfer fluids, characterized by high temperature and consequent significant heat content (e.g. gas turbine exhaust gases, steam or hot water coming from a geothermal source etc.);

In order to account the efficiency of a MS energy system, different performance indicators can be used. They can be divided within two main categories:

- **absolute indexes;**
- **comparative indexes;**

The first one, includes the so called First and Second Law efficiencies. Absolute indexes come as a results of thermodynamics laws. While, the second category includes indexes which compare the output of the MS system with reference separate and single-source systems. Thus, comparative indexes highlight and quantify improvements obtained with the integrated system in comparison with separate productions; as it will be better described in the following paragraphs, comparative indexes need a comparative energetic scenarios to evaluate MS system performance.

8.1.1. Absolute Indexes

First law efficiency

The most commonly used index to evaluate process efficiency conversion is the First Law efficiency, η_I, expressed as the ratio between the useful energy output and the overall supplied energy input, defined as follow:

$$\eta_I = \frac{U}{\sum_{i=1}^{n} F_i} \qquad (4)$$

As known, this index takes into account the amount of energy input and output ignoring the quality of these fluxes; it describes how much energy is needed to perform a particular task, but not how well that energy is used.

Second law efficiency

Since First Law efficiency does not account for the quality of energy being like a quantity-base criteria, it is necessary to introduce a different index to assess the difference between the performance of a system relative to an ideal (reversible) one, which operates between the same thermodynamic limits. The Second Law efficiency, η_{II}, can be defined as First Law efficiency divided by the efficiency of a reversible system, η_{rev}, operating between the same thermodynamic states as:

$$\eta_{II} = \frac{\eta_I}{\eta_{rev}} \qquad (5)$$

Comparing the actual and the reversible processes based on the same input, the above expression becomes:

$$\eta_{II} = \frac{U}{U_{rev}} = \frac{U_{rev} - T_{amb} \cdot \Delta S}{U_{rev}} = 1 - \frac{T_{amb} \cdot \sum_{i=1}^{n} \Delta S_i}{U_{rev}} \qquad (6)$$

Where $T_{amb} \cdot \Delta S$ represents the energy loss due to dissipative processes; the entropy change, ΔS, considers all the no reversible processes which occur into the energy system (e.g. friction dissipations, heat losses to the environment, etc.). Clearly, η_{II} evaluates and gives indications on the potential improvements of each energy conversion process but, on the other side, it is seldom used because of the difficulty in measuring system entropy changes.

8.1.2. Comparative Indexes

To compare the energy output produced with an integrated system receiving multi-input (Figure 2(a)) with respect to single-source energy systems (Figure 2(b)), it is necessary to define new indicators able to evaluate efficiency improvement of the multi-source power plant compared to separate single source generation.

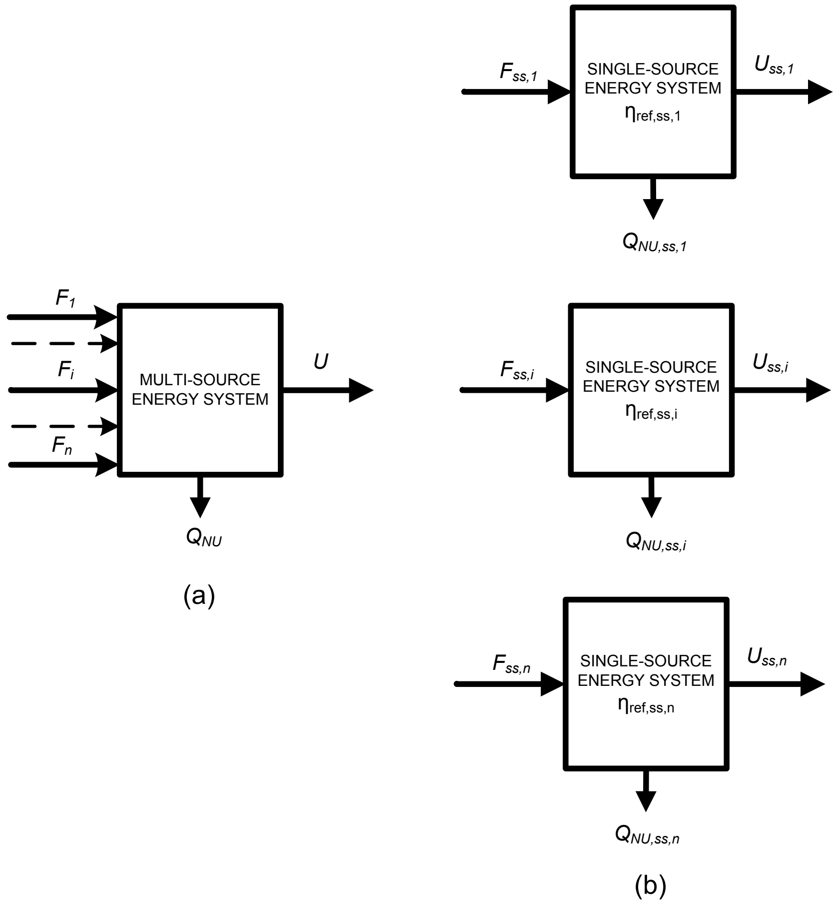

Figure 2 : Basic multi-source (a) and single source (b) energy systems.

One possibility is to evaluate the improvement of the integrated multi-source configuration in comparison with separated and single ones, being equal the energy input. Thus, a comparative index, named Synergy Index , SI, has been defined according to the following explanation:

Synergy Index

$$SI = \frac{U - \sum_{i=1}^{n} U_{ss,i}}{\sum_{i=1}^{n} U_{ss,i}} \qquad (7)$$

The introduced index evaluates the performance of the integrated system giving information about the synergy of the input sources; clearly, when $SI > 0$ the integrated MS energy system is better than single source ones; the output produced with the multi-source system is higher than the sum of output produced with the single-source energy systems. Otherwise, when $SI \leq 0$ the integrated configuration is equal or worse than the collection of separate and single-source systems with the same total input as MS.

Another comparative index which can be introduced is the relative efficiency of the i-th source, η_i, defined as:

Relative Efficiency of the i-source

$$\eta_i = \frac{U - \sum_{\substack{j=1 \\ j \neq i}}^{n} U_{ss,j}}{Fi} \qquad (8)$$

where it is possible to define:

$$U_{sj} = F_{sj} \cdot \eta_{ref,sj} \qquad (9)$$

While the SI evaluates the overall benefit of the exploitation of all the input streams inside the integrated system, η_i try to quantify the benefit of the integration in terms of better exploitation of the i-th inlet stream.

Thus, to evaluate the power output produced with a single-source energy system it is necessary to select an appropriate value of the reference efficiency ($\eta_{ref,ss,j}$) for the considered energy system; clearly, the selection of an appropriate value of $\eta_{ref,ss,j}$ can severely affect the advantages or disadvantages of the integrated system.

Several possibilities to estimate the reference efficiency can be considered; for example the "BAT Reference Document" (BREF) [1] can be used where, for each fuel and technology the BAT (Best Available Technique) are introduced and the value (or the range) for the reference efficiency are defined. For some of the most common fuels, the reference efficiencies are reported in Table 1 as function of the combustion technology. Alternatively, the BREF also expresses a range of efficiency for new and existing plants, as function of the plant type (Table 2).

Table 1 : Efficiency associated with BAT for different fuel as function of the combustion technique [1].

Fuel	**Combustion Technique**	$\eta_{ref,si}$ **[%]**	
		New plants	**Existing plants**
Coal	Pulverized Combustion	43-47	35-40
	Fluidized bed Combustion	> 41	
	Pressurised fluidised bed combustion	> 42	
Biomass	Grate firing	20	-
	Spreader-soker	> 23	
	FBC	> 28-30	

Table 2 : Efficiency associated with BAT for different fuels as function of the combustion technique [1].

Plant type	$\eta_{ref,si}$ **[%]**	
	New plants	**Existing plants**
Gas turbine	36-40	32-35
Gas engine	38-45	-
Gas-fired boiler	40-42	38-40
Combined cycle with or without supplementary firing (HRSG) for electricity generation only	54-58	50-54

8.2. Performance indexes for a WTE-GT integrated plant

To compare the power output produced with a WTE-GT hybrid cycle, receiving waste and natural gas as input, to separate systems output it is necessary to identify and define an index able to evaluate efficiency improvement due to integration. Some proposal on this matter has been introduced in literature but, up to now, a standard definition generally accepted and shared is not yet available.

The difficulty in defining a performance index capable of quantify the efficiency of the integrated system compared to separate generation lies in assign the extra power generated as a consequence of systems integration. In fact, depending on the system taken as reference, performance results can significantly change.

Several indicators are described in the following with reference to WTE-GT HC in order:

- to investigate which is the best way to measure the integrated system conversion efficiency;
- to measure the benefit of this integration in comparison with the scenario in which WTE and GT (or CC) are stand alone systems;
- to establish a criterion for the selection of the best configuration of integrated WTE-GT systems (comparing integrated plant layouts).

Trying to identify the control surface of the integrated system, one possibility is to take into account Figure 3 (control surface A) where system input powers are represented by power introduced with MSW and natural gas, while output are the GT and ST electric powers ($P_{el,GT}$ and $P_{el,ST}$, respectively).

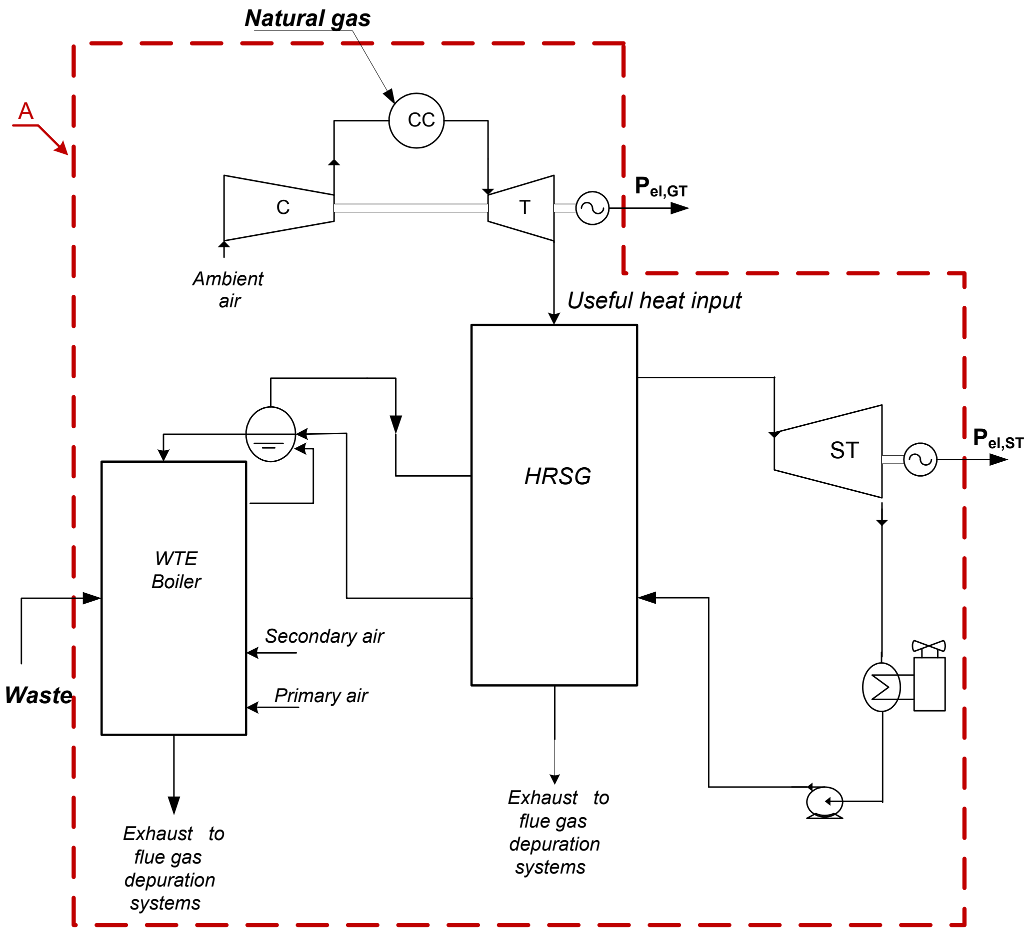

Figure 3 : WTE-GT integrated power plant: control surface A.

<u>First law electric efficiency for the integrated plant:</u>

Taking into account the expression of the First Law efficiency given above, written in terms of power, it is possible to express the efficiency of the WTE-GT integrated plant as:

$$\eta_I = \frac{P_{el}}{\dot{m}_W \cdot LHV_W + \dot{m}_{NG} \cdot LHV_{NG}} \qquad (10)$$

where P_{el} is the total electric power output of the integrated system (sum of the GT, $P_{el,GT}$, and ST, $P_{el,ST}$, electric power produced); $\dot{m}_W$ and LHV_W are, respectively, the mass flow rate and lower heating value of waste input into the WTE section; while, $\dot{m}_{NG}$ and LHV_{NG} respectively, natural gas mass flow rate and lower heating value, refer to the topping GT cycle.

The main drawback of this index is that it does not account for fuels quality difference weighing the same the heat obtained from natural gas and waste combustion.

Natural gas incremental efficiency:

Dealing with comparative indexes, applying definition given in Equation (8), the following efficiency evaluation can be find out, named natural gas and waste incremental efficiency.

The natural gas incremental efficiency, $\eta_{el,NG}$, is the ratio between integrated system power output increase and power input with natural gas, defined as:

$$\eta_{el,NG} = \frac{P_{el} - P_{el,WTE,ref}}{\dot{m}_{NG} \cdot LHV_{NG}} \qquad (11)$$

where $P_{el,WTE,ref}$, is defined as:

$$P_{el,WTE,ref} = \dot{m}_W \cdot LHV_W \cdot \eta_{el,WTE,ref} \qquad (12)$$

Thus, from the total system power output, P_{el}, the electric power output produced with a reference stand alone WTE power plant, $P_{el,WTE,ref}$, fed with the same amount of waste, is subtracted.

This index highlights improvements achieved in the natural gas exploitation through its integration with the WTE section; thus, the extra power generated into the integrated configuration is assigned only to the natural gas section (GT and HRSG).

Waste incremental efficiency:

Differently from the previous one, this index is defined as the ratio between the extra-power generated and the waste thermal input; thus, a reference GT or CC power output is subtracted from the integrated system total power output:

$$\eta_{el,W} = \frac{P_{el} - P_{el,NG,ref}}{\dot{m}_W \cdot LHV_W} \qquad (13)$$

where $P_{el,NG,ref}$, is defined as:

$$P_{el,NG,ref} = \dot{m}_{NG} \cdot LHV_{NG} \cdot \eta_{el,NG,ref} \qquad (14)$$

is the electric power produced with a reference stand-alone GT (or CC) assuming the same amount of natural gas power as input. In this case the extra power achieved with the integrated system is assigned completely to the WTE section; thus, for this indicator it is necessary to define the reference electrical efficiency, $\eta_{el,NG,ref}$, for the conversion of natural gas. Of course, the choice of the reference system can significantly change performance results of the integrated system.
To select reference value for $\eta_{el,GN,rif}$, different possibilities can be considered; for example it can be chosen as:

- the same efficiency of the GT used as topping cycle;
- the average electric efficiency of CC power plants as suggested by AEEG [2];
- the BAT efficiency for the considered technology (GT or CC) according to BREF [1];

Both, Korobitsyn. [3] and Consonni [4] in their studies used $\eta_{el,W}$ to evaluate integrated system performance, but different reference natural gas conversion efficiency have been considered. In particular, Consonni's reference power output, $P_{el,NG,ref}$, is equal to that of a stand-alone CC with the same basic features of the integrated plant: same gas turbine, same HRSG arrangement, same evaporation pressure, etc. Moreover the study reported that, since it is unlikely that one would consider a stand-alone CC with a single pressure level, output for a one pressure level stand-alone CC has to be taken equal to that for a dual pressure.

<u>Ratio of integration</u>

It is defined as the ratio between the electric power produced with steam turbine ($P_{el,ST}$) and the electric power that a stand alone WTE plant would generate if fed with the same waste thermal input ($\dot{m}_W \cdot LHV_W$) of the integrated system.

$$RI = \frac{P_{el,ST}}{P_{el,WTE,ref}} \qquad (15)$$

Thus, even in this case it is necessary to select a reference conversion efficiency, $\eta_{el,WTE,ref}$, for the WTE stand alone power plant. Only if $RI > 1$ the integrated system is better than stand alone WTE.

Bottomer cycle incremental efficiency:

The energy performance of the integrated system can be evaluated taking into account the dotted chain control surface surrounds the integrated plant as shown in Figure 4; taking as reference control surface B, the thermal input are i) waste feeding the WTE section and ii) the GT discharged heat entering the HRSG section.

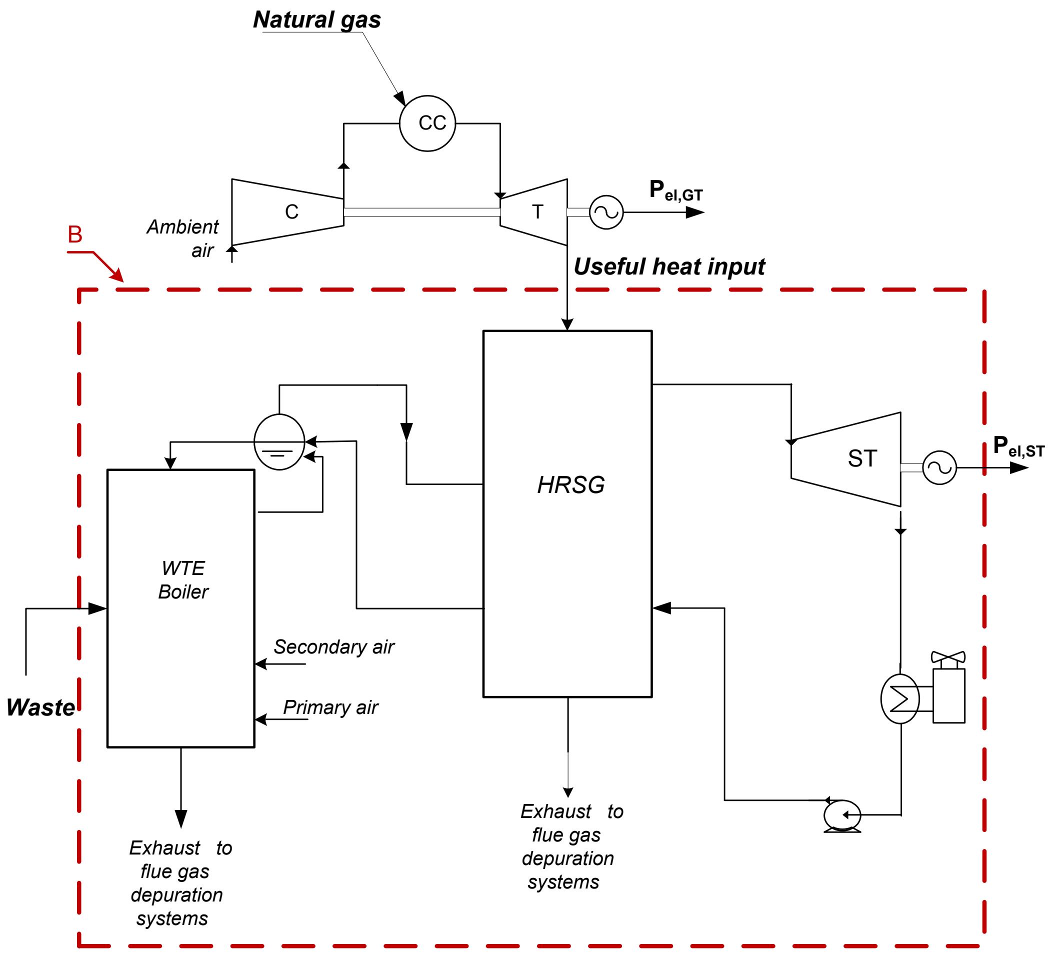

Figure 4 : WTE-CC integrated power plant: control surface B.

In this case, the GT exhaust gases conditions in terms of mass flow rate ($\dot{m}_{exh}$) and temperature ($T_{O,GT}$) are considered and the electric power taken into account, as output from the control surface B, is only the one produced with ST. According to this, it is possible to express the electric efficiency with reference to the bottomer cycle, η^*_{el}, as:

$$\eta^*_{el} = \frac{P_{el,ST} - P_{el,ST,ref}}{\dot{m}_W \cdot LHV_W} \qquad (16)$$

where $P_{el,ST,ref}$ is the power output produced by the steam cycle of a reference HRSG which uses the same amount of GT discharged heat of the integrated plant. In this reference case, the recovery efficiency of the bottoming cycle can be expressed as:

$$\eta_{bottomer} = \frac{P_{el,ST,ref}}{m_{exh} c_{p,exh} \left(T_{O,GT} - T_{ref}\right)} \qquad (17)$$

The expression of the electrical efficiency can be re-written as:

$$\eta^*_{el} = \frac{P_{el,ST} - \dot{m}_{exh} c_{p,exh} \left(T_{O,GT} - T_{ref}\right) \eta_{bottomer}}{\dot{m}_W \cdot LHV_W} \qquad (18)$$

where the value of $\eta_{bottomer}$ can be chosen according to Figure 5. The figure shows calculated values for the bottomer efficiency for typical one and two pressure levels conventional CC, plotted versus the GT outlet temperature. The value of Carnot efficiency is also shown for comparison purpose.

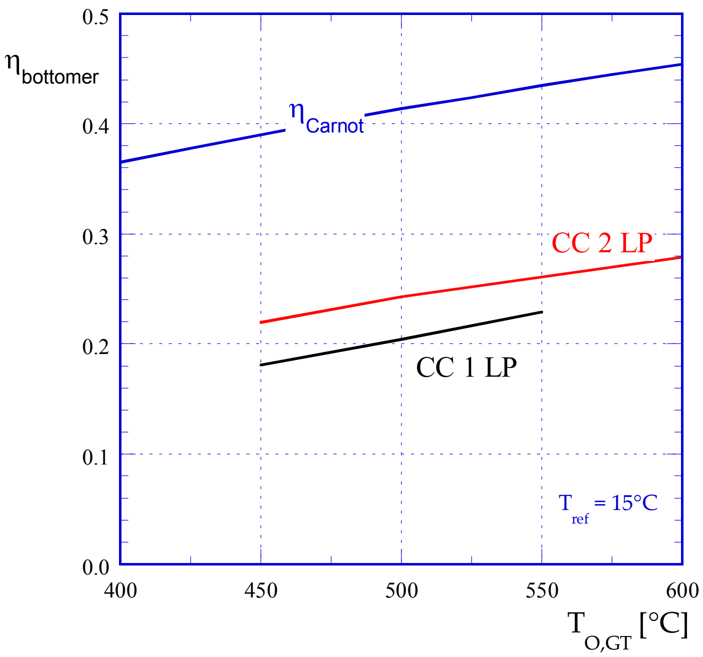

Figure 5 : Bottomer efficiency as function of GT discharged temperature.

In conclusion, the different introduced performance indexes for the WTE-GT integrated plant are summarized in Table 3, reporting their expression, the input and output powers quantity involved and the reference efficiency included inside their definition.

Table 3 : Performance indexes for WTE-GT integrated power plant.

Index name	**Index definition**	**Considered input powers**	**Considered output powers**	**Reference efficiency**
First law electric efficiency	$\eta_I = \frac{P_{el}}{\dot{m}_W \cdot LHV_W + \dot{m}_{NG} \cdot LHV_{NG}}$	Natural Gas and Waste	Total electric power output (GT+ST)	-
Natural gas incremental efficiency	$\eta_{el,NG} = \frac{P_{el} - P_{el,WTE,ref}}{\dot{m}_{NG} \cdot LHV_{NG}}$	Natural Gas and Waste	Total electric power output (GT+ST)	$\eta_{el,WTE,ref}$
Waste incremental efficiency	$\eta_{el,W} = \frac{P_{el} - P_{el,NG,ref}}{\dot{m}_W \cdot LHV_W}$	Natural Gas and Waste	Total electric power output (GT+ST)	$\eta_{el,NG,ref}$
Ratio of integration	$RI = \frac{P_{el,ST}}{P_{el,WTE,ref}}$	Waste	ST power output	$\eta_{el,WTE,ref}$
Bottomer cycle incremental efficiency	$\eta^*_{el} = \frac{P_{el,ST} - P_{el,ST,ref}}{\dot{m}_W \cdot LHV_W}$	GT discharged heat and Waste	ST power output	$\eta_{bottomer}$

References

[1] "Integrated Pollution Prevention and Control Reference Document on the Best Available Techniques for Waste Incineration" European Commission, August 2006.

[2] Autorità per l'Energia Elettrica ed il Gas (AEEG) From: www.autorita.energia.it

[3] Korobitsyn, M. A.; Jellema, P.; Hirs, G. G., 1999, "Possibilities for Gas Turbine and Waste Incinerator Integration", Energy, Vol.24, Issue 9, September 1999, pp. 783-793.

[4] S. Consonni, "Combined Cycles For High Performance, Low Cost, Low Environmental Impact Waste-To Energy". Systems. Proceedings of ASME TURBOEXPO 2000, May 8-11, 2000, Munich Germany, 2000-GT-24.

Conclusion

The increasing interest in waste exploitation as a valuable energy resource has led to a rapid increase in Waste-To-Energy facilities throughout Europe, both in terms of plants number and capacity. At the same time, furthered by the last Waste Framework Directive 2008/98/EC, a reference quality standard for waste conversion has been proposed in order to drive WTE facilities to maximize energy recovery.

Thus, waste is emerging as a good and widely available alternative to substitute conventional fossil fuels. Despite its abundance, due to heterogeneous nature of waste, some differences with respect to conventional fossil fuel have to be considered in the chemical-to-electrical energy conversion process. One of the most important limiting aspect for WTE power plants is corrosion, mainly influenced by heat exchanger wall temperature and flue gases temperature. Thus, the characteristics of the produced steam, pressure and temperature, play a fundamental role in corrosion generation. Because of this, steam cycle parameters are bounded and conversion efficiency of a Waste-To-Energy power plant hardly exceeds 30%, with a net electric efficiency, typically, lower than 25%.

In this context interest in exploring and investigating technologies to maximize waste conversion into energy and to eliminate or, at least, reduce the limiting aspects of the WTE technology has growing.

As a preliminary step of the research activity, a thermodynamic analysis has been carried out with the aim to investigate the influence of the main steam cycle parameters and plant configuration on WTE efficiency. Different strategies have been analyzed and compared. Benefits of a thermodynamic cycle upgrade for a WTE power plant have been quantified, highlighting that different strategies can be performed in order to increase the electric efficiency of a small-medium WTE power plant. Depending on the strategy, or the set of strategies applied, WTE efficiency, starting from 25% can increase up to 33%. Some of the investigated possibilities (like increasing evaporative pressure and steam superheated temperature) may involve specific solutions to protect the integrity of the waste fired boiler; on the contrary, other strategies are simpler, more economic and can be applied without considerable changes. As suggested from the study, regenerative cycle with feed water preheating has a significant effect on WTE power plant efficiency.

Through this research activity the possibility to create a hybrid cycle combining a GT and a WTE, respectively working as a topper and bottoming cycle, has been studied. Two basic types of hybrid dual-fuel combined cycle arrangements has been identified:

- windbox repowering, where the GT exhaust, with or without pre-cooling, are supplied to the bottoming boiler and used as combustion air for waste firing;
- steam/water side integrated hybrid cycle, where GT discharged heat is used for steam superheating and/or feed water preheating and/or additional steam generation parallel to the WTE boiler.

WTE-GT integrated system sharing the steam cycle, sharing the flue gas paths or combining both ways have been investigate, from a thermodynamic point of view.

Focusing on steam/water side integration, the carried out analysis investigates and defines the logic governing plants match in terms of steam production and steam turbine power output, as function of the thermal powers introduced. Results of the study, for one pressure level integrated layouts, suggest that an optimum WTE-GT plant match in terms of system input thermal powers must be pursued to maximize steam generation, steam superheated temperature and to minimize exhaust gas temperature. Moving far from optimum condition means: i) oversize the gas turbine without additional benefits in terms of generated steam mass flow rate or Heat Recovery Steam Generator effectiveness, ii) depress the WTE section decreasing the amount of generated steam or, iii) working with low steam cycle thermodynamic efficiency. Thus, the carried out thermodynamic and parametric analysis provides useful guidelines in selecting optimum gas turbine size to match WTE-GT maximum performance. Results of the study highlight that the higher are the steam cycle parameters (evaporative pressure and steam superheated temperature) the higher must be the gas turbine discharged thermal power on the respect of thermal power introduced with waste.
A correspondence between optimum thermal and electric powers ratios can be achieved only if the gas turbine efficiency is low; otherwise, optimum values in terms of electric powers ratio can be three times that found in correspondence to input thermal powers.
Comparing steam mass flow rate produced in the WTE-GT integrated system with a WTE stand alone, significant increases have been found: steam turbine new capacity for the integrated plant increases from 35 % up to 70 %, depending on steam cycle evaporative pressure and maximum temperature.

Focusing on WTE-GT two pressure level integrated layout, best performance has been found if high pressure saturated steam is generated inside the WTE boiler, leaving the generation of low pressure saturated steam inside the Heat Recovery Steam Generator. The thermodynamic analysis has evidenced that there is a minimum value of GT

discharged thermal power which allows the production of low pressure saturated steam; below the minimum, two pressure levels integrated layouts are about to degenerate in one pressure level integrated configuration.
Performance for WTE-GT two pressure level integrated layouts can be higher that that achieved for one pressure level but the operative range, in terms of thermal power introduced, is skewed towards GT, meaning that higher must be the gas turbine discharged thermal power on the respect of thermal power introduced with waste. Differently from WTE-GT one pressure level integration, an increase in GT discharged heat always leads to additional benefit, increasing the low pressure steam mass flow rate, thus increasing the steam turbine power output.

Significant increase for both WTE-GT hot and cold windbox integrated layout compared to a reference stand alone WTE power plant, in terms of steam mass flow rate and ST power output have been found. By the way, about 56% of increase in WTE boiler flow is expected if GT exhaust replace fresh combustion air this, along with the higher temperature of GT exhaust, can cause a redesign of the furnace burners and of the convective section.
Cold windbox arrangements allow for advantages typical of both, gas side and steam/water side integration, because external superheating, located in the gas turbine exhaust duct, is applied. Moreover, the operative range in terms of thermal input power for integrated windbox layouts is lower than optimum range found for WTE-GT steam/water side integrated layouts, while achieved performance are comparable.

Finally, the issue of conversion efficiency indexes definition for the integrated WTE-GT energy system has been discussed. Several performance indicators, specifically developed to take into account a system receiving different sources as input and producing useful energy output, have been defined, in order to: (i) investigate which is the best way to measure the integrated system conversion efficiency; (ii) measure the benefit of this integration in comparison with the scenario in which WTE and GT are stand alone systems; (iii) establish a criterion for the selection of the best configuration of integrated WTE-GT system (comparing integrated plant layouts).
The difficulty in defining a performance index capable of quantify the efficiency of the integrated system compared to separate generation lies in assign the extra power generated as a consequence of systems integration. Depending on the system chosen as reference, results of the integrated system can significantly change.