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EVALUATING THE POTENTIAL OF INNOVATIVE WASTE-TO-ENERGY PLANTS WITH REHEAT

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Abstract

Waste-to-energy couples the problem of municipal solid waste disposal with the green power production requirement. Using waste as fuel guarantees the diminishing of landfills, which create a certain amount of environmental problems as well as pollutants and space occupation.

However, waste is a poor fuel because of its unforeseeable composition, which does not allow the highest optimisation of the boiler because a certain flexibility must be guaranteed. Moreover, its high content of ash and fixed carbon and its low heating value require big dimensions plants in order to produce the same quantity of power (electric and/or thermal) that a smaller traditional-fuelled power plant would produce. As a result, costs increase because of the higher dimensions as well as the costly cladding that this kind of plant requires.

In fact, another problem of waste as fuel is the great amount of corrosive mixtures it produces during combustion. They require the presence of expensive cladding in the first part of the convective section of the boiler.

As a result, the re-heat procedure, normally adopted in traditional power plants, do not find space in a plant where the warmer convective section is critical.

The present thesis is located in the middle of the last issue, trying to decouple the hot corrosion problem with the re-heat introduction, in order to increase the plant whole efficiency. The introduction of gas-quench allows increasing the plant efficiency with the re-heat possibility but with a higher plant complexion as counterbalance.

Therefore, the thesis aims to analyse performances and costs of the innovative configuration plant, comparing them with the traditional waste-to-energy plant without re-heat as a reference, in order to understand the convenience in applying the innovation introduced.

Keywords: waste-to-energy, power plant, gas-quench, re-heat, innovation, waste, simulation

Sommario

L'incenerimento dei rifiuti accoppia il problema dello smaltimento di rifiuti solidi urbani (RSU) con la richiesta di produrre energia elettrica con fonti rinnovabili. Utilizzare il rifiuto come combustibile garantisce una diminuzione del numero di discariche su territorio nazionale ed internazionale, le quali generano diversi problemi ambientali, così come inquinanti e occupazioni di ampi spazi territoriali.

Tuttavia, il rifiuto è un combustibile di scarsa qualità a causa della sua composizione non prevedibile e sempre differente, che porta ad una non perfetta ottimizzazione della caldaia a causa di una richiesta di flessibilità che deve essere da essa garantita. Inoltre, il suo alto contenuto di ceneri, di carbonio fisso ed il suo basso potere calorifico richiedono impianti di dimensioni maggiori per produrre la stessa quantità di energia (elettrica ed eventualmente termica) che un impianto alimentato da un combustibile tradizionale potrebbe produrre. Di conseguenza, i costi per la costruzione ed il mantenimento in esercizio di un inceneritore aumentano, così come quelli necessari all'aggiunta di costose ricoperture degli scambiatori che questo impianto richiede.

Infatti, un altro problema del rifiuto utilizzato come combustibile, è la creazione di una relativamente grossa quantità di agenti corrosivi prodotti durante la combustione. La loro presenza richiede una ricopertura delle prime parti della sezione convettiva della caldaia, al fine di evitarne danni sostanziali.

Di conseguenza, la procedura del ri-surriscaldamento normalmente adottata negli impianti tradizionali per la produzione di potenza, non può trovare spazio in un impianto in cui le sezioni convettive più calde sono anche le più critiche.

La presente tesi si colloca al centro di quest'ultima problematica, cercando di disaccoppiare il problema della corrosione a caldo con l'introduzione della pratica di ri-surriscaldamento, allo scopo di aumentare l'efficienza globale dell'impianto. L'introduzione del *gas-quench* permette di aumentare il rendimento dell'impianto grazie alla possibilità del ri-surriscaldamento, ma a prezzo di una maggiore complessità.

Lo scopo della presenza tesi è quindi quello di analizzare le prestazioni ed i costi dell'impianto che presenta la configurazione più innovativa, confrontandoli con quelli di un inceneritore tradizionale che non preveda la possibilità del ri-surriscaldamento, allo scopo di valutare la convenienza o meno dell'applicazione dell'innovazione.

Parole chiave: incenerimento, impianto di potenza, gas-quench, ri-surriscaldamento, innovazione, rifiuti, simulazione

Summary

The first two chapters introduce the reader to the waste-to-energy world, in particular, *Chapter 1* develops a critical analysis on the different modalities for waste disposal, analysing advantages and disadvantages and the situation of waste disposal in Italy and around the world.

Chapter 2 illustrates the state of the art for steam cycles and waste-to-energy plants. The first is dedicated to the Carnot and Rankine cycle, explaining the most common methodologies to raise the steam cycle efficiency, while the second part introduces three examples of waste-to-energy plants in Europe.

Chapter 3 introduces the problem of hot corrosion in waste-to-energy plants and suggests possible solutions, exploring even the ones studied only as concept plants and other configurations already implemented in existing waste-to-energy plants.

Chapter 4 shows the methodology and calculation tools used for the analysis of the innovative configuration plant. It introduces the traditional waste-to-energy plant used as reference for the innovative configuration one, the software used to simulate the plant and extract performances and costs, the methodology, the mass and energy balances calculated by the plant in order to provide the correct outcome and finally the heat exchangers dimensioning verifications made.

Chapter 5 illustrates the innovative configuration plant, showing its scheme and detailed configuration and focusing on the gas-quench technology, explaining how it solves the problem highlighted in Chapter 3.

Chapter 6 presents all the assumptions made in order to obtain the outcome provided in the further chapters. The assumptions are divided by data assumptions, thermodynamic, cost and dimensioning assumptions.

The last three chapters are dedicated to outcome analysis, in particular *Chapter 7* shows the performances evaluation, introducing the re-heat pressure choice and presenting all the performances subdivided by evaporation pressure adopted, confronting them with the results obtained by the reference plant.

Chapter 8 introduces the cost estimates provided by the software, subdivided by steam cycle, flue gases cycle and auxiliaries.

Finally, *Chapter 9* comments the outcome of the previous two chapters and evaluate the conclusions about the convenience in applying the innovation to a waste-to-energy plant. It also foresees the future development regarding the plant analysed in the present thesis.

Estratto

Il tema dei rifiuti rappresenta uno dei grandi dibattimenti del mondo odierno. In particolare, lo smaltimento del rifiuto solido urbano (RSU) può seguire diversi percorsi.

La prima strategia si colloca a monte della produzione del rifiuto e richiede proprio una *riduzione* della produzione del rifiuto stesso, agendo sulla progettazione del bene e sul packaging in modo che le parti di scarto risultino minime.

Il rifiuto, una volta prodotto, può poi essere *ri-utilizzato* (ovvero senza subire trasformazioni di sorta) oppure *riciclato* (il rifiuto viene trasformato in un oggetto diverso).

Infine, per quella parte di rifiuto per la quale non è stato possibile applicare le tre voci precedenti, si *recupera* l'energia contenuta nella materia utilizzando il rifiuto come un combustibile negli impianti di produzione di potenza.

Altri metodi per lo smaltimento del rifiuto includono il compostaggio e le discariche controllate (che tuttavia presentano elevati rischi ambientali).

La presente tesi si concentra sul recupero di energia da rifiuto, in particolare sugli impianti di incenerimento: si tratta di tradizionali impianti per la produzione di energia elettrica (ed eventualmente di cogenerazione) che sfruttano il rifiuto come combustibile.

La presente tesi si prefigge lo scopo di incrementare il rendimento complessivo di un impianto di incenerimento per la produzione di potenza.

Il rifiuto presenta una serie di caratteristiche che lo rendono sfavorevole rispetto ai più tradizionali combustibili per un impianto per la produzione di potenza. La sua composizione sempre variabile non permette di ottenere una completa ottimizzazione della caldaia, che dovrà infatti garantire una certa flessibilità per mantenere una buona resa entro tutti gli intervalli di variazione di composizione del rifiuto. Inoltre, la presenza di ceneri e carbonio fisso permette di produrre una quantità molto inferiore di energia elettrica rispetto a quella prodotta da un impianto tradizionale, a pari potenza in ingresso (ovvero portata di combustibile per potere calorifico). Di conseguenza, gli inceneritori risultano impianti di grossa taglia, maggiore rispetto a quella richiesta per un impianto alimentato a metano.

La natura del rifiuto, inoltre, non permette l'applicazione delle più tradizionali tecniche per l'incremento del rendimento del ciclo a vapore. Durante la combustione, infatti, il rifiuto sviluppa una serie di composti di natura altamente corrosiva, che possono seriamente compromettere la resistenza delle prime sezioni convettive della caldaia. La temperatura all'ingresso della sezione convettiva risulta pertanto limitata da questi fenomeni ad un valore di 650°C, il massimo utile per non compromettere la resistenza dei materiali. Per evitare la corrosione a caldo, vengono normalmente adottati dei cladding di Inconel, per i primi banchi di surriscaldamento, che però incidono pesantemente sui costi globali d'impianto.

Il risultato netto è una criticità piuttosto evidente delle prime sezioni convettive, motivo per il quale normalmente non viene adottata la procedura di ri-surriscaldamento, che prevedrebbe

un raddoppio della sezione critica, nonché un raddoppio dei costi che più che supererebbero il beneficio dell'extra energia prodotta grazie all'aumento del rendimento.

A questo scopo, da dieci anni a questa parte, numerose ipotesi e tentativi sono stati svolti al fine di individuare una configurazione d'impianto che permetta l'introduzione del ri-surriscaldamento senza gravare eccessivamente sui costi dell'impianto e, possibilmente, diminuendo la criticità delle sezioni convettive.

L'introduzione del ri-surriscaldamento, per altro, permetterebbe anche di aumentare la pressione di evaporazione, procedura invece non conveniente nel caso di impianto privo di tale accorgimento.

Alcuni impianti sperimentali hanno effettivamente trovato applicazione in inceneritori reali, mentre altri progetti sono stati confinati alla mera sperimentazione, in attesa di ulteriori sviluppi.

La presente tesi si colloca esattamente in questo obiettivo: separare il problema della corrosione a caldo delle prime sezioni convettive con l'aumento di efficienza richiesto per permettere agli inceneritori di avviarsi verso un futuro in cui potranno rendersi economicamente competitivi rispetto alle tradizionali centrali per la produzione di potenza.

A questo scopo viene introdotta l'innovazione del gas-quench. E' una pratica che prevede di ricircolare parte dei gas di scarico in uscita dalla caldaia all'ingresso della sezione convettiva. In questo modo è possibile raffreddare i gas in uscita dalla sezione radiativa, portandoli alla temperatura massima possibile per evitare la corrosione a caldo dei materiali, ovvero 650°C. A causa della temperatura elevata, sarà comunque necessario un cladding di Inconel nel primo banco di scambiatori in parallelo, che tuttavia non verrà intaccato dagli agenti corrosivi grazie all'innovazione introdotta.

Questa pratica permette di aumentare oltre la massima imposta la temperatura di uscita dei fumi di scarico dalla sezione radiativa, diminuendo di fatto la quantità totale di acqua circolante nell'impianto e quindi le dimensioni del waterwall. In questo modo è possibile operare il ri-surriscaldamento dell'impianto senza rischi di corrosione e di aumentare la pressione di evaporazione scambiando comunque tutto il calore richiesto nei banchi di economizzazione.

L'impianto considerato nelle simulazioni viene sempre messo a confronto con un inceneritore a pari potenza introdotta con il rifiuto, ovvero 200MW. L'impianto di confronto presenta parametri tipici degli inceneritori odierni, ovvero una pressione di evaporazione di 70 bar, per una temperatura massima di uscita dei fumi dalla fornace di 650°C, come stabilito dal vincolo spiegato in precedenza.

L'impianto innovativo, d'altro canto, svolge un'analisi di ottimizzazione sia sulla pressione di ri-surriscaldamento (che, come è noto, presenta un punto di massimo del rendimento che corrisponde ad una pressione compresa nell'intervallo tra la pressione di evaporazione e quella di condensazione) che sulla pressione di evaporazione. Per quel che concerne l'ultimo parametro citato, la presente analisi si concentra sulle pressioni di evaporazione di 110, 130, 150 e 170 bar, per una temperatura del vapore all'ingresso della turbina di alta e bassa pressione (ovvero temperatura di surriscaldamento e ri-surriscaldamento) di 450°C.

Al fine di verificare i risultati ottenuti dall’impianto, inoltre, vengono effettuate delle verifiche sui dati, in particolare sul dimensionamento degli scambiatori e sulla composizione chimica dei fumi a camino. In questo modo si garantisce una maggiore consistenza degli output ottenuti dal programma di simulazione.

L’innovazione presenta come contraltare una maggior complessità impiantistica che naturalmente si riflette in un maggior costo globale. La tesi prevede pertanto un’analisi delle performance dell’impianto innovativo e di quella di confronto, strettamente affiancate ad analisi sui costi degli stessi impianti. Nell’ottica di valutare la convenienza effettiva nell’apportare la modifica proposta dalla tesi, la potenza netta prodotta (ovvero quella effettivamente disponibile per la vendita in rete) deve essere affiancata da un costo ragionevole che renda l’impianto conveniente rispetto a quello di confronto.

A questo scopo vengono tratte conclusioni relative tanto alla potenza ottenibile dall’impianto quanto ai suoi costi, per valutare, infine, il break-even point dell’elettricità, ovvero il costo al kilowattora dell’extra energia prodotta grazie all’innovazione introdotta, rispetto all’impianto di confronto. L’individuazione di una configurazione ottimale, combinazione della migliore pressione di ri-surriscaldamento e di evaporazione, valutate nell’ottica del costo minore per l’extra energia prodotta è il fine ultimo della tesi.

A margine, vengono inoltre presentate delle riflessioni relative ad eventuali sviluppi futuri, soprattutto per quel che concerne il comportamento off-design dell’impianto, nonché sulla possibilità (per altro molto comune nei più tradizionali inceneritori) di dedicare l’impianto alla cogenerazione, invece della sola produzione di energia elettrica.

Introduction

Environmental emergences verified in some Italians' regions in the last years make the waste disposal theme actual. In particular, they brought to think if the actual disposal politics are effectively correct of if better politics exist.

The principal thought, when it comes to waste, focuses on, after the prevention, how recover or eliminate them. A disposal system cannot continue basing on landfills, as occurs now in Italy, nor on the thought that waste is completely recyclable. It is always more evident that a unique solution to the problem does not exist, but that it is important to foresee an integrated set of practices capable of solve one part each.

The present work concentrates on energetic recovery from municipal solid waste (MSW) through waste-to-energy plants. One of the many obstacles that these power plants encounter in our country is the many resistances of citizens related to environment and pollution issues, even if it is demonstrated that waste-to-energy plants (provided with the rightful flue gas treatment systems) do not pollute more than traditional power plants for electric energy production. Moreover, smells management has made remarkable advances in the last years and the air outside gates of a modern waste-to-energy plant does not resent at all of the particular nature of its fuel. Waste-to-energy plants constitute, until today, the best solution for waste disposal, because they answer to the problem of growing MSW production and to the requirement always more impellent of energy production with renewable methods. The counterbalance of these plants is their costs compared to the energy they are capable of produce. The variable low nature of waste does not consent to build competitive power plant respect to the more traditional-fired ones. Even if fuel (waste) delivery and supply do not represents a cost (since waste is produced freely and its collection is a service the community pay for), waste-to-energy plants demonstrate to cost more than they earn with (thermal and electric) energy selling. The result is that waste-to-energy plants exist only thank to incentives and green energy politics. In the effort of reducing the gap between earnings and costs, the present thesis bases on increasing the overall waste-to-energy plants' efficiency. Being the efficiency the rate between the energy produced (only electric in this work) and the energy required to produce it (introduced with fuel), a high efficiency consent to obtain higher power production with the same waste amount.

Because of the corrosive nature of waste combustion products, the first convective sections are critical for these plants. As a result, the re-heat practice (which consists in the interruption of superheated vapour expansion within the high-pressure turbine to re-heat it effecting a second passage within the boiler) is not normally adopted in order not to double the critical sections. Re-heat would allow increasing the steam cycle efficiency, as would be furtherly explained in the specific chapter. Obviously, other attempts have been made through the years and some innovative configurations have been created in order to answer to the efficiency increase of waste-to-energy plants issue. Some of them were even actually built, but this configuration is one of the first ones to implement re-heat in the waste-to-energy plants world.

The aim of the present work is to create and simulate a waste-to-energy plant that decouple the hot corrosion problem with the efficiency increase thanks to the implementation of an innovative configuration. In particular, the innovation lays in the gas-quenching technique, which consists in cooling down the flue gas exiting the furnace with part of flue gas recirculated to a required temperature that allows avoiding the dangerous temperature at convective section inlet.

Moreover, the re-heat introduction justifies the effort to raise the evaporation pressure (which represents another way to increase efficiency, but which is a useless cost in case of plants without re-heat) to the most common traditional-fired power plants' values.

The degrees of freedom for the innovative plant, thus, become the re-heat pressure, which presents a maximum for the steam cycle efficiency between condensation pressure and the evaporation pressure, which moves from 110 bar to 170 bar.

Performances and costs are evaluated in order to understand the convenience in adopting a more efficient configuration that, on the other side, would cause an inevitable plant complexion.

The innovative configuration is always placed side by side with a confrontation plant, which objective is to compare performances and costs of the innovative, not-yet realised configuration, with the more traditional and experimented one.

At the end, the cost per kilowatt of electric energy produced would decree the convenience or not of the innovative configuration plant and, eventually, which combination of re-heat pressure and evaporation pressure better suits a potential application.

Chapter 1

Municipal Solid Waste Disposal Methodologies

This chapter analyses the principal methodologies used nowadays in order to dispose of the municipal solid waste. In particular, it is focalised on the current technologies for obtaining energy from waste combustion.

1.1 THE "FOUR R"

As a good practice, the traditional forms of waste disposal (landfill or incineration) constitute the latest destination of waste that is not recoverable in other different ways. The priority, according to the norm in force, is given to the "Four R", which stands for: Reduction, Re-use, Recycle and Recovery.

The treatise below is based on [1].

1.1.1. Reduction

The reduction at source is the primary choice in waste management, since it comes into play upstream in waste production. Thus, design and production phases are asked the major effort, in particular concentrating on the minimization of remaining that is inevitable to be produced downline of the manufacturing phases. This kind of politics often leads to major initial costs, attributed to the necessity of redesigning the product or the package to reduce the quantity of material without reducing its overall quality. Nevertheless, the benefits obtainable in a long-term perspective are very interesting. In fact, using a minor quantity of material per product is the same as reducing the row materials extraction and manufacturing and, at the same time, the necessity of waste disposal.

1.1.2. Re-use

Waste is re-used without being subjected to any transformation. An object that can serve as an example is a glass bottle, which is collected and re-used as it is. Another example are shopping bags: they can be used several times before being thrown away. Re-using the same product means increase the goods value and avoid the production of other goods to perform the same function.

1.1.3. Recycle

Waste is transformed in completely different objects with different functions thanks to special manufacturing treatments. Recycling is intended as the operations of collection, separation, cleaning and treatments necessary to transform waste into secondary raw materials, which can substitute the "primary" raw materials. The advantages in using recycled materials instead of the "conventional" ones are mainly environmental, because they consent energy savings (not obliging to perform all the operations needed to obtain the “primary” raw materials) and a reduction in air, water and ground pollution.

However, if the solution could appear ideal on the environmental point of view, it is not certain that it is the same on the economical perspective. It is important to evaluate with particular attention the costs, the energy required and the environmental impact of the alternative processes and confront them with the characteristics of the traditional ones. In fact, there are cases where the phase of waste treatment entails too high costs, causing the secondary raw materials to be not convenient if compared with the “primary” ones. This difference is often compensated by incentives, but it is necessary to proceed at the productive processes development in order to, in the future, guarantee the secondary raw materials to result more competitive on the market.

1.1.4. Recovery

The remaining waste can be used as a fuel in order to produce energy. One of the ways to dispose of waste, used by humankind since centuries, is to burn them. The waste incineration is, though, a procedure that produces slags and ash, while the heat is wasted. On the other hand, nowadays plants not only burn waste, but they also highly reduce the production of waste elements and capture the energy produced by the combustion. These are called waste-to-energy plants (refer to paragraph 1.3 for further information about waste-to-energy plant typologies).

1.2 OTHER WAYS FOR WASTE DISPOSAL

If waste is not avoidable, it is possible to draw upon other solutions in order to dispose of it.

The treatise below is based on [1].

1.2.1. Composting

A biological process that transforms the organic fraction mechanically separated by the municipal solid waste in an organic fertiliser with low heating value, called compost. The principal problem is represented by its quality, which normally does not result competitive in comparison with synthetic fertilisers. That is because of the high content of contaminants present in the organic waste. Consequently, its value on the market is practically null. Thus, the compost is usable in garden centres applications as well as for gardens and parks maintenance, while its intensive use is not possible in agricultural field.

1.2.2. Controlled landfills

In the hierarchical order proposed by the European Union, this solution appears to be the less desirable, even if it requires the minor technological and economical effort. It is a storage of waste as it is, compacted in order to raise its density, within a site conveniently prepared to avoid liquid and gaseous emissions that could pollute the environment, water and soil. In here are stocked undifferentiated municipal solid waste and all other waste derived from human activities that, after their collection, could not be recycled, re-used or recovered. Since the process of waste disposal can, in some cases, compromise public health, it is essential to predispose a correct system for hygienically and sanitary control and prevention. Thus, it is necessary to avoid the pollutants to transfer from a territory sector to another one, eliminating the propagation vectors of the environmental contamination (introducing waterproof barriers, covering the landfills in order to avoid air pollution, etc.). The sanitary aspect must comprehend all that physical-chemical and biological sectors that can influence human health. When waste is clumped it this way, a series of reactions develop within waste, which can last even for the further fifty years. They entail two major types of emissions:

- **Biological methane**. It is a gaseous mixture composed by CH_4, CO_2, H_2O, H_2S and NH_3, with a LHV of 21,000 kJ/Nm^3. It forms thanks to the biodegradation process of the organic fraction of waste. This gas can be detected and used as a fuel in reciprocating engines, gas turbines and boilers, in order to recover a part of the energy stored in waste. Nowadays, this process of waste-to-energy occurs only in few landfills in the world, so it is esteemed that the release of this gas from landfills in the atmosphere constitutes up to 10% of the methane worldwide emissions, which is a powerful greenhouse gas. However, international standards indicate that up to 90% of biogas can be recovered in a modern landfill.

- **Leachate**. It is a liquid with ultrahigh polluting charge and sinks on the bottom of the landfill. Its presence imposes a preventive waterproof barrier on the bottom of the landfill in order to avoid the contamination of the underlying aquifers. The leachate must be collected and treated with high-efficiency purification machines with high-reliability since it can cause great damages if not correctly treated.

Methane gas can form, with air, explosive mixtures, while carbon dioxide (which is a greenhouse gas, but not a pollutant, as it is), originating the carbonic acid, can contribute to the mineralization of water that circulates in subsoil. Moreover, the oxidative reactions are exothermic so a great amount of heat can be generated within the landfill, producing a significant temperature raise, even in the surface. The temperature increase is likely to cause explosions, thus is important to avoid it. That usually happens in the flat areas subjected to the highest compression. The landfill scarps present, instead, erosion risks that could lead to waterproof barrier damaging and pollutants leakages.
The potentially generable risks of a landfill results to be the following:

- **Air pollution**. Solid waste combustion can generate emission in the atmosphere of undesired pollutant substances (i.e. ash, NO_x, SO_x, hydrochloric acid, hydrocarbons, etc.), which are detrimental for human health. Moreover, the problem of smells propagation exists. They are generated by decomposition of organic substances or by physical-chemical processes: this phenomenon can be limited individuating the more suitable landfill position in respect to wind direction. Generally, fixed or movable windbreak barriers are used in order to limit the wind-effect.

- **Fire primer**. Caused by sparks or flames produced by mechanical vehicles which operate in the landfill, or by operators themselves, or because of the presence of material that is combusting in the waste (i.e. embers, etc.), a principle of flame can prime starting from fuel gasses produced in the landfill (i.e. methane, etc.). Moreover, a risk of self-ignition of fuel gasses originated by waste fermentation is real due to poor materials compaction and consequent presence of oxygen in void spaces. Thus, it is necessary to dispose adequate technologies in order to implement a continuous control of the landfill to signal an eventual presence of self-ignition phenomenon.

- **Water pollution**. The risk of diffusion in superficial aquifers and in soil of toxic substances or microorganisms often present in waste, put forth a difficult sanitary problem, even when it comes to the food chain.

- **Pathogen diffusion**. The landfill rich in domestic waste, with organic substances, constitutes the ideal habitat for animals that are potential carrier of infective diseases. Rodents, birds and insects are the most common in municipal solid waste landfills. Thus, it is important the humid fraction of waste to be collected in a differentiated way or that the waste withstands composting or a mechanical-biological treatment before inserting it in landfills. Moreover, the function of waterproofing and covers can be compromised by the presence of rodents’ burrows, creating fast tracks of meteoric water infiltrations or leachate leakages.

Even though modern landfills can reduce the amount of problems listed above, they are not entirely removable.

1.3 WASTE-TO-ENERGY PLANTS

Having already introduced the recovery theme, this sub-chapter aims to focalise the attention on waste-to-energy plants, explaining their operating principles, advantages and disadvantages. Finally, it concentrates on different plants typologies in order to provide the most accurate range of possible plant configurations.

1.3.1. Aims and operation principles

The major part of waste contains an organic component that can be burned within a waste-to-energy plant in order to obtain energy recovery. Because of the actual waste composition, rich in substances with high LHV as plastics and cellulosic substances, it can be considered as a fuel with full rights.

Energy recovery is operated following the successive simplified steps:

1) Waste is collected and stoked in a room called pit

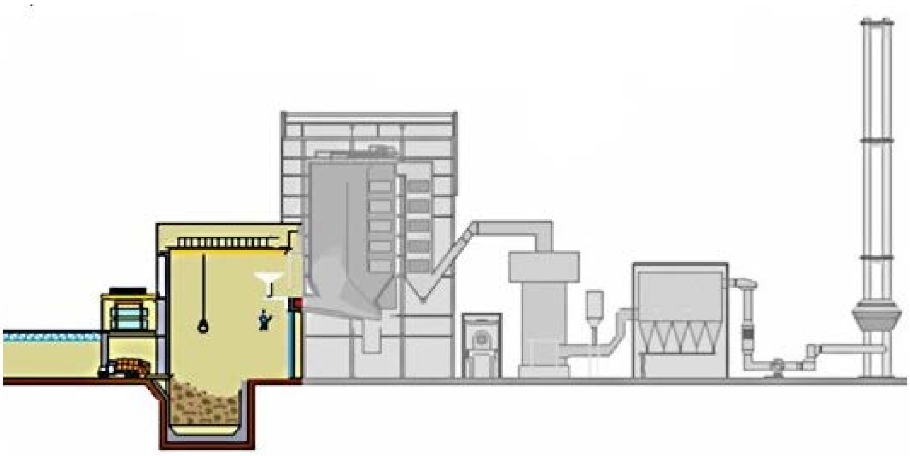


Figure 1. 1 - Waste storage in the pit [2]

2) Waste is moved from the pit to the main boiler, where it is burned in order to recover energy. Hot gasses produced by the combustion flow through the boiler body, heating up and vaporising the water passing through pipes that cover the inner part of the boiler. The vaporised water goes into the turbine body, where it expands, producing electric energy, in a Rankine cycle (details about Rankine cycle and technical specifications about the overall plants will be discussed in Chapter 2)

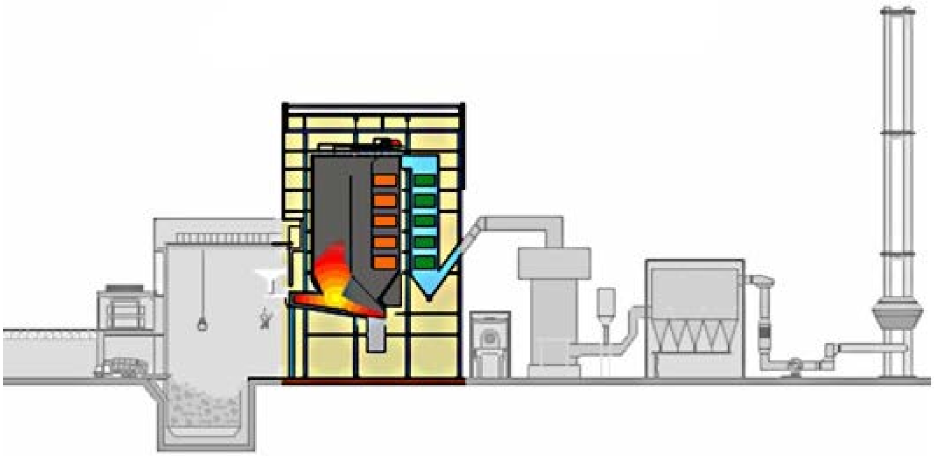


Figure 1. 2 - Waste combustion within the plant's boiler [2]

3) Hot gasses cooled due to the thermic exchange with water, pass through a series of filters that aim to reduce and hold the major pollutants, ash and other dangerous substances produced during the combustion process

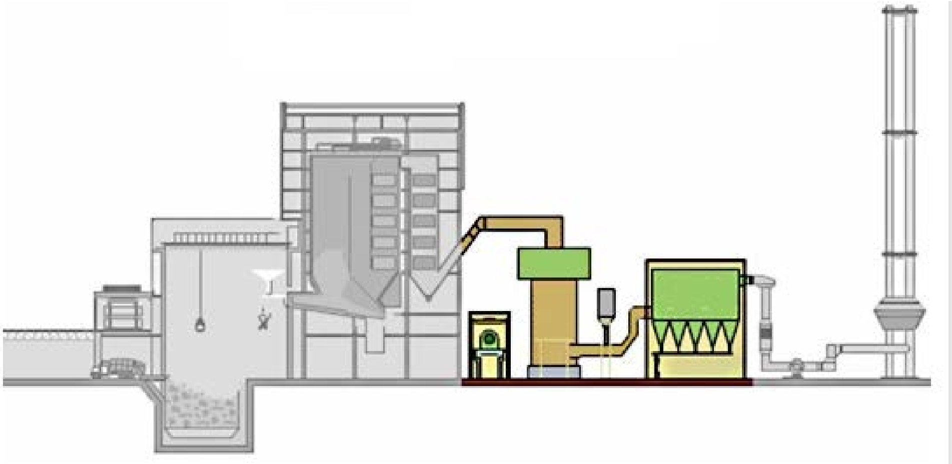


Figure 1. 3 – Flue gas passing through filters before being discharged in the atmosphere [2]

4) Finally, gasses purified of the major pollutants and harmful substances, are discharged in the atmosphere through the stack

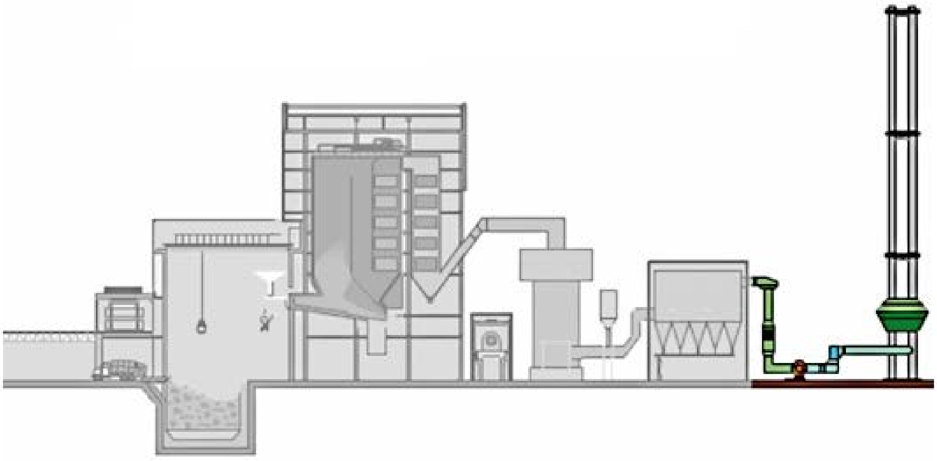


Figure 1. 4 - Gasses discharge in the atmosphere [2]

1.3.2. Advantages and Disadvantages

Through waste combustion, a part of waste's chemical energy is freed and then recovered cooling the combustion gasses. Waste-to-energy constitutes an intelligent and rational approach from one side, but from the other some issues still exist that nowadays do not allow waste-to-energy plants to be competitive with a traditional plant with equal power.

The principal problem about waste-to-energy plants is people perception about waste: instead of seeing it as a resource (it is, with every rights, a fuel produced free of charge), it is perceived as a negative externality, something dangerous, which is important to get rid of. As a result, in many countries among which Italy, a majority percentage of non-treated waste is deployed in landfills.

Moreover, around waste field, there are many prejudices, which mine the diffusion and popular acceptance of waste-to-energy plants. The aim of the following sentences is to debunk some false myths grown around waste theme:

- Waste-to-energy plants **DO NOT** produce more pollutants than traditional plants. [29] While it's true that waste combustion produces more pollutants than traditional fuels combustion, flue gas treatment section within the plants reduces pollutants quantity; it is proven, in fact, that a modern waste-to-energy plant provides the same emissions in atmosphere as a traditional plant.

- Waste-to-energy plants **DO NOT** use more energy than they produce. [29] Energy produced depends on the quantity of vaporised water, which in turn depends on fuel quantity and its LHV. Energy consumed depends on plants auxiliaries, which are not very different from the ones used in traditional energy plants. It is true that, fuel mass rate being equal, waste has a lower LHV than traditional fuels, so the power produced by a waste-to-energy plant should be lower: the difference can, however, being compensated by increasing the waste mass flow (being a zero-cost fuel).

- Waste to energy **IS NOT** opposed to recycling and re-use. [29] In fact, waste-to-energy plants use as fuel, only undifferentiated waste and/or waste that was not possible recycle or re-use.

- Waste-to-energy **IMPLIES** the existence of landfills. [29]. Waste, as wood or other less refined fuels, produces a certain percentage of ashes, which partially remain on the bottom of the boiler and partially are held by the filter system. As a result, there is the need to dispose of ashes, which are inert materials. Thus, even if the landfills do are necessary, many risks listed in sub-paragraph 1.2.2 could be avoided due to the material's not-reactive nature. Waste-to-energy plants allow the energy recovery from waste and provide a volume reduction of ten to thirty times.

Furthermore, waste-to-energy plants show many advantages. When it comes to **economic benefits**, earnings derived from sale of thermoelectric energy produced thanks to waste combustion, contribute to decrease the cost of waste disposal, which is at the expense of service consumers. It can even significantly reduce even expenses in case of integration of a high size waste-to-energy plant with a combined cycle plant. Other disposal modalities beforehand analysed do not present significant incomes.[1]

From an **energetic point of view**, if some energy is produced with municipal solid waste, its production with other sources (among which fossil fuels and nuclear, not-renewable, etc.) can be avoided. It is demonstrable that the TOE amount of primary energy saved thanks to energy recovery from municipal solid waste represents up to 2% of the primary energy necessities of the entire constituency served by the plant.[1]

In **Table 1. 1**, waste is confronted with the principal fuels in terms of a mass equivalent to 1000 TOE.

Fuel	**LHV [kJ/kg]**	**Mass equivalent to 1000 TOE[1] [t]**
Fuel derived from waste	13400	3126

Municipal solid waste Italy	7540	5555
Municipal waste remaining from recycling	10343	4047
Municipal solid waste E.U.	9210	4545
Municipal solid waste U.S.A.	11980	3500
Diesel	45220	926
Fuel Oil	41030	1020
GPL	46060	909
Gasoline	50250	833
Fossil carbon	31820	1351
Anthracite	29310	1428
Wood	18840	2222
Wood coal	10470	4000
Natural gas	34540	1213 [m^3]

[1] TOE = Tonne of Oil Equivalent. 1 TOE = 41860 MJ

Table 1. 1 - Comparison among fuels in terms of a mass equivalent to 1000 TOE [1]

Finally, the general **environmental impact** of grid and fluidised bed waste-to-energy plants of new generation is really restrained and can be even inferior than the one generated by some traditional plants alimented with fossil fuels. Depuration technologies of flue gasses nowadays available and economically sustainable consent to modern waste-to-energy plants to comply, with an ample security margin, to the norm in force on the subject. The flue gas depuration removes flying ashes derived from combustion. These are extremely toxic, because they contain heavy metals (i.e. Hg, As, Cr, Cd, Pb, etc.) that condense during gasses cooling and volatilise during combustion phase. Inert ashes are moved in landfills or, if they have an adequate mechanical resistance, can be employed in construction of products as street curbs, benches, viticulture posts, breakwater blocks, etc.[1]

Moreover, in the environmental balance formulation of a waste-to-energy plant, the plant is supposed to produce thermoelectric energy in substitution of a plant alimented by traditional fuel and thus the traditional plant is supposed to be shut down while the energy from waste would be available. Therefore, in calculating the waste-to-energy plant emissions, the traditional central emissions should be subtracted to the waste-to-energy plant ones, being the energy produced equal.

The most restrictive emissions limits merged with high efficiency for electric generation in modern high size waste-to-energy plants, cause, with the same energy produced, the emissions of the latter to have a greenhouse potential, toxicity, photochemical formation of ozone amount and acidity, lower than flue gas of the central that would be shut down. The advantage is less if small size plants are considered and become negative when it comes to greenhouse potential.[1]

1.3.3. Dedicated plants

Dedicated plants are the ones which main objective is recover energy from waste without integrating it with other plants of fuels.

The treatise below is based on [3].

Grid waste-to-energy plants constitute a well-consolidated solution, widely employed for the combustion of waste as it is and fuel derived from it. The bottom of the boiler, where waste is deposited after being collected from the pit, is made of a series of grids, movable or fixed, which carry waste through its “first” combustion. These plants can burn waste within a wide range of LHV values. However, for the highest LHV values, heat reduction of the grid is operated with water instead of air. The energetic recover from flue gas is realised with an overheated water Rankine cycle. Despite many expedients, a net electric efficiency up to 30% has not yet been reached in plants realised so far.[3] Gasses derived from the waste combustion are highly corrosive, forcing the temperature and overheated vapour pressure limitation. These constitutes a serious limitation to the possibilities to improve the Rankine cycle efficiency, which would be furtherly discussed in Chapter 3.

The aim of the present thesis is to address the problem of temperature limitation, introducing an innovation that should lead to an overall efficiency improvement.

Fluidised bed plants, on the other hand, are so called because waste is burn while being within a bed of inert material, generally silica sand or magnesium carbonate, maintained in suspension in the primary combustive air, inputted from the bottom of the boiler. This solution is not suited for waste combustion, but it is highly recommended for fuel obtained by waste. This fuel is substantially made by paper and plastic contained in waste. It is constituted by small homogeneous pieces as well as a good combustibility: consequently, it burns well in suspension.

A water Rankine cycle with overheated vapour is used for the energetic recover and it presents the same problems highlighted for grid plant. For small size plants, the cost for this solution is generally minor than grid plants.[3]

The last plant presented in this section are the ones that use **pyrolysis and gasification processes**. Pyrolysis is an endothermic process that decompose organic-based materials present in waste and is conducted in a reducing atmosphere. Gasification, on the other hand, is a process of thermic decomposition of organic-based materials with partial oxidation. In both cases the outcome is a gas with a mediocre heating value, which can be furtherly employed in turbogas groups or in reciprocating engines for energy recovery. Both technologies require very complicated plants that, consequently, are delicate and costly in construction, operational and maintenance phases. However, they present some advantages respect to other waste-to-energy plants, such as the minor pollution potentiality of flue gas and the minor quantitative of slags produced, which would be vitrified (in other words make them inert) directly at the end of the process.[3]

1.3.4. Not-dedicated plants

It is possible to increase the energy recovery efficiency of municipal solid waste using it as a fuel in non-dedicated plants. These plants do not use waste as a principal fuel, but it can be integrated as an auxiliary one.

The treatise below is based on [3].

In **thermodynamic cycles integration**, waste-to-energy plant is built beside a combined cycle plant or beside a traditional thermoelectric plant alimented by fossil fuel. A certain water flow is collected from the Rankine cycle of the traditional plant and is transformed in saturated vapour in the waste-to-energy plant’s boiler. Then it is overheated using the fossil fuel combustion gasses (which are cleaner and less corrosive than the waste’s ones), which allows to reach higher overheated vapour temperature and pressure values than the ones obtainable in waste-to-energy plants.

As a result, a certain number of advantages are offered by this solution:

- Higher net electric efficiency than the waste-to-energy plant alone, which leads to an emissions decrease.

- Better scale effect exploitation due to the use of one turbo-alternator with, thus, higher size and efficiency than one that should only serve the waste-to-energy plant. Being equal the amount of fuel, in fact, a waste-to-energy plant would produce less vapour then a traditional plant and so there would be less energy to be elaborated by turbogas group.

- Realisation of a further energetic integration between the two plants, recirculating the waste-to-energy plant’s flue gas in the traditional plant’s boiler, after being conveniently depurated. Their employment as combustive agent is possible because of their decent oxygen content (usually up to 6% weight). The energetic advantage is that flue gas are available with a higher temperature (up to 100°C) than the ambient temperature of combustive air.

On the other hand, the possibility to integrate different kind of fuels exists and is called **co-firing**. It consists in substituting part of the fuel obtained by waste with a traditional fuel as coal-dust or a mixture of coal-dust and pet-coke. This practise has two main advantages: it allows not-renewable fossil fuels saving, due to the integration of waste fuels and it permits to deploy waste in yet-existing plants, which should lead to a cost saving in plants construction.[1],[3]

1.4 MANAGEMENT AND PRODUCTION OF MUNICIPAL SOLID WASTE IN THE WORLD

Existent situations in Europe, U.S.A. and Japan are analysed in this chapter, because detailed information are available for these countries. In a later stage, a focus on Italian situation is considered.

However, the comparative analysis among data results to be difficult due to the existing differences on the MSW (Municipal Solid Waste) definitions among countries. Moreover, within the same country, the same definition also change during time, making the analysis more inaccurate. Consequently, the chapter aims only to show broad results about a comparison among states.

1.4.1. Quantitative data

As first point, it is important to understand the volumes of waste produced per year by every analysed country. The **Table 1. 2** shows absolute and per capita values of waste collected per year.

	Europe	**U.S.A.**	**Japan**
Year	2011	2012	2012
MSW Absolute Value [10^6t]	250.90	250.89	45.22
Population [billion]	506.8	314.1	127.6
MSW per capita Value [kg/citizen]	495.07	798.76	354.39

Table 1. 2 - Overall amount of waste produced per year [12],[13],[14]

Europe is intended as the union of the 28 member states of the European Union, which are (in order of overall population): Germany, France, United Kingdom, Italy, Spain, Poland, Romania, Netherlands, Greece, Belgium, Portugal, Czech Republic, Hungary, Sweden, Austria, Bulgaria, Denmark, Slovakia, Finland, Ireland, Croatia, Lithuania, Latvia, Slovenia, Estonia, Cyprus, Luxembourg, Malta.

Figure 1. 5 compares the temporal evolution of MSW production per capita in Europe, U.S.A. and Japan. Predictably, it follows the national economy trend: basing on the general assumption that the quantity of MSW produced increases with material wealth of population, high economic growth phases are usually associated with sudden increments in MSW production, while in crisis periods, quantity generated generally decrease. Economic state, in

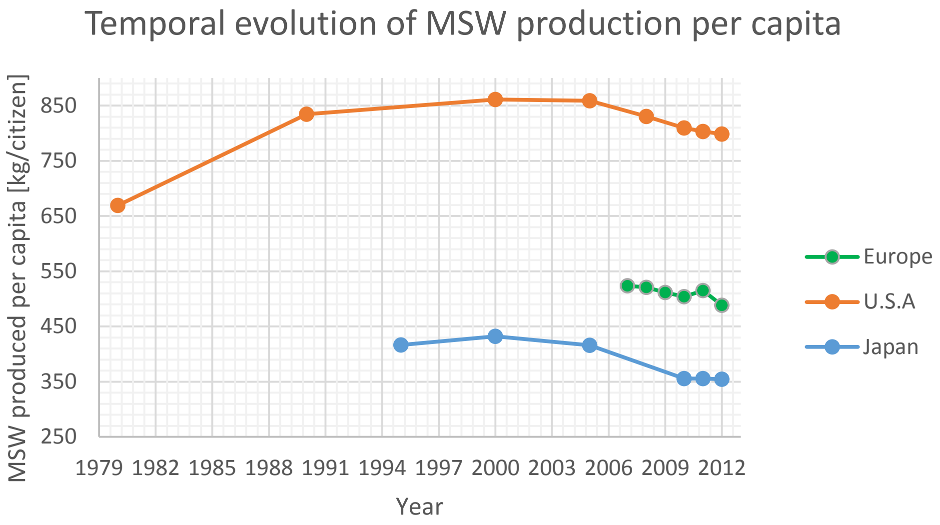


Figure 1. 5 - Temporal evolution of MSW production per capita in Europe, U.S.A. and Japan [12,[13],[14]

fact, is an index of financial possibilities of population and of its inclination to buy material goods and of its frequency to discard and replace goods.

Data in this and further cases in the paragraph are based on diverse sources, so the dissimilarities in waste classification among different countries could lead to misinterpretations of data showed in graphics.

While standard of living in the three countries analysed is roughly the same, the U.S.A. shows a great separation respect to Europe and Japan. This can be explained considering the different lifestyle to which consume parameters are related and the different perception and sensitivity to waste production and management.

1.4.2. MSW composition

Figure 1. 6 shows the municipal solid waste composition. Some MSW sub-categories differ for every different country analysed.

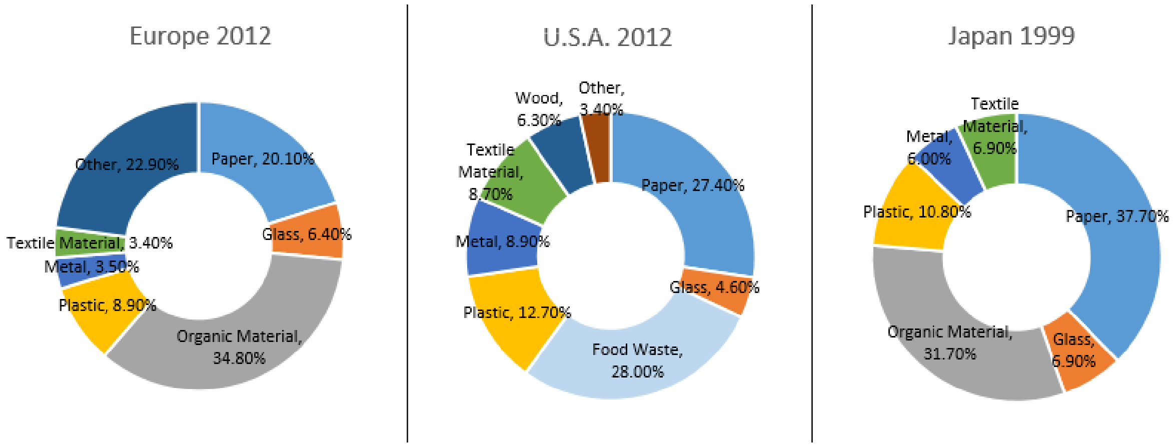


Figure 1. 6 - MSW mean composition in Europe, U.S.A. and Japan [1],[12],[13],[14]

The three diagrams show that in every country paper and organic materials constitute the majority of municipal solid waste collected, followed by plastic, textile material and metals.

1.4.1. Waste management politics

Solutions adopted by Europe, U.S.A. and Japan about waste management are evaluated.

Concerning **Europe**, the strategy is focused on waste management hierarchy. This politics has developed starting from the directive of 1998 “Waste Framework Directive” [22] that aimed to encourage reduction, re-use and recycle of waste good practices instead of deploy waste in landfills, which is seen as ultimate solution. This hierarchy was formally adopted in 1989 within “EU Community Strategy for Waste Management”[22] and it requires thus the adoption of hierarchical behaviour explained in “Four R” principle.

It is important to underline that this strategy is not binding for member states because its ultimate objective is to encourage a correct politics of waste management, which gives the best overall environmental outcome. As a result, not every member state has implemented the hierarchy suggested by the EU: it is, in fact, intuitive that higher the wellness of a country, higher the financial resources it can invest in waste disposal solutions that are technologically sophisticated and costly, as waste-to-energy plants. On the other hand, landfills constitute the cheaper solution, even though they are also the less favoured in the environment preservation viewpoint.

Figure 1. 7 shows data through years of European waste disposal, subdividing it in incineration, landfill and recycling (which also includes composting). The solution of landfills is gradually reducing, but represents even today more than one third of waste final disposal. On the other hand, recycling is increasing, as well as waste-to-energy, even though the latter is limited so far.

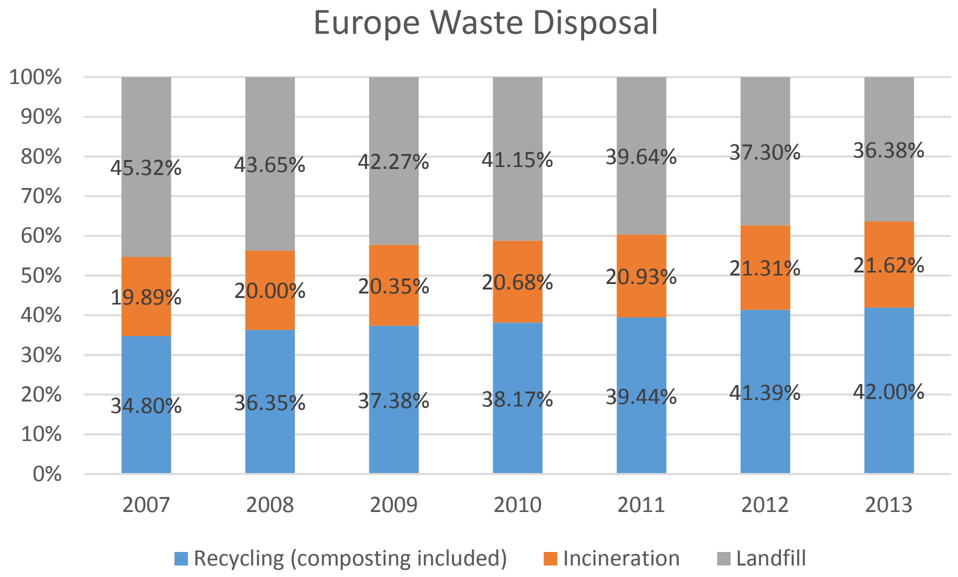


Figure 1. 7 - Temporal evolution of Europe waste disposal, divided in Recycling, Incineration and Landfill [13]

The **U.S.A.** are characterised by a low population density; consequently, they have less difficulty in finding lands to designate as landfills. This solution has, thus, represented and represents the most used method for waste disposal, as can be seen in the **Figure 1. 8** below. Nevertheless, regulations about safety and emissions control made management costs of landfills to increase. That is the reason why landfills number have reduced in recent years, increasing waste recycle and waste-to-energy plants. However, waste deploy in landfills represents still today the most used solution for MSW disposal.

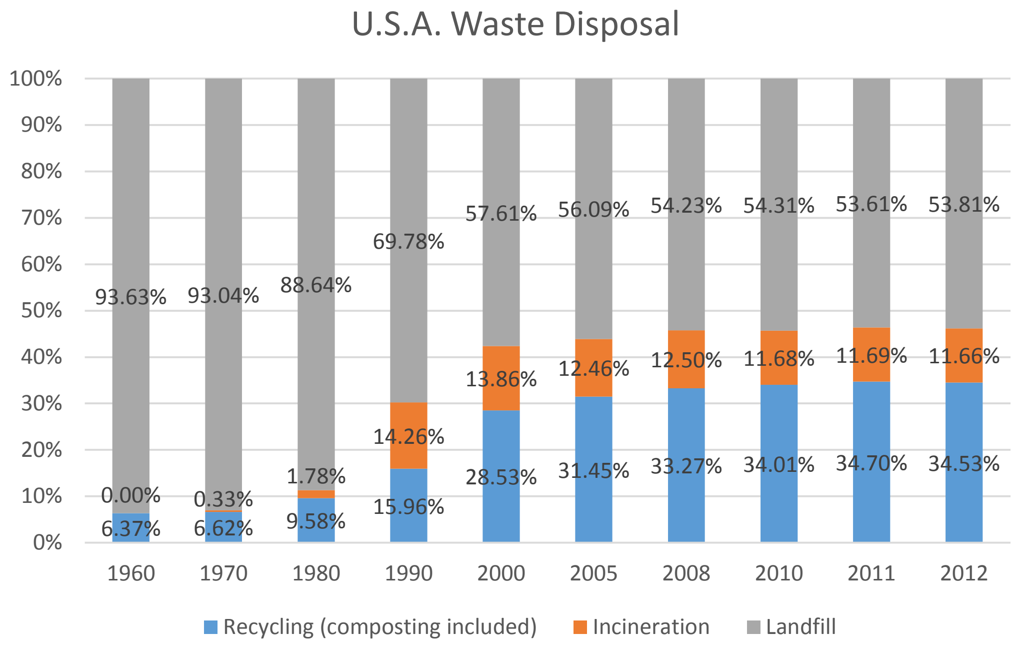


Figure 1. 8 - Temporal evolution of the U.S.A. waste disposal, divided in Recycling, Incineration and Landfill [12]

Japan has a mainly mountainous territory, so only a small part of it is available for residential use. That influences greatly solutions utilised for waste disposal, because increasing the number of citizen per square metre, the land cost increases, making waste deployment in landfills less desirable. Therefore, the country shows a great percentage of waste being recycled and an increasing percentage of it used as fuel in waste-to-energy plants, while landfills are progressively disappearing.

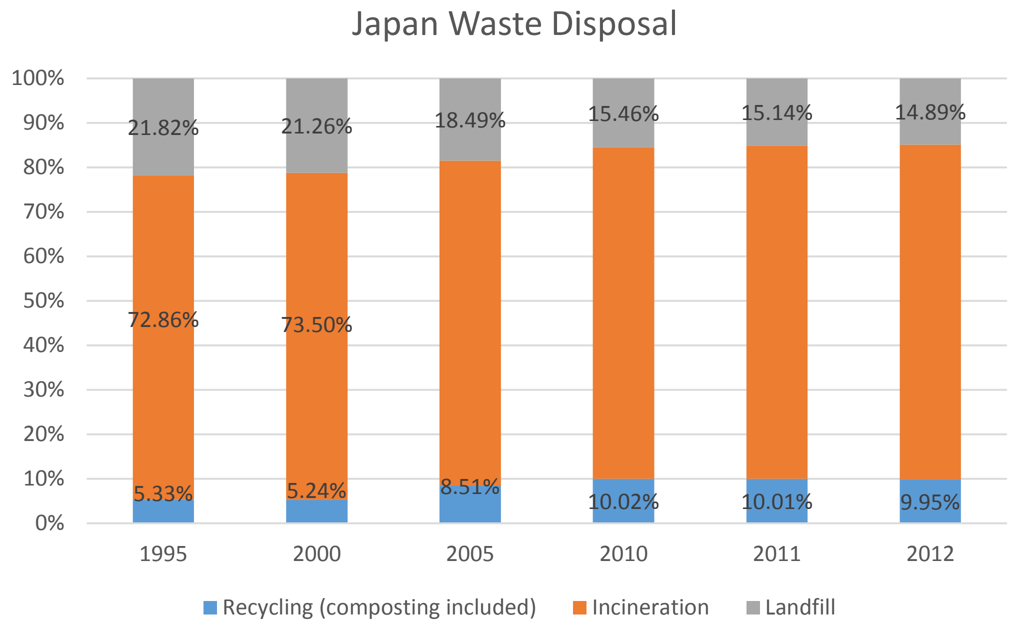


Figure 1. 9 - Temporal evolution of Japan waste disposal, divided in Recycling, Incineration and Landfill [14]

1.4.3. Focus on Italy

Analysing temporal evolution of municipal solid waste produced in Italy and comparing it with the one produced by Europe, it can be observed that Italy produces almost the same quantity of waste per capita than Europe, in all years analysed (**Figure 1. 10**).

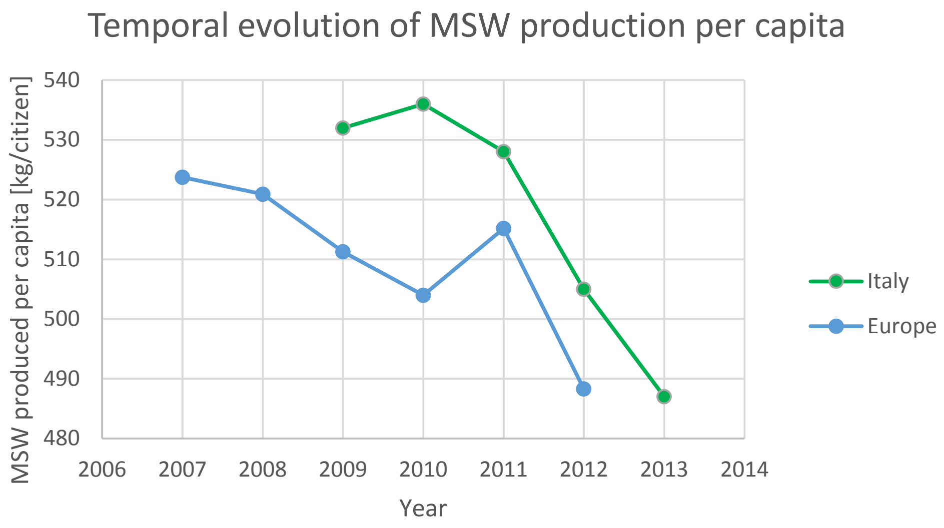


Figure 1. 10 - Temporal evolution of MSW production per capita in Italy [13],[16]

Moreover, waste composition can be analysed. On the contrary, in this case Italian waste composition is pretty close to composition of European municipal solid waste, differing mainly in organic material percentage and glass and plastic.

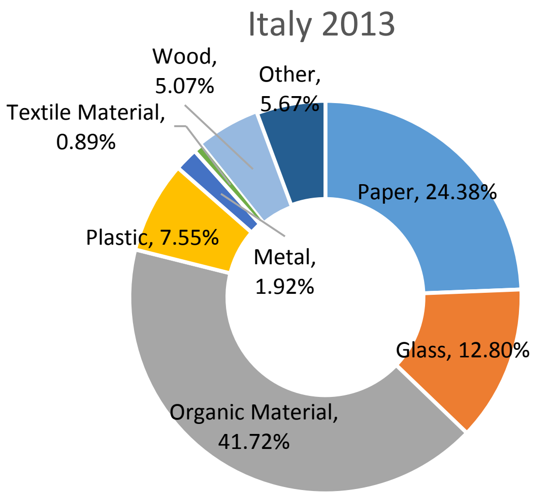


Figure 1. 11 - MSW mean composition in Italy [16]

Observing the temporal evolution of waste disposal, the presence of a majority quote of organic material respect to the European Union data is confirmed in **Figure 1. 11** above, where percentage of recycled municipal solid waste is major than in the EU. Italy is also progressively reducing the amount of waste destined to be deployed in landfills, increasing environmental quality of lands. The percentage of waste used as fuel in waste-to-energy plants is still low, as it is in the rest of Europe, confirming that in the continent, this technology has not yet affirmed and waste is still seen as something that creates negative externalities. On the other hand, more than 50% of waste dedicated to recycling activities confirms that the politics required by European Union has been accepted and applied in Italy with discreet success. The hope is to reduce for the further coming years the amount of waste deployed in landfills, even for the risk of organised crime to interfere in correct good administration of landfills, jeopardising the air, soil and water quality that a well-managed landfill should guarantee. Moreover, being Italy one of the countries affected by under-dimensioning in waste-to-energy plants theme, Italian waste is exported in other countries that can capitalise on it. Changing people and government perspective about waste is a challenge for the future, in order to reduce costs bounded to waste relocation (especially if it moves over states) and use waste as a precious resource in waste-to-energy plants.

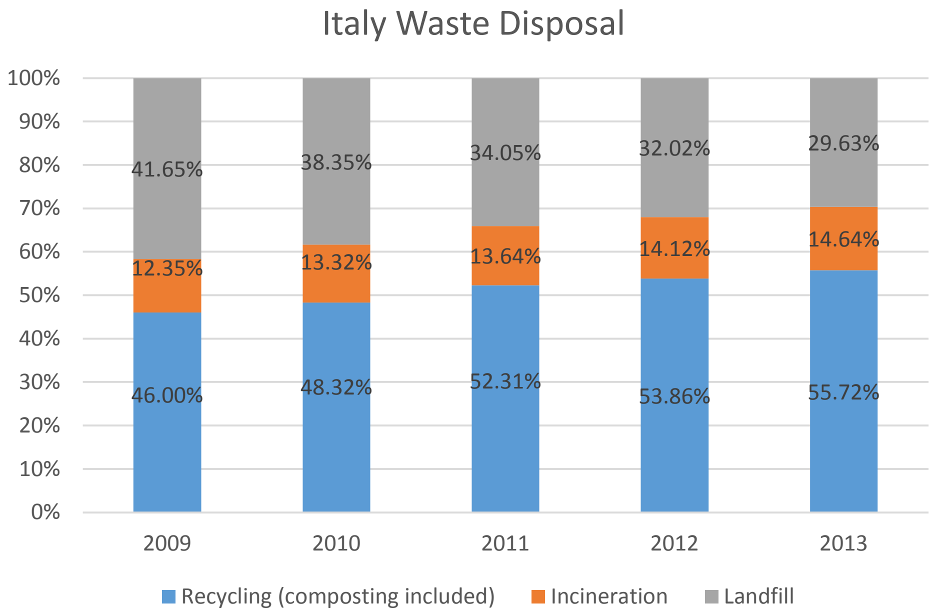


Figure 1. 12 - Temporal evolution of Italy waste disposal, divided in Recycling, Incineration and Landfill [16]

Chapter 2

State of the Art of Waste-to-Energy Plants

This chapter aims to analyse the elementary thermodynamic cycle that operates at the base of all thermoelectric plants for energy production. Techniques on how to increase a plant's overall efficiency acting on the Rankine cycle and some examples about existing plants would be presented furtherly in the chapter.

2.1.CARNOT CYCLE

A close elementary thermodynamic cycle is a finite succession of thermodynamic transformations (i.e. adiabatic transformations, isobars, isotherms, etc.) at the end of which the systems returns to its initial state. In thermodynamics, Carnot cycle is the simplest thermodynamic cycle, realised using only two energetic sources. Its name derives from the French physicist Sadi Carnot.

Carnot cycle is a purely theoretical cycle and its realisation requires the study of a theoretical thermal machine where a theoretical fluid realise a thermodynamic cycle. This statement confirms that realise a real thermal machine to which apply the Carnot cycle is impossible. The theoretical machine that execute the cycle is called Carnot machine.

Four reversible transformations constitutes the cycle. Two isotherms (1-2) and (3-4) to different temperature, respectively $T_h > T_l$ and two adiabatic curves (2-3) and (4-1):

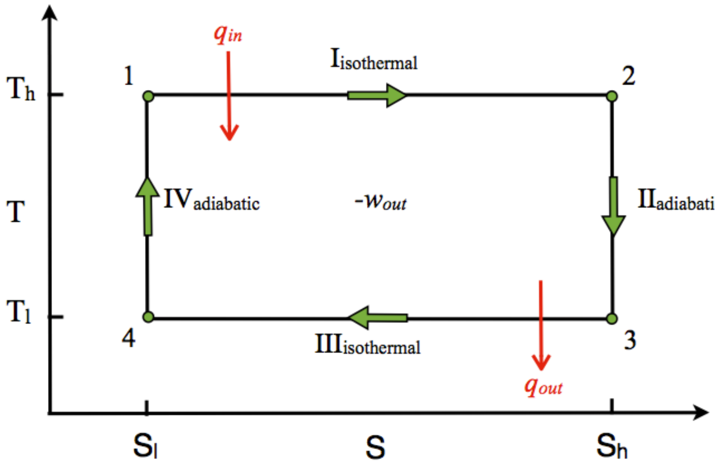


Figure 2. 1 - Theoretical Carnot cycle on T-s diagram

- **Reversible isothermal expansion (1-2).** The fluid extracts the heat q_{in} from the warmer source T_h, provoking the fluid volume increase. The temperature results to remain constant.

- **Reversible adiabatic expansion (2-3).** The fluid expands in order not to exchange energy with the environment, producing work. Consequently, the temperature decreases.

- **Reversible isothermal compression (3-4).** The fluid cedes the heat q_{out} to the source with the lowest temperature T_l, provoking the fluid volume decrease. The temperature results to remain constant.

- **Reversible adiabatic compression (4-1).** When the fluid finishes to cede its heat to the low-temperature source, it continues to compress but it does not exchange energy with the environment. Consequently, the temperature increases.

The outcome of this cycle demonstrates that, having an ideal Carnot machine, with a perfect fluid and two sources at different temperatures, it is possible to obtain work, bringing the system back to the initial conditions.

The fundamental characteristic of Carnot machine is that its efficiency does not depend on the fluid used in the cycle, but only on the temperatures of the sources with which it exchanges heat (more precisely, from the temperature ratio). This important theoretical thermodynamic result is named Carnot Theorem.

The efficiency of a thermal machine is, in general, the ratio between the useful work that the machine executes and the total heat absorbed by the system in order to accomplish that work. If a cycle is executed *n* times, the efficiency of the machine is:

$$\eta = \frac{|W|}{|Q_1|}$$

Where W indicated the total work accomplished by the machine and Q_1 the total heat absorbed by it. In the Carnot cycle case, the efficiency would be:

$$\eta = \frac{|q_{in} - q_{out}|}{|q_{in}|}$$

The heat exchange occurs only during the isothermal curves, so the efficiency only depends on the temperature T_1 and T_2.

$$\eta = \frac{|q_{in} - q_{out}|}{|q_{in}|} = \frac{T_h - T_l}{T_h}$$

The efficiency is maximum (100%) only when $T_l = 0$ K, temperature unreachable for whichever body. Consequently, the efficiency theoretically reachable for a Carnot cycle would ever be lower than one.

Carnot cycle is a thermodynamic cycle that evolves between two sources with the highest thermodynamic efficiency. No other cycle with the same isotherm as extreme temperatures as the Carnot cycle has the same or superior efficiency. For this reason, it is the reference cycle for real applications as, for example, Rankine cycles.

2.2. ELEMENTARY THERMODINAMIC CYCLE: RANKINE CYCLE

Whichever real cycle that want to achieve high efficiencies should approximate the Carnot cycle's transformations. However, the major part of heat exchanging occurs at constant pressure. In order to maintain even the temperature constant, it is necessary to resort to a phase change.

The cycle which interests thermoelectric plant is Rankine cycle, named after its inventor. Rankine cycle is not tied to one specific fluid, even though water, conveniently depurated, is the most common in thermoelectric applications. Other fluids are used in Rankine cycles; Organic Rankine Cycle (ORC), for example, uses organic liquid hydrocarbons with medium or low molecular weight, which has changing state temperature lower than water. This permits to use power sources that have lower temperatures, as heat collected by solar panels, but less energy can be extracted.

This thesis is concentrated on waste-to-energy plants, so Rankine cycle hereinafter mentioned would be considered as water Rankine cycle.

Four machines operates the four fundamental thermodynamic transformations of Rankine cycle:

- **Pump**. It extracts saturated liquid from condenser and injects it into the boiler. Enthalpy, entropy and temperature variations are negligible (about 30-35 °C while in the boiler and in the turbine the temperature gap is several hundred Celsius degrees), while pressure arises (starting from 10^{-2} bar to approximately 10^{2} bar). This is an operating machine, which means it needs external power to execute pressure raising. However, mechanical power absorbed by the device is usually negligible respect to the one produced by the turbine.

- **Boiler**. It is a heat exchanger between flue gas and sub-cooled liquid; in particular, gasses heat up water that passes through pipes within the boiler. Sub-cooled liquid is transformed thus, in saturated liquid, than in saturated vapour and later in superheated vapour. In particular, starting from the flame source, the boiler is crossed by the evaporator, which is constituted by tubes where water is vaporised, from saturated liquid to saturated vapour, super-heaters and re-heaters, where saturated vapour is superheated and economisers, where water is warmed up to the state of saturated liquid. Transformation is isobar (in other words water is warmed up at constant pressure), but enthalpy of superheated vapour increases due to temperature raise.

- **Turbine**. It is the machine where useful work is produced. Superheated vapour with high enthalpy and pressure enters in the turbine with the maximum temperature reached in the entire cycle and here it expands to the minimum pressure, which is the condensation pressure. Usually fluid exits the turbine as saturated vapour with the higher vapour quality possible (in order to avoid cavitation phenomenon within the turbine), with high specific volume and low temperature. Superheated vapour expansion makes turbine's blades rotate, producing mechanical power; turbine is, in turn, linked to an electric generator, which transforms mechanical power in electric power. The difference between expansion power and compression power (operated by the pump) is the net mechanical power.

- **Condenser**. It is a heat exchanger that condensates saturated vapour that exits the turbine, until its vapour quality becomes zero, transferring its heat to a thermic spring (which could be a lake, a river, the see, or, without a water source, in the atmosphere through a stack). The thermodynamic transformation is isobar and isotherm.

The underlying scheme illustrates the cycle and machines explained above. Arrows entering the cycle indicate work (W) or heat (Q) entering the system (work required by the pump and heat released to heat up the water within the boiler), while arrows exiting the cycle indicate work or heat exiting the system (work extracted by turbine body and heat released in saturated fluid condensation).

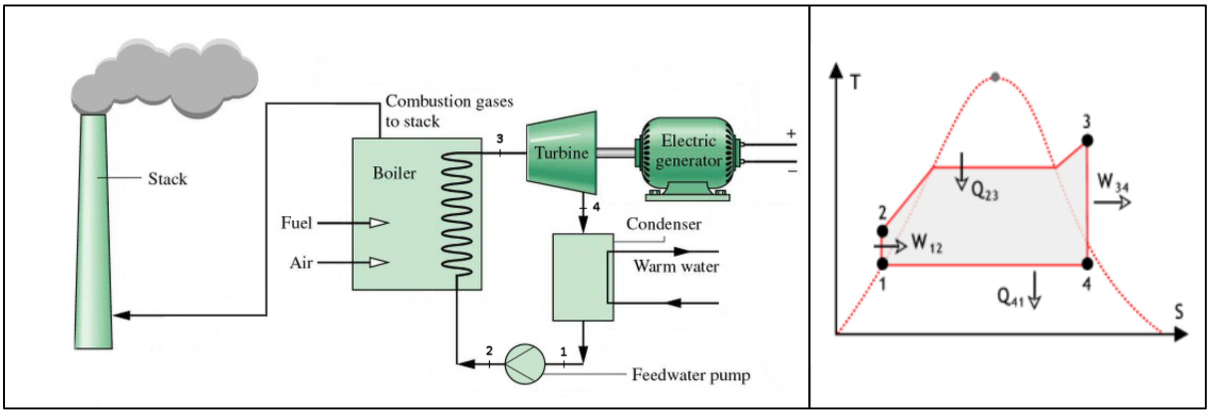


Figure 2. 2 – Rankine cycle scheme and T-s diagram

2.3. WAYS FOR RAISING THE OVERALL EFFICIENCY

The following paragraph aims to provide a brief recap about the elementary characteristics of a thermodynamic cycle.

Cycle efficiency is the term that mainly influences the overall plant efficiency, so it is necessary to evaluate in detail the influence exercised by principal thermodynamic parameters on cycle efficiency in order to arrange the necessary technical adaptations to improve it.

The entire treatise about calculations method and figures below are based on [23].

2.3.1. Decreasing condensation pressure

Consider two cycles *oabo* and *o'ab'o'* characterised respectively by condensation pressures p_k and $p_{k'}$, $p_k<p_{k'}$ (**Figure 2. 3**). Cycle *oabo* efficiency is:

$$\eta_{th} = \frac{W}{Q_1} = \frac{h_a - h_b}{h_a - h_o}$$

While cycle *o'ab'o'* efficiency results to be:

$$\eta'_{th} = \frac{W'}{Q_1{}'} = \frac{h_a - h_{b'}}{h_a - h_{o'}}$$

Indicating with W and Q_1 the useful work produced within the cycle and the heat necessary to obtain that work (that is the heat exchanged within the boiler). Terms with *h* indicate the enthalpy of the points indicated with letters in subscript.

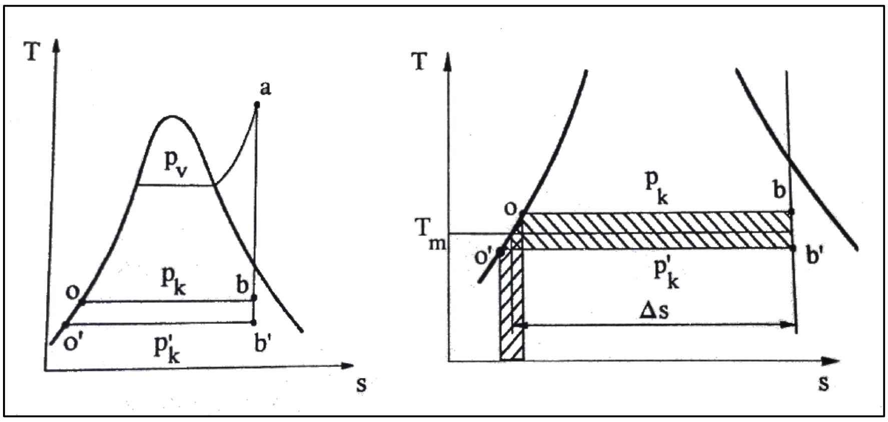


Figure 2. 3 - Cycles oabo *and* o'ab'o' *that show the differences between cycles with different condensation pressures [23]*

From the two previous equations is not possible to deduce which kind of variations the thermodynamic efficiency have undergone because both useful work and specific heat increase, moving from the first cycle to the second. Rewriting W' and Q'_1:

$$W' = W + \Delta W; \qquad Q'_1 = Q_1 + \Delta Q_1$$

The second cycle efficiency can be rewritten:

$$\eta'_{th} = \frac{W'}{Q_1{}'} = \frac{W + \Delta W}{Q_1 + \Delta Q_1} = \frac{W}{Q_1}\frac{1 + \frac{\Delta W}{W}}{1 + \frac{\Delta Q_1}{Q_1}} = \eta_{th}\frac{\left(1 + \frac{\Delta W}{W}\right)}{1 + \frac{\Delta Q_1}{Q_1}}$$

In order to obtain a higher efficiency of the second cycle respect to the first one, it must be:

$$\frac{1 + \frac{\Delta W}{W}}{1 + \frac{\Delta Q_1}{Q_1}} > 1 \quad \rightarrow \quad \frac{\Delta W}{\Delta Q_1} > \frac{W}{Q_1} = \eta_{th}$$

The inequality can be immediately be demonstrated looking at the **Figure 2. 3**above. ΔW is equal to the mixtilinear trapezoid area *o'obb'*, which in turn is equal to the area of the rectangle having ($T_o - T_{o'}$) as height and Δs as base (being Δs the entropy variation underwent by the fluid to transform to the saturated liquid state to liquid-vapour mixture with vapour quality x, at T_m temperature, equal to the arithmetic mean between the two temperatures T_o and $T_{o'}$):

$$\Delta W = \Delta s(T_o - T_{o'}) = \frac{rx}{T_m}(T_o - T_{o'})$$

Where r (≈ 2500 kJ/kg) is latent heat of vaporisation at T_m temperature and x represents the mixture vapour quality in point *b*. ΔQ_1 is also expressible as:

$$\Delta Q_1 = h_o - h_{o'} = c_p(T_o - T_{o'})$$

Being c_p the specific heat of water. Bringing together the latest two expressions:

$$\frac{\Delta W}{\Delta Q_1} = \frac{rx}{c_p T_m} \frac{T_o - T_{o'}}{T_o - T_{o'}} = \frac{rx}{c_p T_m}$$

Executing a numeric estimate, taking into account usual values of condensation pressures (p_k ≈ 0.05bar, which corresponds to a temperature T_m ≈ 300K) and of vapour quality x (0.8 – 0.9), the following result is obtained:

$$\frac{\Delta W}{\Delta Q_1} = \frac{rx}{c_p T_m} = \frac{2500 * 0.8}{4.187 * 300} > 1$$

The ratio ΔW/ΔQ1 is thus greater than one and is consequently greater of whichever possible efficiency value η_{th} of the starting cycle. This last estimate consents to affirm that a reduction in condensation pressure determines a thermodynamic efficiency increase.

The inferior limit of the pressure is bounded to condensation temperature of the vapour, which in turn is bounded to the temperature of cooling water in the condenser: in Nordic countries, where water's annual mean temperatures are lower than 20°C, condensation pressures can be lower than 0.04bar. On the other side, in countries characterised by hot climate, the inferior limit of condensation pressure is higher because the cooling water is available at higher temperatures. In Italy, cooling water is collected on average at (20-22)°C (variable through the year) and released at circa 30°C, consequently the usual condensation pressures are around 0.05bar.

Vapour condensation requires huge cooling water flow rates, because the law for thermal difference ΔT, underwent by cooling water during condenser crossing, must be lower than 8°C, as can be deducted in the following equation for condenser thermic balance:

$$(h_b - h_o)\dot{m}_v = \dot{m}_{H_2O}\, c_p\, \Delta T$$

This made the choosing of the establishment place for thermoelectric power stations problematic, which are preferably located near the see or near big streams.

2.3.2. Influence of vaporisation pressure

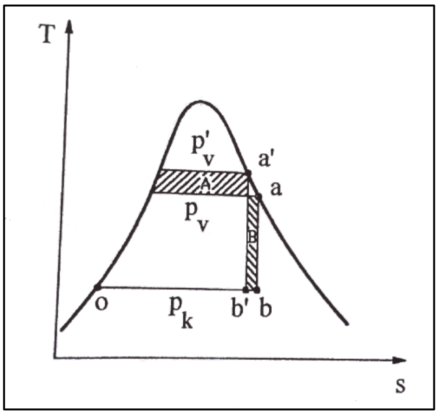


Figure 2. 4 - Cycles oabo *and* o'ab'o *that show the differences between cycles with different vaporisation pressure [23]*

Consider two Rankine cycles *oabo* and *oa'b'o*, characterised by the same condensation pressure values and by different vaporisation pressure values, respectively equal to p_v and $p_{v'}$, while $p_v<p_{v'}$ (**Figure 2. 4**). Defining the thermodynamic efficiency as follows:

$$\eta_{th} = \frac{W}{Q_1} = \frac{W}{W + Q_2} = \frac{1}{1 + \frac{Q_2}{W}}$$

Naming Q_2 the heat required in order to condense the fluid exiting the turbine, it is demonstrable that with low boiler pressures, an increase in vaporisation pressure determines an increase in thermodynamic efficiency, because work increases (A area is greater than B area) and discharged heat decreases (area under the segment *ob'* is less than the one under *ob*). On the other hand, with high boiler pressures, close to critical pressure, an increasing in vaporisation pressure causes a decrease in cycle efficiency. In fact, work decreases more than discharged heat (A area becomes smaller than B area, while discharged heat continues to decrease).

Superheating is usually used for high-pressure cycles (which is the case study of the present thesis).

2.3.3. Introducing superheating process

Superheating determines a double benefit: vapour quality increase in the final expansion phase and cycle efficiency increase. Concerning the first aspect, turbine blades are designed to work with a single-phase fluid: the presence of water in high percentage provokes a decrease in the turbine fluid-dynamic efficiency. The positive thermodynamic effect derived from vapour superheating is easily comprehensible confronting the saturated vapour cycle *ocdo* and the superheated vapour cycle *ocabdo*, operating between the same isobar curves (**Figure 2. 5**).

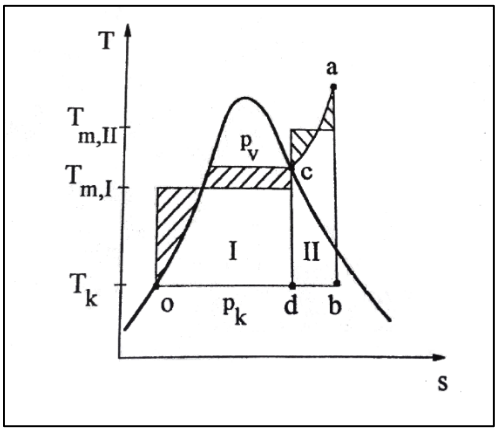


Figure 2. 5 - Cycles ocdo *and* ocabdo *show the differences between cycles with or without superheating [23]*

The latter is the merging between the saturated vapour cycle (I Cycle) which efficiency η_I and cycle *cabdc* (II Cycle) with efficiency η_{II}.

The efficiency of superheated vapour cycle η can be calculated as weighted average of efficiencies η_I and η_{II}, assuming as weights the heat quantities entering the two partial cycles *ocdo* and *cabdc*:

$$\eta = \frac{\mathrm{W_{(I+II)}}}{\mathrm{Q_{1,(I+II)}}} = \frac{\mathrm{W_I + W_{II}}}{\mathrm{Q_{1,I} + Q_{1,II}}} = \frac{\mathrm{\eta_I Q_{1,I} + \eta_{II} Q_{1,II}}}{\mathrm{Q_{1,I} + Q_{1,II}}} = \frac{\mathrm{\eta_I(h_c - h_o) + \eta_{II}(h_a - h_c)}}{\mathrm{(h_c - h_o) + (h_a - h_c)}}$$

In order to have $\eta > \eta_I$, must be $\eta_{II} > \eta_I$. This condition is always true. In fact, substituting I Cycle and II Cycle with two equivalent Carnot cycles, obtained by tracing two ordinates at temperatures $T_{m,I}$ and $T_{m,II}$:

$$\mathrm{\eta_I = 1 - \frac{T_K}{T_{m,I}}}; \qquad \mathrm{\eta_{II} = 1 - \frac{T_K}{T_{m,II}}}$$

Where T_K represents the condensation temperature. $T_{m,II}$ is always greater than $T_{m,I}$, so $\eta_{II} > \eta_I$ and consequently the efficiency η of the cycle with superheating is higher than the efficiency η_I of the cycle without superheating, included between the same isobar lines.

In conclusion, superheating always carries a thermodynamic advantage. The more the superheating temperature T_a, the more the benefits. Materials mechanical resistance at high temperature with which super-heater tubes are realised imposes the limit value for the maximum superheating temperature. In Europe the superheating temperature reaches (540-570)°C while in Japan some plants can reach (630-640)°C. However, waste-to-energy plants are limited in maximum superheating temperature, because of the corrosive nature of flue gas, to a temperature of (420-450)°C.

2.3.4. Re-heat introduction

Re-heat is largely employed in thermoelectric plants. In fact, the problem of too low vapour quality of vapour at turbine exit (x<0.8) is present even if vaporisation pressure is high, even in superheating cases. As discussed above, presence of water drops in the vapour at turbine exit generates friction and erosion problems of turbine blades with consequent power and efficiency losses. In order to avoid this inconvenient, an intermediate vapour re-heat is usually introduced.

Superheated vapour in physical state *a*, expands within the high-pressure turbine until it reaches the intermediate pressure p_{rh} (re-heat pressure). At this point, it is brought back to the boiler, where it is re-heated until the temperature is almost equal to the superheating one. Re-heated vapour in physical state *a’* is sent again in the turbine, where it completes the expansion.

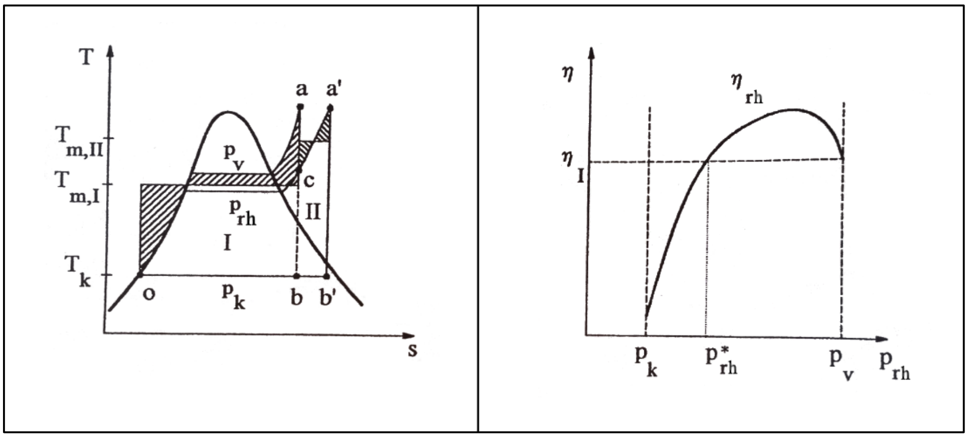


Figure 2. 6 - Cycles oabo *and* oa’b’o *show the differences between cycles with or without re-heat; the second diagram shows the efficiency trend, varying the re-heat pressure [23]*

An opportune choose of the re-heat pressure value would consent to obtain a higher efficiency than the one obtainable with the same cycle without re-heat. In fact, re-heat cycle efficiency η_{rh}, obtainable by the merge of superheating cycle *oabo* (I Cycle) and *ca’b’bc* (II Cycle), can be written as follows:

$$\eta_{rh} = \frac{\eta_I Q_{1,I} + \eta_{II} Q_{1,II}}{Q_{1,I} + Q_{1,II}} = \frac{\eta_I (h_a - h_o) + \eta_{II}(h_{a'} - h_c)}{(h_a - h_o) + (h_{a'} - h_c)}$$

To obtain $\eta_{rh} > \eta_I$, must be $\eta_{II} > \eta_I$. Substituting I Cycle and II Cycle with two equivalent Carnot cycles, obtained by tracing two ordinates at temperatures $T_{m,I}$ and $T_{m,II}$:

$$\eta_I = 1 - \frac{T_K}{T_{m,I}}; \qquad \eta_{II} = 1 - \frac{T_K}{T_{m,II}}$$

As explained in the superheating case, to obtain $\eta_{II} > \eta_I$, $T_{m,II}$ must be greater than $T_{m,I}$. That is not always true: it depends on the value of re-heat intermediate pressure p_{rh}. In fact, $T_{m,II}$ is minor than $T_{m,I}$ when p_{rh} is near the condensation pressure p_K. $T_{m,II}$ becomes higher than $T_{m,I}$ only when p_{rh} exceeds a specific value (p_{rh}* in **Figure 2. 6**) and consequently $\eta_{rh} > \eta_I$. Moreover, **Figure 2. 6** shows that η_{rh} surpass η_I only within the interval of pressures included between p_{rh}* and p_v. In particular, it demonstrates that η_{rh} shows a maximum for the value of p_{rh} for which $T_{m,I} \approx T_c$.

In industrial applications, only one re-heat is adopted because of the remarkable plant complications. In waste-to-energy plants, due to the corrosive nature of flue gas, re-heat is usually not adopted in order not to redouble the critical boiler sections (which normally is the superheating section).

2.3.5. Introduction of bleeding cycles (with regenerators)

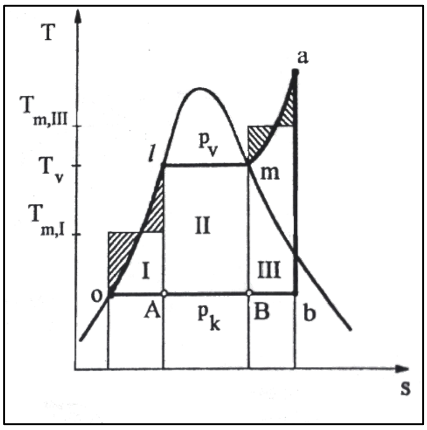


Figure 2. 7 - Superheating cycle divided in three sub-cycles [23]

Observing the transformations that compose the cycle above, the area under the curve *ol* represents the heat necessary to warm up the liquid, while areas under the curve *lm* and *ma* represent, respectively, the vaporisation heat and the heat supplied during the superheating phase.

The cycle *olmabo* can be thought as the merge of three cycles: *olAo* (I Cycle), *AlmB* (II Cycle) and *Bmab* (III Cycle). The efficiency η of the cycle *olmabo* can be calculated, as seen before, as weighted average of the three cycles efficiencies η_I, η_{II}, η_{III} of which it is composed:

$$\eta = \frac{\eta_I Q_{1,I} + \eta_{II} Q_{1,II} + \eta_{III} Q_{1,III}}{Q_{1,I} + Q_{1,II} + Q_{1,III}}$$

II Cycle is a Carnot cycle working between the temperatures T_V (vaporisation temperature) and T_K (condensation temperature), so the efficiency value of η_{II} is the maximum possible:

$$\eta_{II} = 1 - \frac{T_K}{T_V}$$

Substituting III Cycle with its equivalent Carnot cycle, obtained by tracing the ordinate at temperature $T_{m,III}$:

$$\eta_{III} = 1 - \frac{T_K}{T_{m,III}}$$

The III Cycle efficiency is higher than the II Cycle's because $T_{m,III} > T_V$. In fact, as just discussed in sub-paragraph 2.3.2, superheating increases the saturated vapour cycle efficiency (II Cycle).

The I Cycle is now considered, where *ol* supplies heat to the fluid in order to warm it up to T_V. Substituting I Cycle with the equivalent Carnot cycle:

$$\eta_I = 1 - \frac{T_K}{T_{m,I}}$$

Since $T_{m,I}$ is lower than T_V, η_I is minor than η_{II}, which in turn is lower than η_{III}. In conclusion, the efficiency of the I Cycle affects the overall efficiency in a negative way the more the heat supplied to warm the fluid up to the vaporisation temperature is. It is possible to limit the negative effect of preheat, adopting bleedings cycles (or cycles with regenerators): during expansion into the turbine, a fraction of vapour is spilled and sent to a pre-heater *R*. Here the spilled vapour mass condensates, transferring heat to the liquid coming from the condenser: although the bleeding introduction reduces the power available at the shaft (spilled vapour flow rate doesn't completely expand to condensation pressure), the plant overall efficiency increases.

In fact, referring to the scheme in **Figure 2. 8**, (1+m) kg of vapour enters the turbine, of which m kg is spilled at an intermediate pressure p_{sp}, included between p_V and p_K (physical state *l*). The remaining 1kg of vapour completes the expansion until reaches p_K and later it condenses. In the pre-heater *R*, 1kg of water obtained by the precedent condensation comes in contact with m kg of vapour spilled from the turbine. The latter transfers its heat to the water, condensing. (1+m)kg of water exits the pre-heater *R* in the physical state *f*.

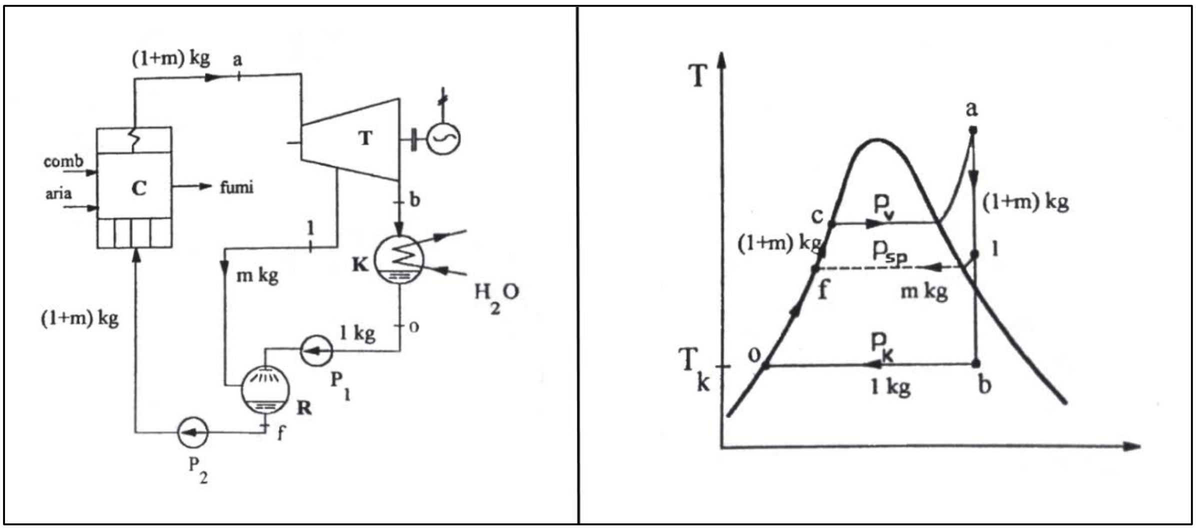


Figure 2. 8 - Bleeding cycle [23]

The thermal balance of the pre-heater *R* can be written, indicating with *o* the physical state of the liquid exiting the condenser and indicating with *m* the vapour mass spilled per fluid mass unit that comes to the condenser:

$$\mathrm{m(h_l - h_f) = 1(h_f - h_o)}$$

The first member represents the heat transferred from m kg of vapour that condenses (transformation *l* → *f*). From the previous equations, the quantity of vapour that must be spilled per kg of vapour that arrives at the condenser *K* can be calculated:

$$\mathrm{m = 1\frac{(h_f - h_o)}{(h_l - h_f)}}$$

The heat Q_1 that has to be given to (1+m) kg to bring it from the physical state *f* to superheated vapour *a* is:

$$\mathrm{Q_1 = (1 + m)(h_a - h_f)}$$

The presence of a bleeding determines a decrease both of the heat quantity Q_1 supplied from outside to the fluid (without bleeding it would have been: $Q_1 = (1+m)(h_a-h_o)$), and of the work available at the shaft (only 1kg of vapour expands completely). The presence of a bleeding produces a thermodynamic benefit as a whole, as it would be demonstrated. Naming η_0 the efficiency of the cycle without bleeding:

$$\eta_0 = \frac{W}{Q_1} = \frac{(h_a - h_b)(1+m)}{(h_a - h_o)(1+m)} = \frac{(h_a - h_b)}{(h_a - h_o)}$$

The efficiency η_1 of the cycle with one bleeding is:

$$\eta_1 = \frac{W_1}{Q_{1,1}} = \frac{W_1}{W_1 + Q_{2,1}}$$

$Q_{2,1}$ represents the condensation heat of 1kg of vapour , while the work W_1 is the sum of the one obtained by (1+m)kg of vapour in the expansion between p_V and p_{sp} (enthalpy interval (h_a-h_l)) and the one produced by the expansion of 1kg of vapour from p_{sp} to p_K (enthalpy interval (h_l-h_b)). Thus, the latter equation can be rewritten:

$$\eta_1 = \frac{(1+m)(h_a - h_l) + 1(h_l - h_b)}{(1+m)(h_a - h_l) + 1(h_l - h_b) + 1(h_b - h_o)} = \frac{(1+m)(h_a - h_l) + 1(h_l - h_b)}{(1+m)(h_a - h_l) + 1(h_l - h_o)} = \frac{m(h_a - h_l) + (h_a - h_b)}{m(h_a - h_l) + 1(h_a - h_o)}$$

The bleeding cycle efficiency is higher than the cycle’s without bleeding, because the expression of η_1 is the same as η_0 a part from the quantity $m(h_a-h_l)$, which is summed both to numerator and denominator. This thermodynamic benefit is possible due to the fact that the heat transferred during the pre-heat phase, from the outside to (1+m) kg of liquid fluid is given to higher temperature in the bleeding cycle than the cycle without bleeding.

Figure 2. 9 shows the variation percentage of the efficiency trend, with one bleeding, as a function of the regeneration grade *r*, defined as follows:

$$r = \frac{h_f - h_o}{h_c - h_o}$$

The maximum thermodynamic benefit is obtained in correspondence to the regeneration grade value r=0.5.

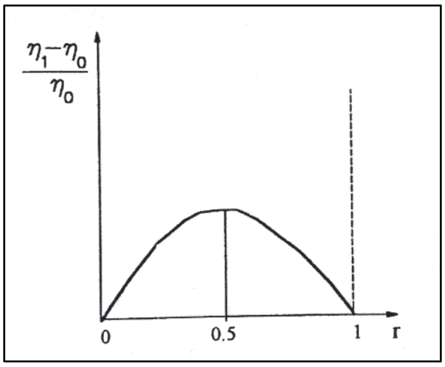


Figure 2. 9 - Trend of variation percentage of the efficiency of the cycle with one bleeding as a function of the regeneration grade [23]

The efficiency increases further in case of two bleedings or more than two. The efficiency expression with n bleedings is:

$$\eta_n = \frac{(h_a - h_b) + \sum_{i=1}^{n} m_i (h_a - h_i)}{(h_a - h_o) + \sum_{i=1}^{n} m_i (h_a - h_i)}$$

Where m_i is the mass of vapour spilled per unit mass that enters the condenser. From the thermodynamic viewpoint, the cycle efficiency increases when the bleedings number increases, because temperature gaps between condensed vapour and water decreases and, consequently, even entropy increments decreases.

However, a maximum number of bleedings exists. More bleedings, in fact, mean more costs bounded to plants complexity, so, at a certain point, the thermodynamic benefit obtained from adding another bleeding to the cycle does not justify the expense. In waste-to-energy plants, regenerators (or pre-heaters) are not present or they are only in a small amount in order to limit costs and limit the water temperature at the entrance of the boiler.

2.3.6. Conclusions

In conclusion, a traditional plant configuration for thermoelectric energy production is shown in the **Figure 2. 10** below: fuel and air enter the boiler, where the combustion takes place. Flue gas go through the boiler body, exchanging heat with the saturated water in the evaporator, superheated vapour in super-heaters and re-heaters and sub-cooled water in the economiser. Finally, it transfers its heat to the air entering the boiler (Ljungstrom), in order to increase the combustion efficiency. Then it passes through a filtering system to eliminate the most dangerous combustion products and finally is discharged in the atmosphere through the stack (not represented in the present scheme). The vapour, after expanding in the turbine body, condenses in the condenser and enters the deaerator, which is a heat exchanger which aim is to eliminate a good part of gasses from the water, warming it up with a small amount of superheated vapour spilled from the turbine. Water exists the deaerator *DE* as saturated water and enters the regenerator R, where it exchange heat with another part of superheated vapour (with higher temperature than the one that enters the deaerator because water is warmer in *R* than in *DE* because it has already been warmed up in the deaerator).

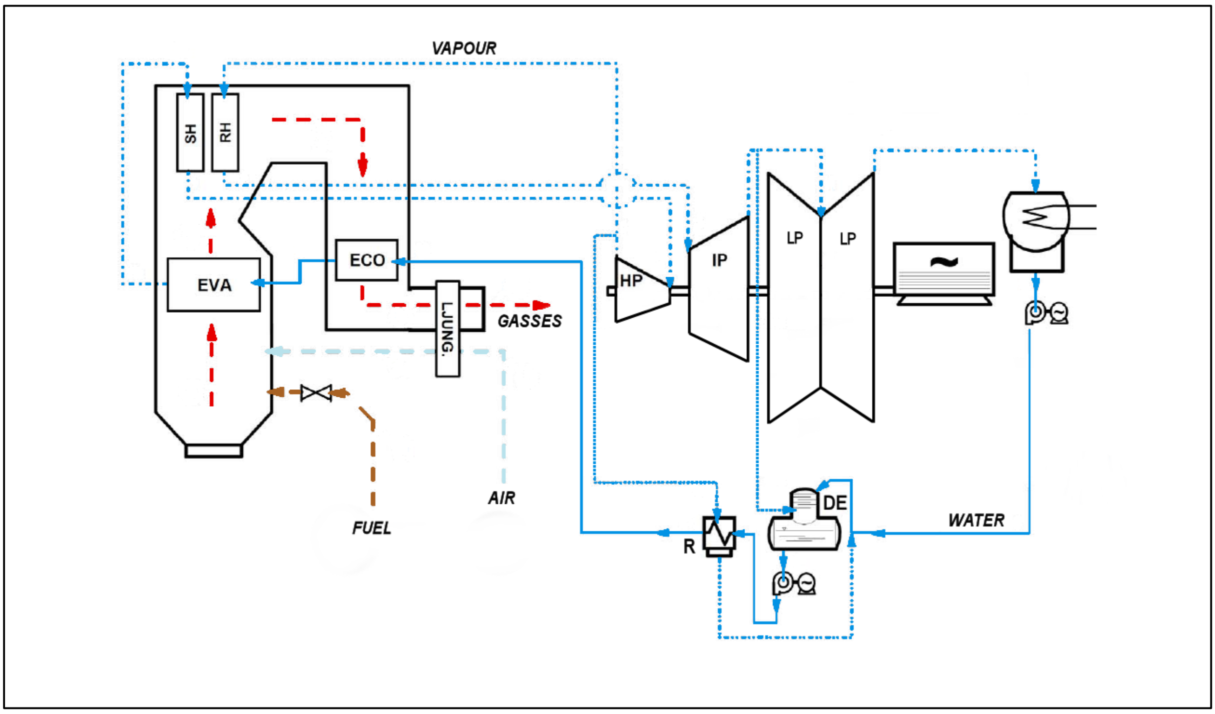


Figure 2. 10 - Traditional plant configuration for thermoelectric energy production

As discussed in the paragraphs above, waste-to-energy plants are slightly different from traditional ones.

In fact, in addition to what has been clarified before in the previous paragraphs, the maximum plant pressure is 65-70 bar, to avoid excessive liquid formation within the turbine and to limit the evaporator tubes temperature within the combustor. Ljungstrom pre-heater is usually not present in waste-to-energy plants because of the high percentage of ash in flue gas, which would provoke its blockage in a relative short time. Finally, because the plant is of small size due to the fuel nature, turbine efficiency is low.

Examples of waste to energy plants in the world would be presented in the following paragraph.

2.4. EXAMPLES OF WASTE-TO-ENERGY PLANTS

The aim of this paragraph is to present some case studies of waste-to-energy plants existent all around the world in order to analyse the state of the art before presenting the innovative plant configuration that is the main content of this thesis.

The plants presented in the next sub-paragraphs aims only to provide brief examples about the rough principal data about waste-to-energy plants in Europe, without expecting precise values, but only aiming to offer a general idea about the plants characteristics.

2.4.1. Turin waste-to-energy plant [24]

Turin waste-to-energy plant burns undifferentiated waste recovering energy contained in waste, producing electricity and thermal energy (hot water) for district heating. The plant can operate in electric mode or in cogenerating layout: in the first case, it produces energy to satisfy annual needs of about 175,000 families composed by three people. In the second case, it generates every year the thermal energy that can warm up 17,000 houses of 100m^2 and the electricity consumed by 160,000 families. The **Table 2. 1** below shows the general parameters of the plant:

Waste burned in a year	Electric energy produced in electric mode	Thermal energy produced in cogenerating mode	Electric energy produced in cogenerating mode	Traditional fuel saved in a year
[t]	**[MWh]**	**[MWh]**	**[MWh]**	**[t]**
421,000	350,000	170,000	320,000	70,000

Number of lines	Mean lower heating value of waste	Nominal thermal load	Total vapour produced	Vapour pressure	Vapour temperature
[-]	**[kJ/kg]**	**[MW]**	**[t/h]**	**[bar]**	**[°C]**
3	11,000	206	220	60	420

Table 2. 1 - Turin waste-to-energy plant general data

Moreover, it is possible to analyse the plant configuration, as shown in the **Figure 2. 11** below. Waste trucks access the plant passing through the radioactivity control portal. Then they are weighted and registered. Afterwards they discharge the waste in the pit. Two buckets extract the waste and deposit it on three hoppers, which transport the waste into the boiler. Waste remains on the mobile grids of the waste combustor for about an hour, where it burns at about 1000°C. Gasses generated by the combustion go up and enter the boiler conduits, which are above each grid. Every boiler contains vertical tubes (heat exchangers) where water circulates and, heating up thanks to the heat provided by the gasses, becomes water vapour. The pressure of the vapour activates the turbine, linked to an alternator, and produces electric energy. In cogenerating mode, part of the vapour passing through the turbine is spilled and send to heat up the water of the district heating grid. The combustion remains constitute the slags, which are collected on the bottom of the grid and cooled in water basins. Afterwards, they are moved in a special areas. Gasses are depurated before the emission in the atmosphere, passing through the flue gas treatment system. Then, the gasses enter the three stack pipes, aspired by a fan, and are expelled at about 120°C at a height of 120m.

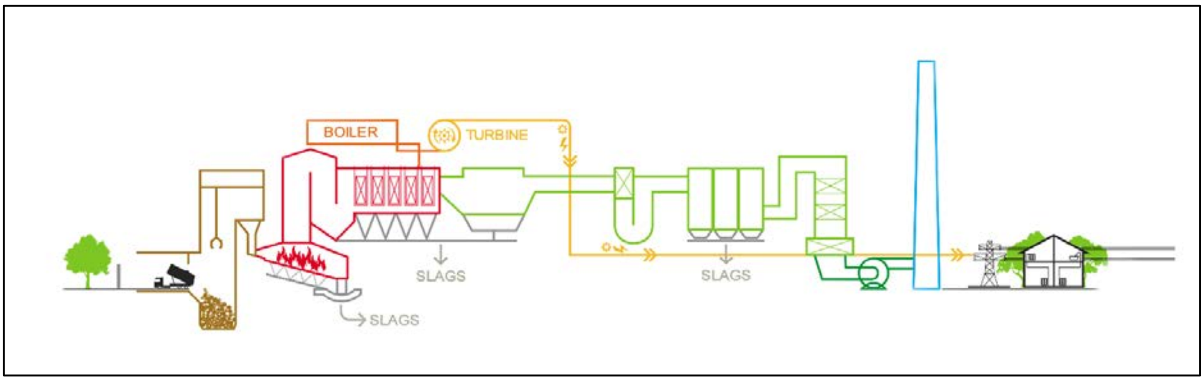


Figure 2. 11 - Turin waste-to-energy plant scheme

2.4.2. Paris waste-to-energy plants [25]

Paris has three waste-to-energy plants: the Issy-les-Moulineaux waste-to-energy plant came into operation in 2007 and treats 460,000 tonnes of residual waste per year. It has been built partially underground and the chimney has been designed in such a way that no large stack is visible on the Paris skyline. The Saint-Ouen plant, however, is in operation since 1990 and treats about 600,000 tonnes of household waste. Finally, the Ivry-Paris XIII plant was built in 1969, has been regularly modernised over the years and currently has a treatment capacity of 700,000 tonnes of household waste a year. The three plants heat about 1/3 of Paris' buildings, circa 245,000 home equivalents are heated by the three w-t-e plants. The **Table 2. 2** below shows the general parameters of the plants:

Waste burned in a year	Thermal energy produced in cogenerating mode	Electric energy produced in cogenerating mode	Traditional fuel saved in a year
[t]	[MWh]	[MWh]	[TOE]
1,760,000	2,400,000	200,000	300,000

Table 2. 2 - Paris waste-to-energy plant general data

2.4.3. Vienna waste-to-energy plant [26]

Vienna's three waste incineration plants, Flötzer-Steig, Spittelau and Pfaffenau, as well as the fluidised bed furnace 4 are used for the thermal treatment of residual waste, bulky waste and other combustible mixed waste collected in the city. In all, these facilities offer a treatment capacity of approximatively 780,000 tonnes per year, which corresponds to an average weekly treatment capacity of roughly 16,000 tonnes. The **Table 2. 3** below shows the general parameters of the plants:

Waste burned in a year	Thermal energy produced in cogenerating mode	Electric energy produced in cogenerating mode
[t]	[MWh]	[MWh]
780,000	1,200,000	81,000

Table 2. 3 - Vienna waste-to-energy plant general data

Chapter 3

Problems and possible solutions in WtE plants

This chapter aims to analyse the problems of a waste-to-energy plant and possible solutions than could be implemented in the new configuration plant. Examples on how the world tried to address the main problems are explained further in the chapter.

3.1.THE PROBLEM

As touched on in previous paragraphs, while waste incineration grants hygiene, great volume reduction (about 90%) and considerable mass reduction (about 80%), waste-to-energy also includes the recovery of a part of the energetic content of waste. From the energetic viewpoint, waste can be assimilated to solid fuel, even though its quality is poor. In fact, it has:

- **Modest lower heating value (LHV)**. Confronting a waste-to-energy plant with a traditional plant with equal power produced, the first has greater flow rate (which implies greater auxiliaries power consumes) and greater dimensions (which means greater investment costs).

- **Unfavourable physical properties**. The solid nature of waste requires the presence of the grate within the furnace and long residence times, which leads to the need of large combustion chambers.

- **Elements content that could transform in toxic and corrosive mixtures (i.e. Cl, F, Br, S, metals, etc.)**. That causes a higher environmental impact and lower performances than the ones obtainable with fossil fuels. These waste criticalities make the adoption of high vapour parameters or normal plant’s sophistications extremely onerous. Moreover, the high percentage of pollutant production requires a heavy flue gas treatment, increasing the costs.

- **Unverifiable composition**. Waste treatment plants must have great flexibility in order to cope with the variability of waste composition and dimensions. This, obviously, leads to a greater plant complexity, which in turn leads to higher investment and designing costs.

- **Plant size lower than the traditional plant’s typical one**. Maximum some tens of electric megawatt respect to some hundreds of traditional plant’s. Modest efficiencies due to the small size of the turbine body, which undergoes the scale effect. That leads, in turn, to a cost scale effect, making uneconomical the plant solutions adopted in big size plant.

Because of extremely high capital costs and low efficiency, energy production from waste is economically acceptable only if fuel price is negative. However, reliability than efficiency of waste-to-energy plants is privileged, because waste exists and must be disposed of, despite the costs.[6]

3.1.1. Corrosive mixture formation

While other problems highlighted in the previous paragraph are characteristics of waste fuel, the corrosive mixture formation is a problem related to waste combustion and not to its nature. Hereinafter, an in-depth analysis In order to better understand the mechanism that generates corrosion problems. The report “Metallic materials behaviour to corrosion in steam generators in waste-to-energy plants” published on 31.12.2000 by CESI [27] presents the results of corrosion tests on metallic materials in simulated waste-to-energy combustion atmosphere. Waste combustors are subjected to high temperature corrosion due to the presence of chlorinated mixtures, derived from chlorine presents within waste and molten salts, with low melting points, which deposits on heat exchanger’s tubes during combustion. This corrosive phenomenon severely reduces plants efficiency because it causes frequent non-programmed plant stops for damaged components substitution. These stops penalise waste-to-energy plants managements due to the lacking waste disposal and interruption of electric energy production.

The corrosion test has been conducted to a vapour temperature superior of about 30°C to the vapour produced in the more modern waste-to-energy plants (400-420°C), in order to take into account the temperature gradient through heat exchanger’s tubes walls. Materials samples, as parallelepiped (6x4x8mm) have been placed in quartz crucibles and covered in slags extracted from a tube wall of *AMSA MILANO 1* plant, in order to simulate combustion atmosphere. Ash analysis, operated by *Fuel Experimental Station*, is reported in the following table.

Elements	Unit of Measure	Value
S	%	7.81
Cl	%	1.86
Al	mg/g	51.6
Ba	mg/g	0.8
Ca	mg/g	201.5
C tot	%	<1.0
Cr	μγ/γ*	237
Cu	μγ/γ*	513
Fe	mg/g	16.9
K	mg/g	43.6
Mg	mg/g	18.1
Mn	μγ/γ*	580
Na	mg/g	26.3
Ni	μγ/γ*	92
Sr	mg/g	0.3
Ti	mg/g	10.04
Zn	mg/g	31.86
Pb	μγ/g*	987
Cd	μγ/γ*	45
V	μγ/γ*	18

T deformation	°C	1280
T softening	°C	1288
T melting	°C	1305
* 1γ = 1 µg		

Table 3. 1 – Analysis of slags that covers the materials in corrosion test [27]

Unlike coal ash, waste-to-energy plants ash has a great amount of metals (Al, Cu, Fe, Mn, Ti, Zn, Pb), responsible, together with chlorides and alkaline metals (K, Ca, Na), for low-melting eutectic mixtures. Samples were exposed to the flux (25l/h) in an oxidising sulfuric atmosphere of N_2, containing 5% O_2, 2.5% H_2O and 0.3% SO_2

Duration of tests is 1000 hours, with intermediate stops every 250 hours for the corrosive kinetics determination through gravimetrical measures. Every stops required the fresh ash deposit restore.

Even though the report analysis focuses on five materials, the aim of the present paragraph is to underline the corrosive effect of waste combustion-derived flue gases. Consequently, only two materials has been taken into account, the T22, a “traditional” steel, compared to the nickel alloy IN625, more suitable for corrosive and high-temperature environments. Their chemical composition is shown in the table below.

Material	**C [%]**	**Si [%]**	**Mn [%]**	**Cr [%]**	**Mo [%]**	**Fe [%]**	**Ni [%]**	**Ti [%]**	**Other [%]**
T22	0.12	0.3	0.45	2.25	1.0	98.13	-	-	-
IN625	0.02	-	-	20	9	5	61.38	1.3	Cb = 3.1 Al = 0.2

Table 3. 2 - Chemical composition of examined materials for corrosion analysis [27]

Figure 3. 1 shows the specific materials weight variations as functions of exposition time. Their values are approximate because they could be distorted by scale loss and/or deposit remaining that interacts with sample’s surface. However, they can be useful for a preliminary evaluation.

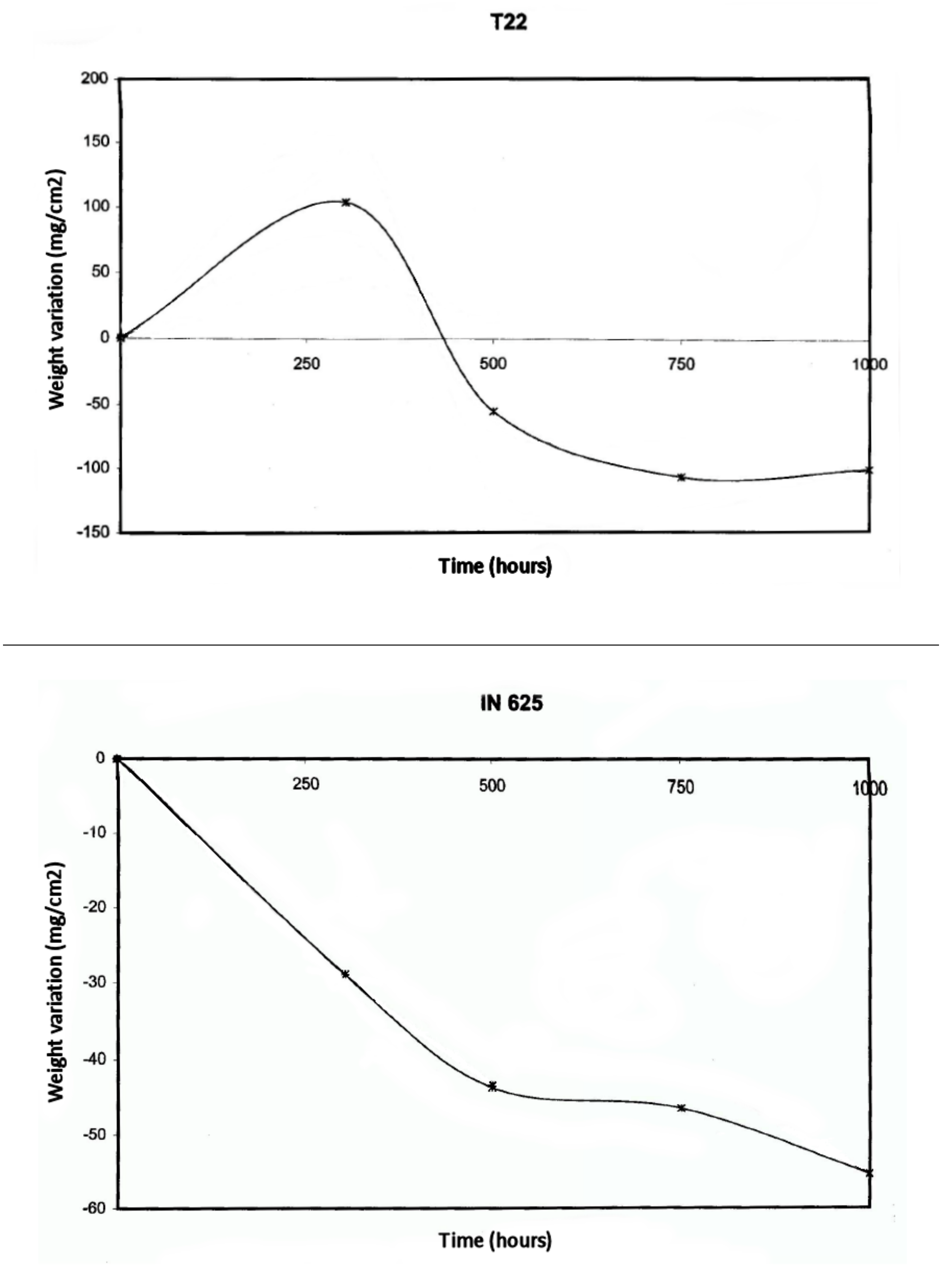


Figure 3. 1 - Weight variation as function of time of T22 and IN625 exposed to a simulated waste combustion atmosphere [27]

T22, a low-alloyed steel, has the worse behaviour and presents considerable weight variations (-150 mg/cm^2), after only 500 hours, while IN625 does not reach T22 weight variation values even at the end of the test. Even though IN625 shows a better reaction in such a corrosive environment, it loses about 55 mg/cm^2 after 1000hours, proving that flue gas derived from waste combustion constitutes a great risk for materials resistance.[27] The material degradation increases with temperature, as will be furtherly presented in the next chapter.

3.2. POSSIBLE SOLUTIONS

A list of possible solutions to problems explained in the paragraph above are provided. They can be divided in four main groups:[8]

a. Decrease costs
  - Adopt a simpler configuration
  - Use less materials or more economical ones
  - Decrease maintenance
  - Co-fire existent plants

b. Increase efficiency
  - Improve boiler efficiency
  - Improve thermodynamic cycle efficiency
  - Adopt a cogenerate configuration, producing heat and power

c. Adopt a different technology
  - Fluidised bed boiler
  - Gasification/pyrolysis

d. Generate additional output
  - Recover materials from ash

The present thesis is concentrated on improving the thermodynamic efficiency of the cycle. Taking into account the considerations made in the **Chapter 2** about ways for raising the cycle efficiency, better cycle parameters can be adopted (higher p_V, T_{SH}, lower p_K) or improve the plant configuration, adding reheat. However, as explained before, the condensation pressure p_K is determined by ambient conditions and water availability, so it can’t be furtherly reduced unless the possibility of moving the plant in a more cold country is feasible. Higher p_V requires either higher T_{SH} or reheat to limit liquid fraction at steam turbine outlet (to limit the cavitation phenomenon).

As a result, the two remaining conditions to improve the steam cycle efficiency are either increase the superheating temperature T_{SH} and/or introduce the reheat practise. The two possibilities are further considered in the following paragraphs in order to understand and choose on which improvement (and how to develop it) is better to concentrate.

3.2.1. Increase T_{SH}

Because of the high content of species that form corrosive mixtures at high temperature, superheating temperature is limited in traditional configurations around 450°C. However, some attempts to overcome the problem have been developed.

The first plant concept presented is by **Hans Hunsinger** (Karlsruhe Institute of Technology). Its functioning principle is explained in the **Figure 3. 2** below.

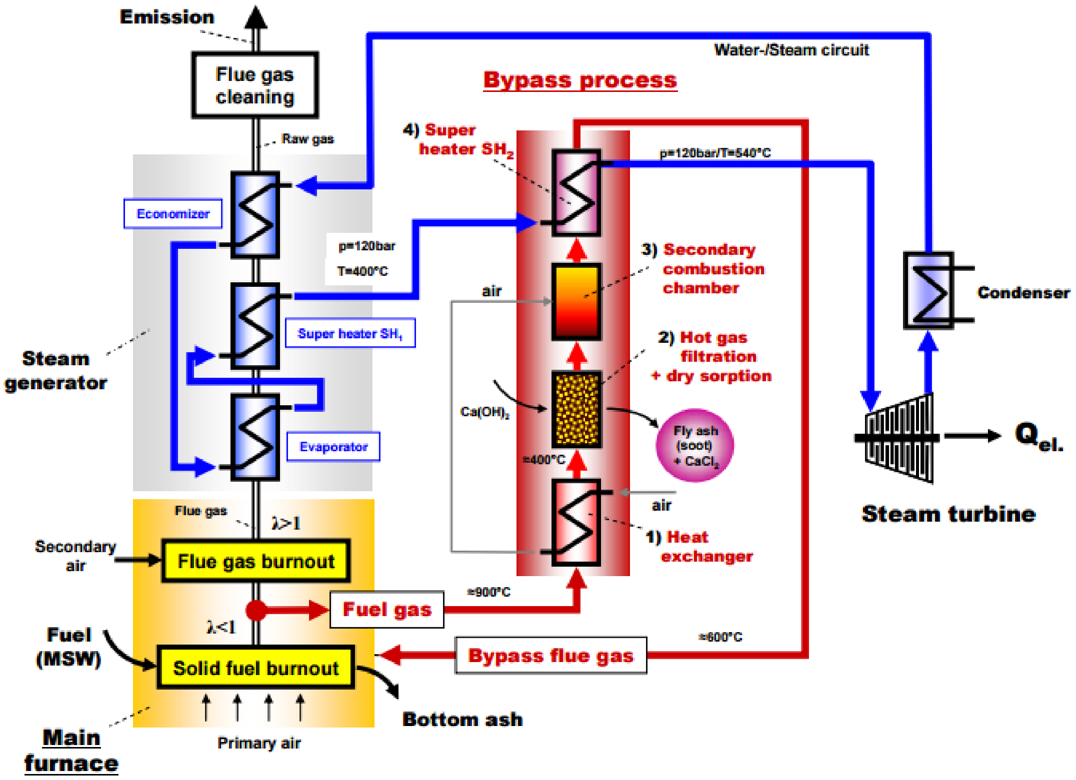


Figure 3. 2 - Bypass plant concept by Hans Hunsinger [8]

A small part of the primary formed calorific flue gas can be extracted prior entering the flue gas burnout zone (where the air excess λ is lower than 1). In fact, solid fuels combustion can be subdivided in three sub-processes: drying, pyrolysis and burnout of the fixed carbon. Every sub-process can be roughly associated with a part of the boiler grid, calling them "zones". The major quantity of corrosive and toxic mixtures have been proven to form in the burnout zone of the grid. This fuel gas passes a small bypass system equipped with the subsequently arranged process steps gas cleaning, combustion and further super-heating of the pre-generated steam from the main boiler. The off-gas and the removed solid residues are recycled back to defined locations of the main furnace. The process combines the advantages of combustion and gasification and allows generation of very high temperature steam without corrosion and fouling problems even when burning such difficult fuel as municipal waste. The process performs also in minimized NO_x formation.[28]

The second plant concept is by **Ole Hedegaard Madsen**, **Thomas Wagner Sødring** (Babcock & Wilcox Vølund). The superheating temperature can be increased without the risk of super-heaters corrosion integrating the new concept in the boiler without raising its complexity.

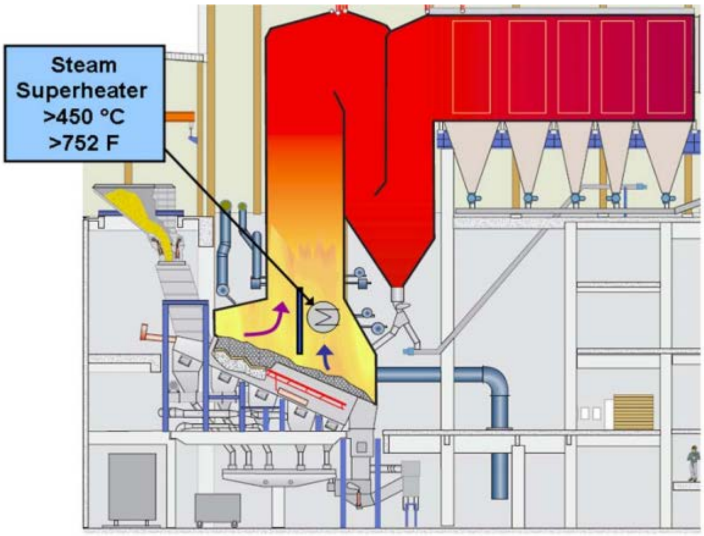


Figure 3. 3 - Steam Booster plant concept by Ole Hedegaard Madsen and Thomas Wagner Sødring [8]

The idea is to separate the flue from the grate into two or more fractions having one fraction of the flue gas with a high heat flux and a low chlorine concentration (**Figure 3. 3**). This is possible thanks to a study operated by the same firm, which trace the corrosive mixture formation as a function of the waste position on the grate. The result is that chlorine formation can be detected in a particular zone of the grid: flue gas can thus be collected in zones far from the chlorine formation's. This isolated portion of the flue gas may be directed to a separate super heater section, where it can raise the steam temperature up to the limit of traditional waste-to-energy plants because the corrosion risk is no longer present. The elevated super heater steam temperature (≈ 500°C) could then increase the electrical efficiency of the waste fired power plant. [30]

The third plant is only a hypothesise one and is partly inspired by the first concept explained before. It considers the presence of an outer biogas fired combustor that heats up flue gas from the furnace, exiting the first super-heater (within the boiler) in a heat exchanger where the secondary combustor's flue gas (warmer than the gas exiting SH1) enters. (**Figure 3. 4**) Biogas is a cleaner fuel than flue gas from waste combustion, so superheating temperature can be raised in the heat exchanger without corrosion problems. Flue gas exiting the heat exchanger pre-heat the air entering the combustor, before being discharged through the stack.[8]

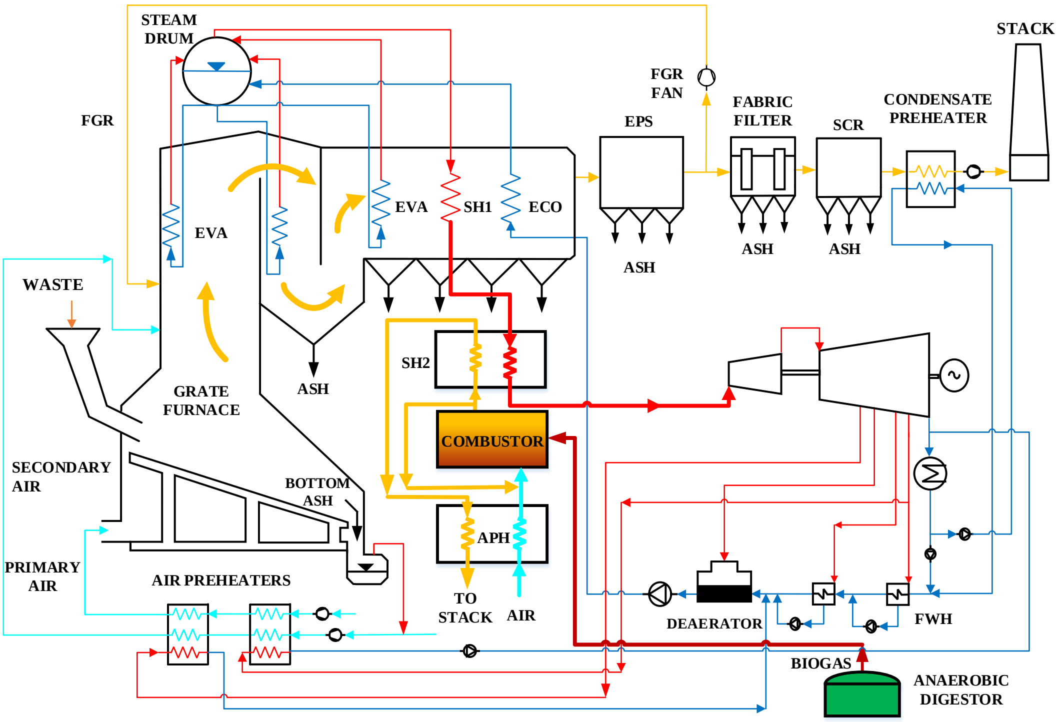


Figure 3. 4 - Hypothesise waste-to-energy plant with external superheating with biogas [8]

3.2.2. Reheat introduction

The other possibility to increase the thermodynamic cycle efficiency is to introduce the reheat. It is not usually introduced in traditional waste-to-energy plant because it would be equivalent to duplicate the critical sections when it comes to corrosive problems. However, some successful attempts to bypass the problem has been made, resulting in perfectly functioning plants.

The first example is the Fisia-Babcock plant in **Rüdersdorf**, which operates a reheat in its thermodynamic cycle. In order not to raise superheating and reheat temperature (which are critical for corrosive mixtures formation), they are maintained at 420°C, reducing the overall performances (**Figure 3. 5**).[31]

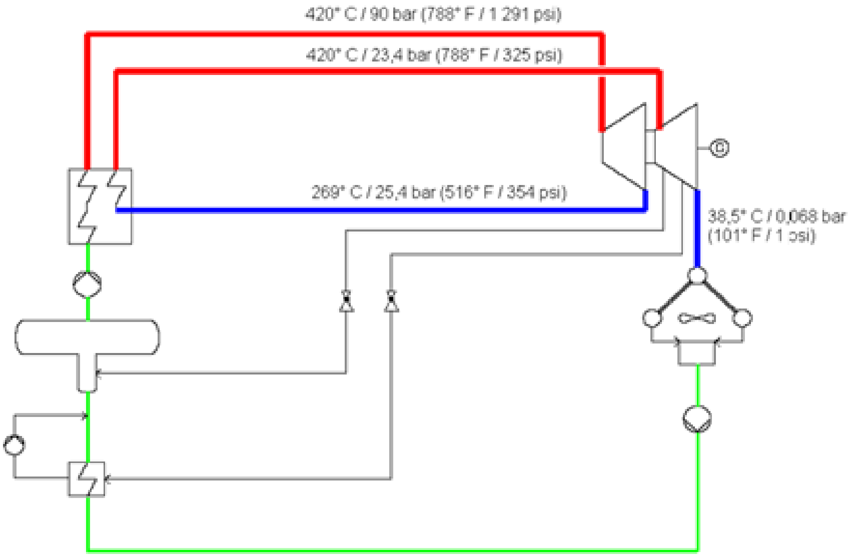


Figure 3. 5 - Waste-to-energy plant in Rüdersdorf with reheat [8]

The second example is the waste-to-energy plant in **Amsterdam**. Reheat is operated through high-pressure saturated fluid. The evaporator, in fact, is maintained to high pressure (145 bar). Saturated water coming from economiser is sent within the boiler to heat up and transform in saturated vapour. The latter is then split in two flows: the first one is superheated in the super-heaters, as in traditional plants and sent within the turbine body. The second part, though, enters a heat exchanger, where it cools down, transferring its heat to superheated vapour exiting the high-pressure turbine, operating the reheat. Saturated vapour becomes, thus, saturated liquid and is sent again in the evaporator, to be heated up again. The overall steam cycle is shows in the **Figure 3. 6** below.[32]

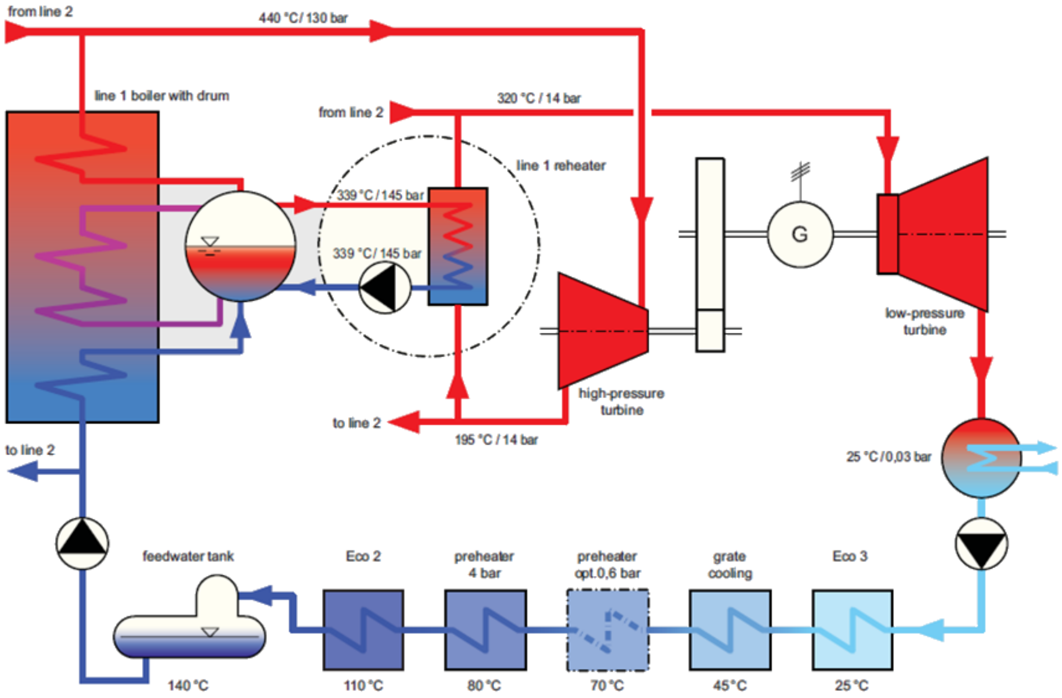


Figure 3. 6 - Waste-to-energy plant in Amsterdam with high pressure liquid [8]

3.3. CONCLUSIONS

Corrosion is the main problem of waste-to-energy plants when it comes to increase their thermodynamic cycle efficiency. Superheating temperature can be increased as long as an alternative solution is provided, which doesn't include a temperature increase of waste-to-energy flue gas as it is. Numerous concept solutions can be proposed, but their realisation would require a great capital expenditure because of the elevated plant complexion that it would lead to. The better solution seems to be reheat introduction. Numerous successfully functioning plants were realised through Europe. The first one includes reheat without increase plant complexity, but renouncing to high performances. On the other hand, the second plant prefers a higher plant complexion in order to achieve better cycle efficiencies. In Chapter 5, an innovative configuration would be presented. The aim is to merge the positive aspects of the two plants with reheat (plant complexion as little as possible and high performances). In further chapters, the innovative plant configuration would be analysed, varying vaporisation pressure and reheat pressure in order to achieve the better parameters combination in the efficiency viewpoint. Innovative plant performances would be confronted with a waste-to-energy plant's with the same input values, but without reheat.

Chapter 4

Methodology and Calculation tools

This chapter aims to illustrate the principal methodologies used for the overall calculations in the plants simulation, introducing the software used for performances and costs evaluation.

4.1.REFERENCE PLANT

The innovative configuration for the waste-to-energy plant would be furtherly confronted with a more traditional plant configuration, without reheat. The aim is to demonstrate that innovative plant has higher performances than the more traditional one. The plant with which the confrontation has been made is shown in the two figures below.

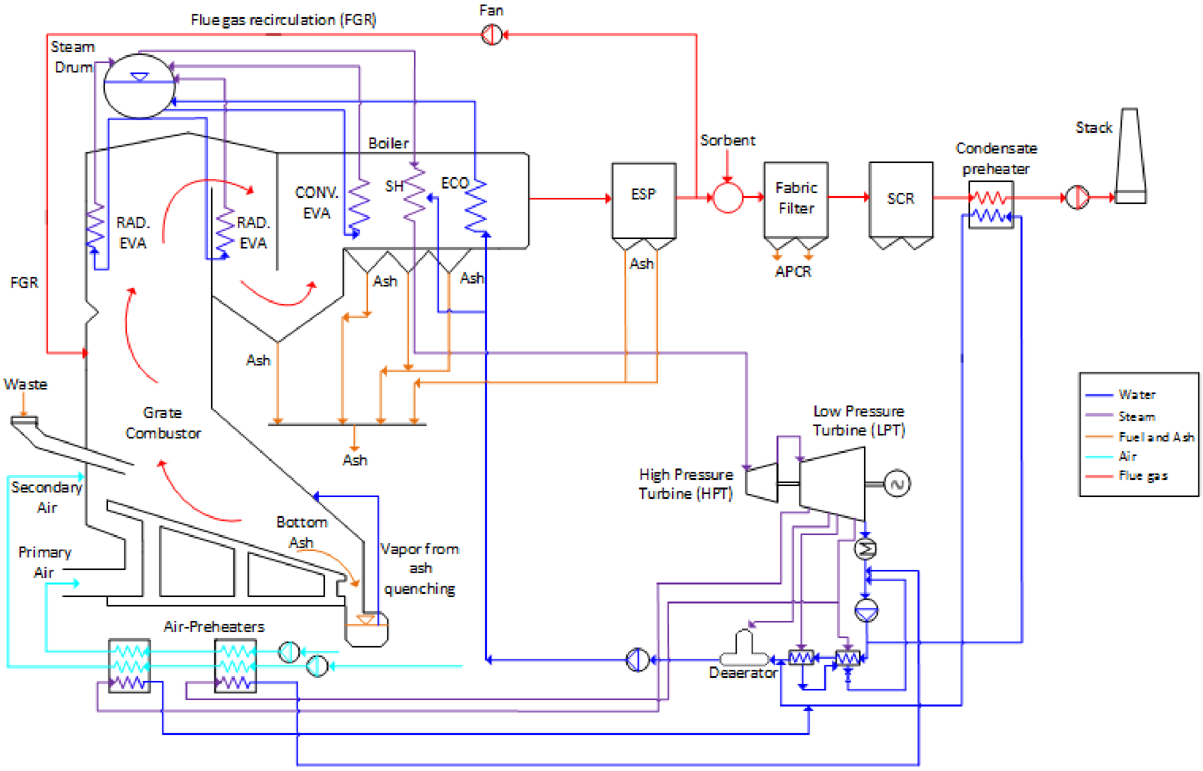


Figure 4. 1 - Comparison waste-to-energy plant configuration

The primary and secondary air are heated up in two pre-heaters, exchanging heat with two vapour fluxes spilled from the turbine. On the grate, waste and primary air start pyrolysis thanks to burners located on the combustor’s walls. The secondary air guarantees the real combustion. Flue gases resultants from combustion mixes with flue gases recirculated after the electron-filter. Then flue gases exchange their heat with water passing through the evaporator, super-heater and economiser. They pass then through a series of filters and are finally discharged in the atmosphere through the stack. The steam cycle comprehend two low-pressure regenerators and a deaerator.

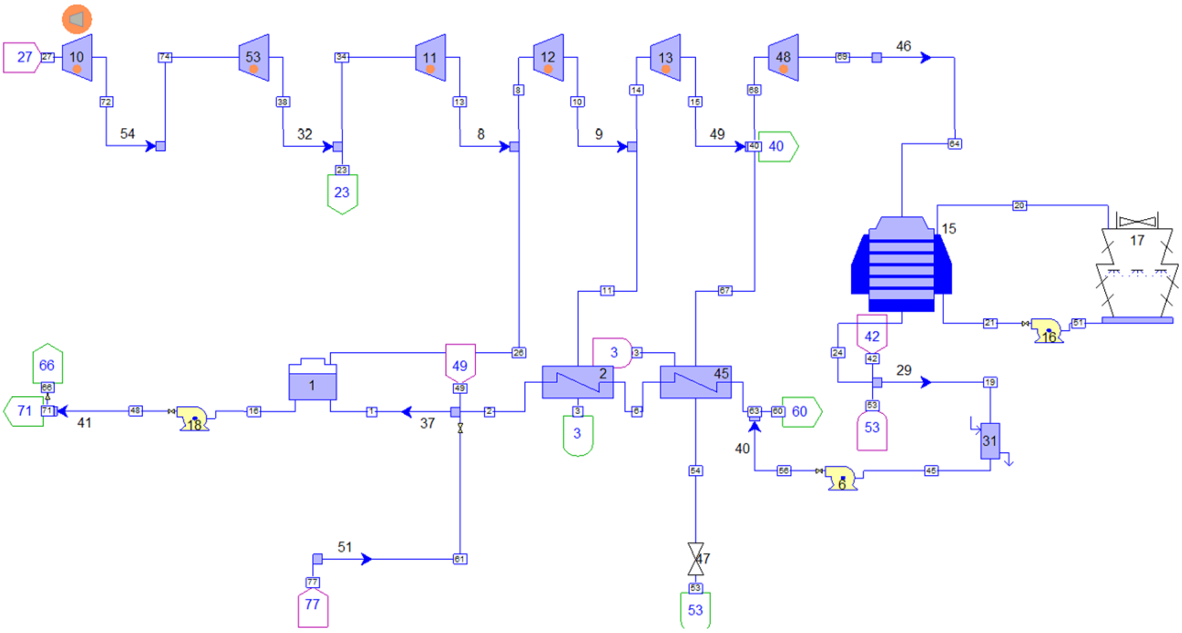
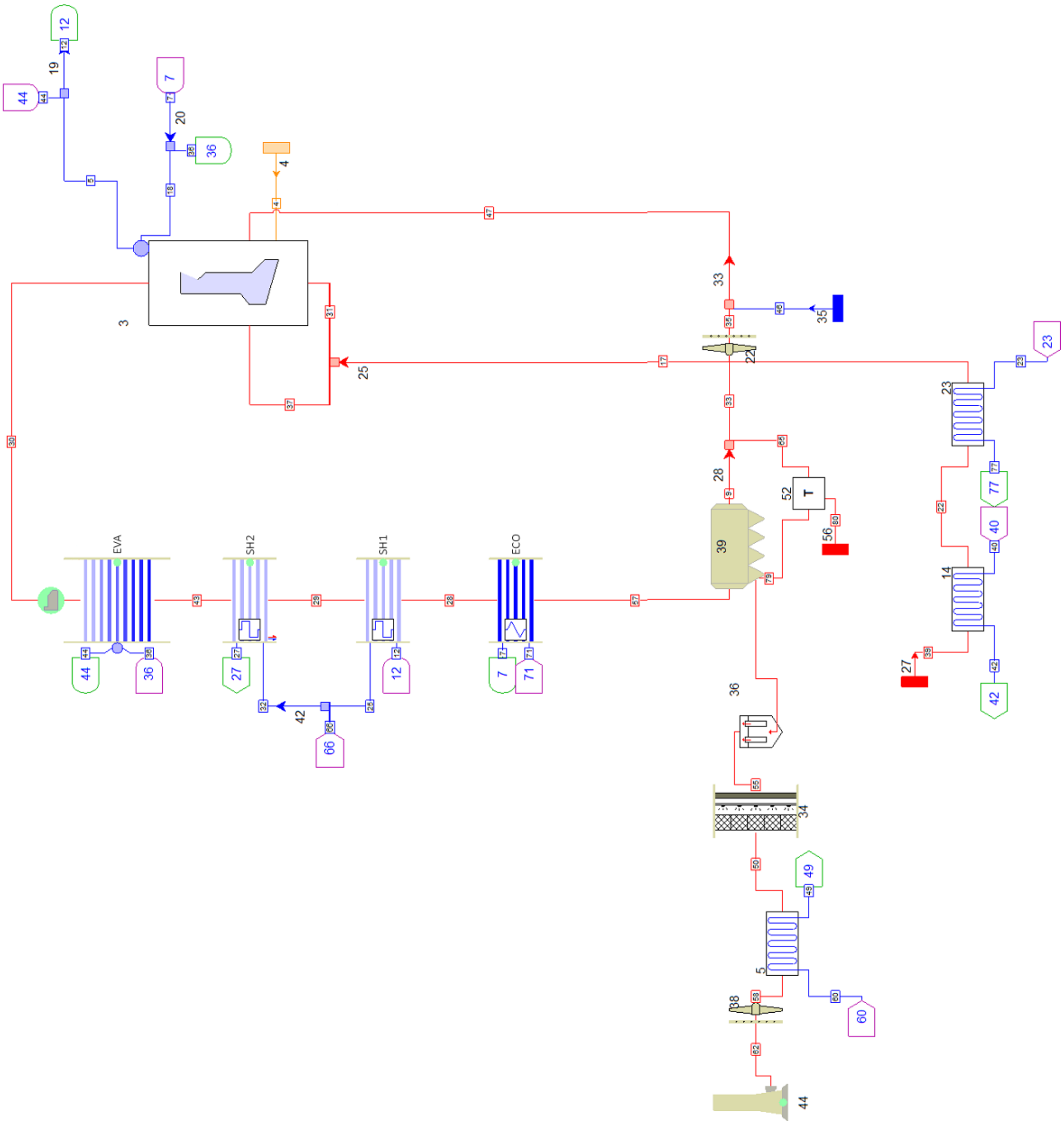


Figure 4. 2 - Reference plant scheme

Flue gases exit the boiler *3* and heats up the fluid in the convective evaporator (*EVA*), super-heaters (*SH2* and *SH1*) and economiser (*ECO*). Component *39* constitutes the electrostatic filter. The splitter *28* divides flue gases in two fluxes: part of them are sent again within the boiler, after being cooled down by vapour produced in ash quenching. The major part of the flux is cooled down of about 10°C (to simulate the temperature loss caused by flue gas treatment system), enters the condensate preheater and is then discharged in the atmosphere through the stack. Electrostatic filter, bag filter *36* and SCR *34* constitute the flue gas treatment system. Their contribution is represented in the Thermoflex model, through pressure drop within each component. However, temperature drop and air infiltrations are simulated thanks to component *52*, which add ambient air from the outside to flue gas directed to the stack until they cool down of 10°C.



The turbine body has five bleedings: two for the low-pressure regenerators, one for the deaerator and two to heat up the air in air preheaters (fluxes *23* and *40*). Cooled vapour from the high-pressure regenerator enters the low-pressure ones, which in turn inserts its cooled vapour within the condenser *15* (flow *53*). Part of water flux is separated from the principal to be heated up to deaerator temperature within the condensate preheater. A pump raises then saturated water pressure (exiting the deaerator *1*), which is sent to the main boiler. It passes through the economiser and is heated up in the evaporator: part of the flow rate goes to the convective evaporator (flux *36*), re-joining the principal flux before entering the super-heater (flux *44*). The subdivision between convective (*EVA*) and radiative evaporator (which is integrated within the boiler component, represented through the blue circle in the right corner of the boiler) is only a modelling necessity, in order to highlight the two heat sources of this heat exchanger. Superheated vapour exits then the super-heaters *SH2* and *SH1*, entering the turbine body (flux *27*).

4.2.SOFTWARE - THERMOFLEX

One main software has been used to develop the plant model, estimate operating parameters and performances of each component and operate mass and energy balances evaluation. The aim of the software is to evaluate the plant overall performances and provide its rough cost analysis. The software is realised by Thermoflow, the leading developer of thermal engineering software for the power and cogeneration industries.

Thermoflex is a modular program with a graphical interface that allows to assemble a model from icons representing the needed components (**Figure 4. 3**). The program covers both design and off-design simulation, even if in the present thesis only the design mode has been considered.[33]

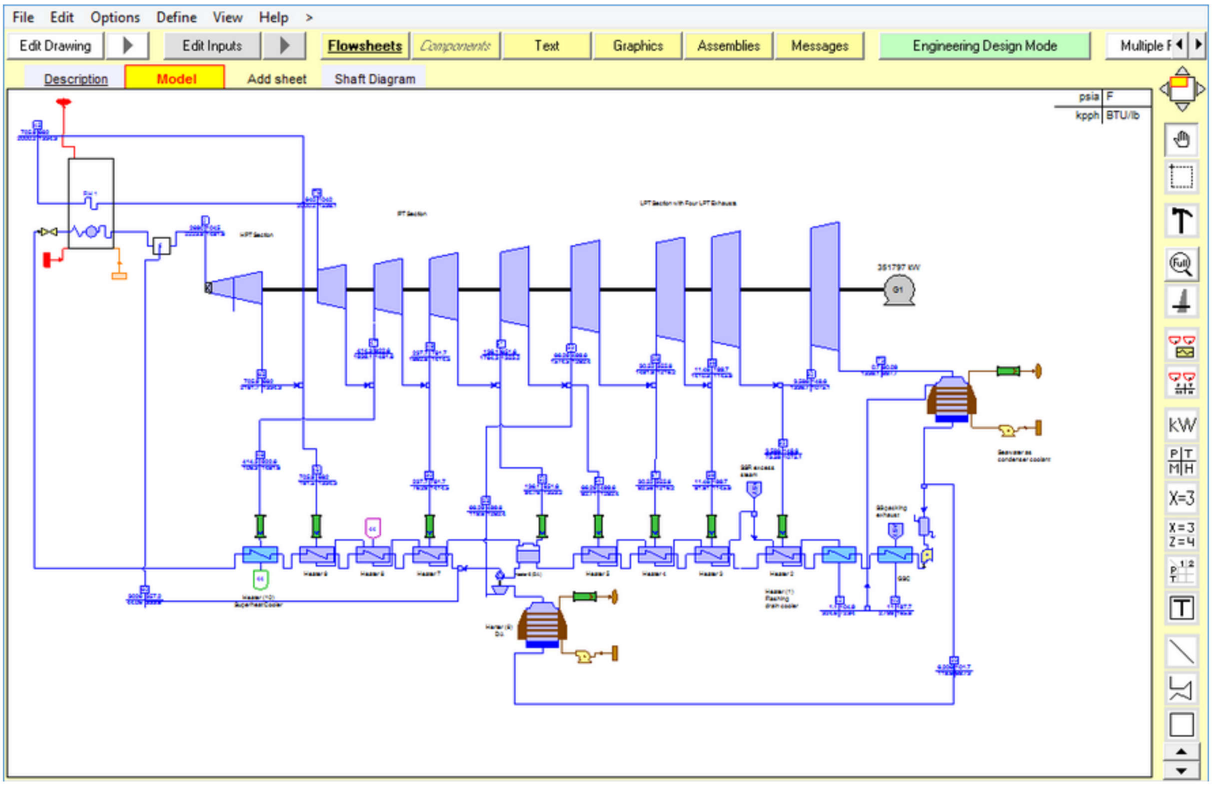


Figure 4. 3 - Sample Thermoflex window

While the button “Edit Drawing” consents to modify the arrangement of components, their links and permits to add or remove them, the button “Edit Inputs” consents to modify the components parameters.

In particular, clicking on every icon, it is possible to modify every input of the specific component through a user-friendly window. The program considers some values only as “first attempt” values, because of the iterative nature of the calculations conducted by Thermoflex. For example, while feedback rings are present, Thermoflex assumes as first attempt value the one specified in the specific components input, which would be updated with the one derived from the following calculation. Furthermore, the programme updates some values after every iterative cycle because the degrees of freedom are limited: every plant is a discrete complex system, with a fixed number of dependant and independent variables. In particular, some parameters appear to be dependant from others, so not every input inserted in every component specific page, would be maintained as specified by the user. However, the user can force the programme to reconsider the priority scale of dependent and independent variables specifying a feedback loop, which control that the wanted variable maintains the required value in every iterative cycle.

After completing the inputs specification, the plant can be computed clicking on the arrow beside the “Edit Inputs” button. Assigned the inputs for every component, the programme resolves mass and energy balances for each and every of them, respecting the constraints imposed and iteratively repeating the calculations until it reaches the sufficiently stable, stationary solution (if it exists).

The labels “Text” and “Graphics” allow verifying outputs and overall performances of the plant as well as performances of the single plant components. Output pages provide values of mass flow rate, temperature, pressure, specific volume, enthalpy, entropy and principal characteristics of every components as well as energetic performances of the overall plant.

The button “Assemblies”, on the other hand, permits to examine turbine or boiler specific data. Finally, clicking on the button “Messages”, a window opens, showing all the program messages about computation (warnings, errors, advisories or general messages). The program allows setting up control loops over different parameters changing others, in order to impose the control and the program attention over specific required outputs. Thermoflex was used mostly for the performances evaluation and plant simulation, as well as the cost analysis of the convective part of the boiler. Radiative part of the boiler, not calculated by the cost-analysis software part, has been extrapolated starting from data acquired in previous simulations, related to a different plant. The difference among plants does not reduce the overall accuracy, because costs are then re-proportioned over the cost of the boiler in the innovative configuration plant. Further explanation would be available in the Chapter 5 and 6.

Modular structure of the programme makes it capable of simulating whichever energetic system: if default components do not consent an adequate representation of the plant, new constituents can be created.

4.3. MASS AND ENERGY BALANCES

Thermoflex operated cycles of approximation in order to reach the values combination that allows satisfying all the mass and energy balances for every and each component. Hereinafter the principal components mass and energy balances would be shown, in order to give a clear idea on the calculations operated by the software. Obviously, the calculations shown below are simplified of some factors or corrective coefficients that Thermoflex applies in order to better adhere on the reality (i.e. the cavitation and condensate drops impact on the mass and energy balances at the end of low-pressure turbine, water leakages within components, etc.). The presence of these corrective coefficients, implemented within the programme with data and graphics extracted from literature most advanced texts, together with the plant’s complexion, are reasons why the necessity of a software is compulsory in order to produce sufficiently reliable output data.

4.3.1. Condenser

Starting from the steam cycle, the first component analysed is the condenser (**Figure 4. 4**).

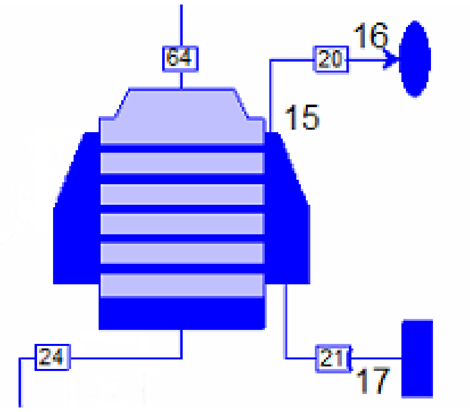


Figure 4. 4 – Condenser

The condenser allows the heat exchanging between water from steam cycle (fluxes *64* and *24*) and water in an evaporative tower. Not considering energy or mass leakages, the heat quantity lost from steam cycle's water has to be the same as the one acquired by the cooling water.

$$\mathrm{Q_{cond}} = \dot{\mathrm{m}}_{64}\mathrm{h}_{64} - \dot{\mathrm{m}}_{24}\mathrm{h}_{24} = \dot{\mathrm{m}}_{20}\mathrm{h}_{20} - \dot{\mathrm{m}}_{21}\mathrm{h}_{21}$$

Having indicated the enthalpy of every numbered flow with the letter h. In every component, excluding mass leakages, the water mass flow that enters has to be the same as the one that exits the components, so the equation becomes:

$$\mathrm{Q_{cond}} = \dot{\mathrm{m}}_{64}(\mathrm{h}_{64} - \mathrm{h}_{24}) = \dot{\mathrm{m}}_{20}(\mathrm{h}_{20} - \mathrm{h}_{21})$$

Hereinafter, this hypothesis would be taken for granted. As explained before, the condenser receives vapour from low-pressure turbine (in particular saturated vapour with high vapour quality) and condenses it until it becomes condensed water (vapour quality equal to zero).

So, known the low-pressure turbine discharge pressure (which is a data hypothesis), not considering pressure drops, the exit water enthalpy is known.

$$\mathrm{h}_{24} = \mathrm{f}(\mathrm{p}_{24} = \mathrm{p}_{64}, \mathrm{x}_{24} = 0)$$

The same is valid for the inlet steam cycle's water enthalpy, a part from the vapour quality, which is now an unknown quantity, but which depends from the vapour expansions within the two turbine's bodies.

4.3.2. Pump and Fan

The next **Figure 4. 5** shows a pump and a fan. They are regulated by the same principles since they both have the objective to increase a fluid pressure (liquid and gaseous respectively). The following equations would be shown for the pump case, but the same considerations are valid for the fan as well.

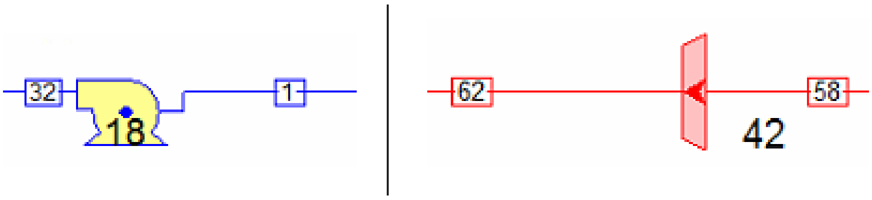


Figure 4. 5 – Pump and Fan

Every pump in the cycle has a hydraulic efficiency, which is linked to the fluxes pressures and enthalpies. The hydraulic efficiency is the ratio between the ideal and real work required by the pump.

$$\eta_{i,P} = \frac{W_{ideal}}{W_{real}} = \frac{v_1 * (p_{32} - p_1)}{h_{32} - h_1}$$

Where the terms in *p*s indicate the pressures and the *v* term indicates the specific volume of inlet water flow. Usually the temperature difference before and after a pump is negligible, but can be calculated as the difference between the real and ideal work, divided by the water specific heat (indicated with c_L):

$$\Delta T = \frac{\Delta u}{c_L} = \frac{\Delta h - \Delta h_{is}}{c_L} = \frac{(h_{32} - h_1) - v_1 * (p_{32} - p_1)}{c_L}$$

Obviously, heat losses are not considered in the present expressions.

4.3.3. Regenerator

The regenerators too are machines where heat is exchanged from one fluid to another and where mass flow rate is considered a constant.

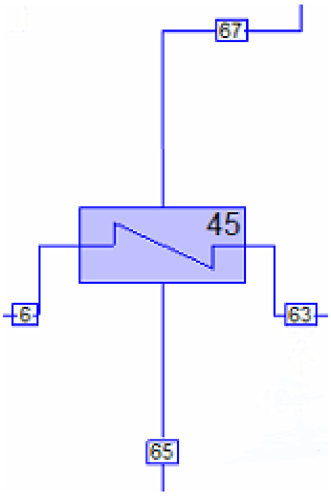


Figure 4. 6 – Regenerator

As explained for the condenser case:

$$Q_{reg} = \dot{m}_{63}(h_6 - h_{63}) = \dot{m}_{67}(h_{67} - h_{65})$$

Obviously, in this case, the fluid that cedes heat is the water (flows *63* and *6*) and the fluid that absorbs this heat is the vapour spilled from the turbine. Usually flow *65* becomes sub-cooled water, so its final state can be calculated splitting the heat exchangers in three fictional parts, where the de-super-heating (I), vapour condensation (II) and sub-cooling (III) take place.

$$Q_{I} = \dot{m}_{67}(h_{67} - h_{II})$$

Q_I indicates the heat exchanged during the de-super-heating phase. The enthalpy of the fictional point *II* is the propriety of the fluid when it starts to condensate. Therefore, its enthalpy is easily calculable knowing the bleeding pressure (hypothesising the absence of pressure drops), because it depends only on this parameter and on the vapour quality, which is equal to one.

$$Q_{II} = \dot{m}_{67}(h_{II} - h_{III})$$

The heat exchanged in the vapour condensation part depends only on the bleeding pressure, because the enthalpy of condensation start (*II*) and condensation finish (*III*) corresponds to the vaporisation points with vapour quality equals to one and to zero, respectively.

Finally, the heat exchanged in the last part calculation can be approximated as follows:

$$Q_{III} = \dot{m}_{67}(h_{III} - h_{65}) = \dot{m}_{67}c_p(T_{III} - T_{65})$$

Where the terms with T indicates obviously the temperature of the point specified with the numbers and the c_p is the water mean specific heat from T_{III} to T_{65}.

4.3.4. Deaerator

The deaerator (**Figure 4. 7**) serves as a machine that heats up the water in order to eliminate insoluble gases infiltrated within the steam cycle mostly from the turbine shaft. Its purpose thus is not to exchange heat, even because all the fluids merge within this device.

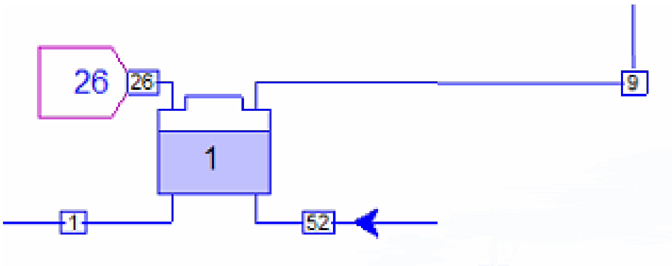


Figure 4. 7 – Deaerator

For this equipment, the outlet water flow must be saturated water at bleeding pressure (so with a vapour quality equal to zero) and the programme must respect this constraint. Moreover, the mass balance must be respected.

$$h_1 = f(p_9, x_1 = 0)$$

$$\dot{m}_{52} + \dot{m}_9 + \dot{m}_{26} = \dot{m}_1$$

Where the term *26* comes directly from the high-temperature air pre-heater and the eventual high-pressure regenerator outlet. The pressure of all fluids should be the same and remain the same without considering pressure drops. Temperature is the vaporisation temperature at bleeding pressure and in order to avoid high exergy losses, flow *52* has a temperature close to the vaporisation one.

Enthalpy of the exiting flow is known, knowing the bleeding pressure and the vapour quality, which is equal to zero.

4.3.5. Turbine

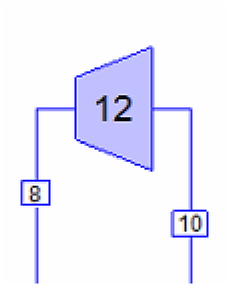


Figure 4. 8 - Turbine element

Every turbine element (**Figure 4. 8**) faces a certain pressure gap, evolving as stabilised by its isentropic efficiency, defined as follows:

$$\eta_{turb} = \frac{\Delta h_{real}}{\Delta h_{ideal}} = \frac{h_8 - h_{10}}{h_8 - h_{10,is}}$$

Where the term $h_{10,is}$ indicates the point *10* on the isobar if the expansion were isentropic. The **Figure 4. 9** below shows the difference between the two expansions.

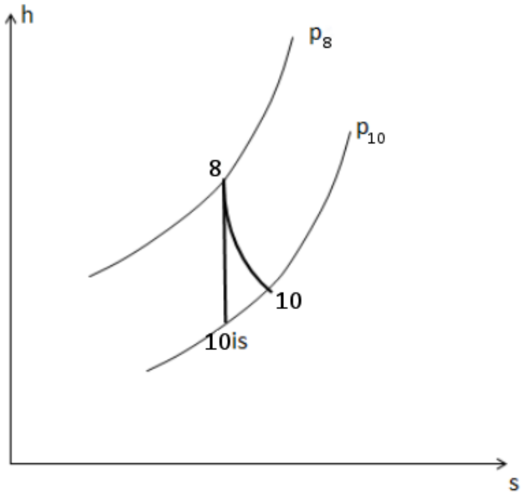


Figure 4. 9 - Turbine ideal and real expansion

From the diagram which indicates the evolution of the enthalpy dependent from the entropy, the point *8* entropy must be the same as the point *10,is*.

$$s_8 = s_{10,is}$$

These equations do not consider the presence of vapour leakages and gas introduction as well as the presence of condensation drops and their effect of the fluid-dynamic and efficiency of the turbine. However, Thermoflex uses a series of coefficients to take these effects into account, which would be furtherly discussed in Chapter 6.

4.3.6. Valve

A valve is a device that allows reducing the pressure of a flux ideally maintaining the same entropy (**Figure 4. 10**).

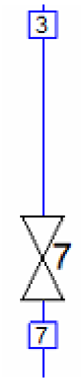


Figure 4. 10 – Valve

As a result, the condition that must be imposed at the fluxes entering and exiting the valve is the following:

$$s_3 = s_7$$

As the case of isentropic expansion.

4.3.7. Splitter/Mixer

The splitter/mixer is a simple device that divides one flow in two flows or mixes them in one, respectively, as is easy to comprehend in the **Figure 4. 11** below.

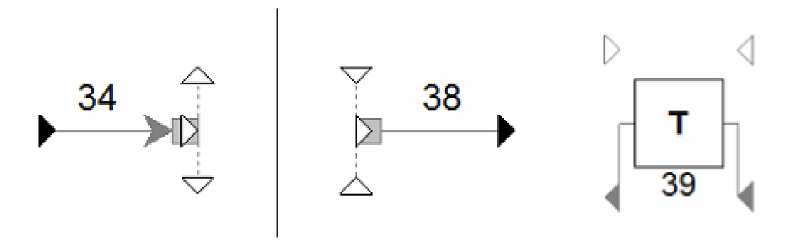


Figure 4. 11 - Splitter and Mixer

Therefore, the conditions of the fluid before and after these devices must be the same:

$$T_{in} = T_{out}$$

$$p_{in} = p_{out}$$

$$h_{in} = h_{out}$$

$$s_{in} = s_{out}$$

The flow mass rate follows the Kirchhoff junction rule for electrical circuits that states that the current entering any junction is equal to the current leaving that junction. As a result, the system have to satisfy the following equations when it faces a splitter or a mixer.

$$\dot{m}_{in} = \dot{m}_{out,1} + \dot{m}_{out,2} \qquad \dot{m}_{in,1} + \dot{m}_{in,2} = \dot{m}_{out}$$

The temperature controller is a particular mixer that unites an inlet flow and another flow which has the purpose to heat up (or cool down) the inner flux. The output flux is thus a fluid with the required temperature, imposed by the device itself. Flow mass rate of the "heating flow" is decided by the temperature controller in order to reach the required temperature of the outlet flow.

4.3.8. Heat exchanger

Examining the flue gases cycle, the first device to be examined is a heat exchanger. This includes all the super-heaters, re-heaters, economisers and evaporators as well as the condensate pre-heater and air pre-heaters.

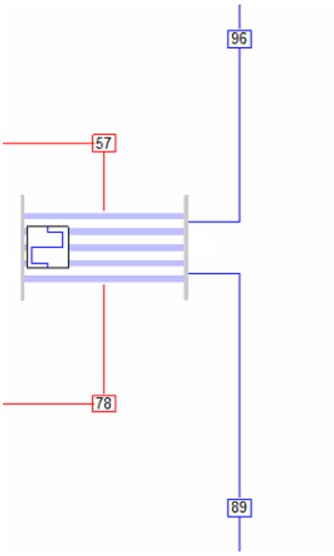


Figure 4. 12 - Heat exchanger

As explained for the condenser and regenerators, the heat exchanger requires the heat transfer from flue gases that cools down to the water/vapour, which heats up. Not considering the heat losses, heat quantity absorbed by the water/vapour must be the same as the one transferred from the flue gases.

$$Q_{HE} = \dot{m}_{57}(h_{57} - h_{78}) = \dot{m}_{89}(h_{96} - h_{89})$$

Some heat exchangers place some additional constraints other than the one above. For example, in evaporators the temperature of condensate have to remain constant. In the last economiser, the temperature of exiting water is equal to the vaporisation temperature minus the ΔT of sub-cooling. Condensate pre-heater has specific constraints over the temperature of all the fluxes.

4.3.9. Combustor

The **combustor** combines the waste coming from the pit, the air, pre-heated within the two pre-heaters and the recirculated flue gases. Water is heated up within the device that operates as an evaporator as well (the blue circle in the up right corner of the **Figure 4. 13**), so the same laws applied for the heat exchangers are valid even in this case.

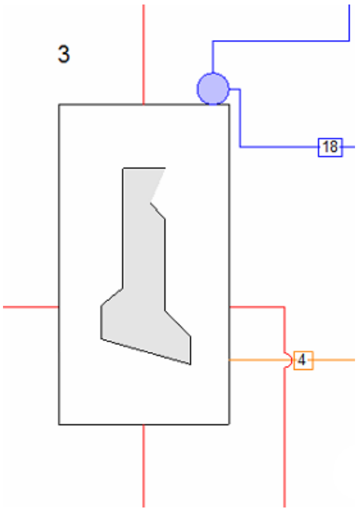


Figure 4. 13 - Boiler

For this reason, the energy equation must be taken into account. The combustion emanates energy. Incoming energy is also provided by waste and air which enters the furnace; higher their temperature higher their energy. Part of this energy is wasted due to heat losses, another part is transferred to flue gases (which are the main combustion reaction products) and the last part is used to heat up the water within the radiative evaporator. The equation is thus as follows.

$$\begin{aligned}\dot{m}_{waste}\bar{c}_{p,waste}(T_{waste}-T_{ref})+\dot{m}_{air}\bar{c}_{p,air}(T_{air}-T_{ref})+\dot{m}_{waste}LHV_{waste}(1-\varepsilon) = \\ = (\dot{m}_{waste}+\dot{m}_{air})\bar{c}_{p,FG}(T_{FG}-T_{ref})+\dot{m}_{water}\left(h_{water,out}-h_{water,in}\right) \\ +\dot{Q}_{heat\_losses}\end{aligned}$$

The terms represents respectively waste and air initial energy and the combustion energy, with the term ε that's indicate the incomplete combustion losses. The terms on the right side of the equal sign corresponds to flue gases energy, energy absorbed to vaporise the water within the radiative evaporator and heat losses from furnace walls.

The terms with dotted *m*s represents fluid's mass flux rates. The specific heat terms are considered as the mean between the two temperatures indicated for each element, since they depends from temperature. The rigorous equation would require the presence of an integral of c_p between the reference and the specific fluid temperature. Every temperature can be chosen as reference, but the common one is ambient temperature (15°C or 20°C).

4.4. FLUE GASES COMPOSITION

The boiler performs the combustion, which can be simplified in a chemical transformation. Combustion is a reaction in kinetic equilibrium, where many free radicals creates during the reaction as well as the incomplete combustion products. The following equation aims only to show the main combustion reaction in the not-realist hypothesis that the combustion occurs in chemical equilibrium. The presence of free radicals and incomplete combustion products are the main reasons for the pollutants existence and creation. The following equation shows a very simplified model of the real chemical reaction that occurs within the boiler:

$$\begin{gathered}(\mathrm{aH_2O + bAsh + cC + dH + eN + fCl + gS + hO}) + (\mathrm{jO_2 + kCO_2 + iH_2O + lN_2 + mAr}) \\ + (\mathrm{nO_2 + oCO_2 + pH_2O + qN_2 + rAr + sAsh + tSO_2}) \rightarrow \\ \rightarrow \ \mathrm{uH_2O + vAsh + wO_2 + xCO_2 + yN_2 + zAr + zzSO_2}\end{gathered}$$

The components within brackets are members of the same constituents, which are respectively fuel, air and recirculated flue gases, which have a different composition respect to the flue gases exiting the boiler (on the right side of the arrow) because of the merge with ash cooling water. Ashes presents within the waste are not the same as the ones present among the products, because of the presence of fixed carbon within waste as well as chloride. Fixed carbon is a certain amount of carbon (specified in the waste composition) that does not volatilise in the combustion reaction, so remains unburned. As a result, its presence increases the amount of ashes within the reaction products (both fly and bottom ashes) and decreases the amount of energy and CO_2 theoretically obtainable from the ideal combustion. Flue gas transport a negligible quantity of ash, because they pass through the electrostatic filter, which blocks the major part of them.

The real reaction comprehend the same reactants components, but a great amount of final products, as NO_x (NO, NO_2). In ideal reaction, N_2 remains as it is since it is an inert gas at controlled temperatures. However, the high temperature reached during intermediate reactions in the combustion, in presence of high quantities of oxygen, promotes the nitrogen oxidation, forming these highly pollutants. Moreover, a certain amount of incomplete combustion products or intermediate products remains, such as OH, SO_3, HCl, which further react to create final products or remain within flue gases and/or ashes at the end of combustion.

Humid air composition is known from the data hypothesis, as well as the fuel and flue gases composition. The fuel is knows in weight composition, but molar composition is needed in order to apply the equilibrium shown before (again, the reaction presented is only a mere simplification of the more complex real reaction). Moreover, flue gases and air composition are provided as molar percentage composition, but actual molar composition is needed in order to calculate the equation above.

As a result, it is important to operate a conversion in order to verify the programme calculations (while in this paragraph only the calculation method would be explained, in Chapter 7 numerical results would be provided).

In the case of fuel composition, its overall mass is known (since is known its flow mass rate) as well as its weighted percentage composition. As a result, the mass of every fuel constituents can be calculated as follows.

$$\text{constituent mass[kg]} = \text{constituent mass percentage}\left[\frac{\mathrm{kg_{constituent}}}{\mathrm{kg_{fuel}}}\right] * \text{fuel mass}\ [\mathrm{kg_{fuel}}]$$

Knowing the molar mass of every constituent, their mass in kilograms can be converted in mole numbers.

$$\text{constituent mole number [mol]} = \frac{\text{constituent mass [kg]}}{\text{constituent molar mass } \left[\frac{\text{kg}}{\text{kmol}}\right]} * 1000$$

The numbers from *a* to *h* are the results of this last equation repeated for each fuel constituent.

The only exception is the ash calculation. Many sub-constituents compose the ashes and the calculation must be repeated for each and every of them. Ashes weight composition is known, so the procedure is the same. In this case, however, *b* is the sum of all the ashes constituents' mole numbers.

In case of air and flue gases, molar percentage is known as well as the total mass in kilograms. The calculation would be performed for air case, but the same could be applied for flue gases (even if in the last case the procedure for ashes must be performed as well).

The mass in kilograms of one mole of air is calculated and then a simple proportion allows to know the molar composition of every air constituent.

Knowing the molar percentage of every constituent (which, in one mole of air, becomes their mole number) and its molar mass, merged with the hypothesis of one mole of air, allows to calculate the mass of every constituent within a mole of air.

$$\begin{gathered}\text{constituent mass in one mole of air [g]} = \\ = \text{constituent mole number [mol]} * \text{constituent molar mass } \left[\frac{\text{g}}{\text{mol}}\right]\end{gathered}$$

All of the constituent's masses in one mole of air can be summed, in order to obtain the mass in kilograms of one mole of air. In order to reproduce the air mole quantity necessary to have the weight of the total mass required, a simple proportion is operated.

$$\text{constituent mass in one mole} : \text{mass of one mole} = \text{constituent mass} : \text{total mass}$$

Which can be rearranged in order to obtain the constituent real mass within the actual mass of air necessary for the combustion.

$$\text{constituent mass [g]} = \frac{\text{constituent mass in one mole [g]} * \text{total mass[g]}}{\text{mass of one mole [g]}}$$

Having the constituent mass for each air component, the previous calculation can be repeated and the constituents mole number found.

$$\text{constituent mole number [mol]} = \frac{\text{constituent mass [g]}}{\text{constituent molar mass } \left[\frac{\text{g}}{\text{mol}}\right]}$$

This procedure, repeated for flue gases as well, allows knowing numbers from *j* to *t*. Numbers from *u* to *zz*, however, can be calculated from the reactants with simple balances.

$$\text{Water: } \mathrm{a} + \frac{\mathrm{d}}{2} + \mathrm{i} + \mathrm{p} = \mathrm{u}$$

$$\text{Ash: } \mathrm{b} + \mathrm{s} = \mathrm{v}$$

$$\text{Carbon dioxide: } c + k + o = x$$

$$\text{Nitrogen: } \frac{e}{2} + l + q = y$$

$$\text{Sulphur dioxide: } g + t = zz$$

$$\text{Oxigen: } \frac{h}{2} + j + n - c - s - \frac{h}{4} = w$$

$$\text{Argon: } m + s = z$$

Chlorine is only present among reactants because its contribution is dedicated in a small part to hydrochloric acid creation that would be a part of flue gases (and be one of the hot corrosion causes). The major part of this just small part of chlorine, however, would constitute ashes within the boiler, with unknown composition.

Moreover, ash can be subdivided in fly ash, which would exit the boiler with flue gases and bottom ash, which remains within the boiler and would be specifically disposed of. However, in addition to ash coming from fuel and recirculated flue gases, final ash improves also for the presence of fixed carbon within fuel. The combustion, in fact, does not have a hundred percent efficiency, which means that a part of carbon does not complete the combustion and remains as unburned carbon within both fly and bottom ashes.

These changings, which is thus not possible to take into account in the present simplified treatise, do not excessively modify the output values, the difference is so negligible that calculations and real output values can be optimally confronted as well.

4.5. HEAT EXCHANGERS DIMENSIONING

Heat exchangers dimensioning programme has been verified as well. Tubes' diameter and thicknesses are input data in Thermoflex, while the programme automatically calculates their number. Since the calculation procedure is not clear (because the algorithms at these calculations' base are intellectual property of the producer and covered by industrial secret), a calculation for tube number is proposed furtherly in this sub-paragraph.

The treatise below has been developed thanks to the methodology indicated in [4]

At first, convective coefficient has to be calculated both of water and flue gases side.

Starting from **water side**, all water properties are known if its pressure (which remains practically constant from inlet to outlet), temperature (inlet, outlet) and wall temperature are also known.

In particular, thermodynamic texts indicate for heat exchangers and for fluid within tubes (as the water case), a calculation pressure defined as follows.

$$p_{calc.H_2O} = \frac{p_{H_2O,in} + p_{H_2O,out}}{2}$$

It is an arithmetic mean between inlet and outlet pressures. Since their difference is negligible, one of them can be adopted indifferently as calculation pressure. Calculation temperature, however, is defined starting from tube wall temperature and mean temperature between inlet and outlet.

$$T_{calc,H_2O} = \frac{T_{wall} + T_{mean}}{2}$$

Wall temperature can be obtained by output files of Thermoflex.

Calculation temperature and pressure can be used to extract the principal water properties, as dynamic viscosity μ_{H2O}, specific heat cp_{H2O}, thermal conductibility λ_{H2O} and density ρ_{H2O}.

Thank to water principal properties as well as its velocity within the tubes of the heat exchanger (w_{H2O}) and the internal tubes diameter (D_{in}), both provided by Thermoflex, Prandtl, Reynolds and Nusselt numbers can be calculated.

$$Pr_{H_2O} = \frac{\mu_{H_2O} * c_{p,H_2O}}{\lambda_{H_2O}}$$

Reynolds number is calculated over the internal diameter D_{in}.

$$Re_{H_2O} = \frac{\rho_{H_2O} * w_{H_2O} * D_{in}}{\mu_{H_2O}}$$

Nusselt number is calculated with a formula called Petukhov equation, used for forced convection in tubes.

$$Nu_{H_2O} = Re_{H_2O} * Pr_{H_2O} * \frac{\left(0.78 * \ln\left(Re_{H_2O}\right) - 1.5\right)^{-2}}{8} * \frac{\left(1 + \left(\frac{D_{in}}{L_t}\right)^{\frac{2}{3}}\right)}{1 + 12.7 * \sqrt{\frac{\left(0.78 * \ln\left(Re_{H_2O}\right) - 1.5\right)^{-2}}{8}} * \left(Pr_{H_2O}^{\frac{2}{3}} - 1\right)}$$

Where L_t states for tube length.

From these three non-dimensional numbers, water convective coefficient can be finally calculated, since the Nusselt number is also equal to a combination of convective coefficient and water principal parameters properties.

$$h_{H_2O}\left[\frac{W}{m^2 °C}\right] = \frac{Nu_{H_2O} * \lambda_{H_2O}}{D_{in}}$$

When it comes to **flue gases** side, their inlet and outlet pressures and temperatures are known, as well as their velocity and the heat exchanger dimensioning.

For fluids that pass externally respect to the tubes, calculation pressure and temperature are simply the arithmetic mean of inlet and outlet pressures or temperatures respectively.

$$p_{calc_{FG}} = \frac{p_{in,FG} + p_{out,FG}}{2}$$

$$T_{calc_{FG}} = \frac{T_{in,FG} + T_{out,FG}}{2}$$

As in the water case, calculation pressure and temperature can be used to obtain the principal flue gases properties, as dynamic viscosity μ_{FG}, specific heat cp_{FG}, thermal conductibility λ_{FG} and kinematic viscosity η_{FG}.

While water properties are present in the most common thermodynamic manuals, flue gases are a mixture of different components without a general rule to determine their peculiar characteristics. As a result, the flue gases principal properties has been calculated as a weighted mean of principal properties of each and every flue gases constituent, at calculation pressure and temperature, weighted on their mass percentage within flue gases themselves.

Again, Prandtl and Reynolds numbers can be calculated.

$$\mathrm{Pr}_{FG} = \frac{\mu_{FG} * c_{FG}}{\lambda_{FG}}$$

Differently from the water case, in flue gases case the characteristic dimension for Reynolds number is also a quantity that has the dimension of a length [m], and is bounded to external tube diameter, a sort of hydraulic diameter.

$$L = \frac{D_{out} * \pi}{2}$$

Moreover, their velocity must be affected by the presence of many close-range tubes and their presence cause an effect on the flue gases velocity, which vary respect to the single tube situation. As a result, the velocity is bounded to tubes vertical and horizontal distance thanks to the following non-dimensional coefficients.

$$a = \frac{s_1}{D_{out}}$$

Where s_1 indicates the flux orthogonal distance between two contiguous tubes and s_2 the flux parallel distance between two contiguous tubes.

$$b = \frac{s_2}{D_{out}}$$

These two coefficients would be useful in the Nusselt number calculations as well. The coefficient that corrects the flue gases velocity is ψ, defined as follows.

$$\psi = 1 - \frac{\pi}{4a}$$

So, the velocity with which the Reynolds number is calculated.

$$w_m = \frac{w_{FG}}{\psi}$$

As a result, the Reynolds number can be written as follows.

$$\mathrm{Re}_{FG} = \frac{w_m * L}{\eta_{FG}}$$

The calculation for Nusselt number is affected by the low amount of turbulence outside the tubes, so a laminar and turbulent Nusselt are calculated with two different formulas and are then put together in a Nusselt that is a combination of the first two.

Laminar Nusselt number is the equation for a single tube for laminar flux in presence of forced convection on cylinders.

$$\mathrm{Nu_{lam,FG}} = 0.664 * \mathrm{Re}_{\mathrm{FG}}^{\frac{1}{2}} * \mathrm{Pr}_{\mathrm{FG}}^{\frac{1}{3}}$$

Turbulent Nusselt number, on the other hand, is the equation for a single tube for turbulent flux in presence of forced convection on cylinders.

$$\mathrm{Nu_{turb,FG}} = \frac{0.037 * \mathrm{Re_{FG}}^{0.8} * \mathrm{Pr_{FG}}}{1 + 2.443 * \mathrm{Re_{FG}}^{0.1} * \left(\mathrm{Pr_{FG}}^{\frac{2}{3}} - 1\right)}$$

The two numbers can be put together in order to obtain the Nusselt number for single tube for a turbulence intensity from 6% to 10% for forced convection on cylinders.

$$\mathrm{Nu_{singletube,FG}} = 0.3 + \left(\mathrm{Nu_{lam,FG}} + \mathrm{Nu_{turb,FG}}\right)^{\frac{1}{2}}$$

The present number does not take into account the presence of multiple tubes, which proximity causes a multiple interference among flue gases velocities near and far each tube. As a result, the final Nusselt number must depend on the heat exchanger dimensioning. For this reason, a correction factor is introduced.

$$\mathrm{f_{corr}} = 1 + \frac{2}{3\mathrm{b}}$$

So, the final Nusselt number can be written as follows.

$$\mathrm{Nu_{FG}} = \mathrm{Nu_{singletube,FG}} * \mathrm{f}_{corr}$$

As in the water case, convective coefficient can be extrapolated from Nusselt number definition, using L as characteristic length.

$$\mathrm{h_{FG}}\left[\frac{\mathrm{W}}{\mathrm{m^2 °C}}\right] = \frac{\mathrm{Nu_{FG}} * \lambda_{\mathrm{FG}}}{\mathrm{L}}$$

Finally, the **heat exchanger** as a whole can be considered. The first data needed in order to obtain the number of tubes is the external global heat exchanging coefficient. It can be calculated considering the electric similarity. The tube outside which flue gases flow and within which water flows has a resistance to heat transfer, and its opposite is the material thermal conductibility (in this case steel thermal conductibility λ_{steel}). Other resistances are provided by fouling coefficients, in other words they are numbers, one for water side and the other for flue gases side, which take the fouling provided by both fluids into account (in the case of water it could be lime-scales deposition), indicated respectively as $Foul_{H2O}$ and $Foul_{FG}$. Finally, heat from water and flue gases is exchanged by the convection phenomenon, which is characterised by the two convective coefficients calculated above. As a result, the formula for the global thermal exchanging coefficient must take all of these parameters into account.

$$U_{ext}\left[\frac{W}{m^2{}^\circ C}\right]=\left(\frac{1}{h_{FG}}+\frac{\ln\left(\frac{D_{out}}{D_{in}}\right)}{2*\frac{\lambda_{steel}}{D_{out}}}+\frac{Foul_{H_2O}}{\frac{D_{in}}{D_{out}}}+Foul_{FG}+\frac{1}{h_{H_2O}*\frac{D_{in}}{D_{out}}}\right)^{-1}$$

Fouling factors and steel thermal conductivity are provided by Thermoflex.

The mean ΔT logarithmic is important as well for the conclusion of the method. It is the difference in Kelvin that best represents a “mean” temperature gap between water and flue gases during heat exchange. Its calculation depends on the heat exchanger nature. In particular, if it operates in direct current or inverse current. In other words, the direct current heat exchanger sees the cool water facing the hot gases, while the hot water faces the cool gases. On the other hand, if the heat exchanger operates with inverse current, water flow is opposite to flue gases flow, so hot water faces hot gases and cool water faces cool gases. Both calculations for direct and inverse current heat exchangers would be shown.

$$\Delta T_{ml,DC}[K]=\frac{\left(T_{in,FG}-T_{in,H_2O}\right)-\left(T_{out,FG}-T_{out,H_2O}\right)}{\ln\left(\frac{T_{in,FG}-T_{in,H_2O}}{T_{out,FG}-T_{out,H_2O}}\right)}$$

$$\Delta T_{ml,IC}[K]=\frac{\left(T_{in,FG}-T_{out,H_2O}\right)-\left(T_{out,FG}-T_{in,H_2O}\right)}{\ln\left(\frac{T_{in,FG}-T_{out,H_2O}}{T_{out,FG}-T_{in,H_2O}}\right)}$$

Knowing the heat quantity exchanged between flue gases and water and reversing its definition, the external exchange area is obtainable. Even in this case, heat quantity is provided by Thermoflex and is, thus, known.

$$A_{ext}\,[m^2]=\frac{Q_{exchanged}}{U_{ext}*\Delta T_{ml}}$$

Finally, reversing the definition of external exchange area, the number of tubes can be found.

$$N_{tubes}=\frac{A_{ext}}{\pi*D_{out}*L_t}$$

The number obtained in the present equation would be confronted with the one calculated by the program in Chapter 7.

4.6.COSTS CALCULATION

Thermoflex does not calculate the cost of the radiative section of the combustor. As a result, for the radiative evaporator only, costs from a different simulated plant has been taken into account. The different plant is alimented by the same fuel and produced the same power (200 MW). Moreover, evaporation pressure are the same as the innovative configuration plant, so the radiative sections can be considered reasonably similar and so their prices.

The price of the actual radiative section is calculated using the proportion below.

$$\frac{\text{waterwall equipment cost (IP)}}{\text{(convective section + waterwall) equipment cost (IP)}} = \\ = \frac{\text{waterwall equipment cost (B)}}{\text{(convective section + waterwall) equipment cost (B)}}$$

Where the convective section cost of the specialised equipment is provided by Thermoflex, which obviously does not comprehend the waterwall cost, which is unknown. The *ip* abbreviation indicates the innovative plant data while the data with the *b* abbreviation are related to the benchmark plant from which data have been collected. The cost of the whole boiler can be then calculated, calculating the cost of the waterwall for the innovative configuration plants (since all other data are known).

The plant from which radiative evaporator costs have been extrapolated shows additional costs (in other words different from the mere cost of the component, such as civil costs, mechanical costs, etc.) grouped for the entire boiler. Knowing the cost of the actual boiler comprehensive of the waterwall cost, they can be re-proportioned to the whole boiler cost of the innovative plant.

$$\frac{\text{(convective section + waterwall) equipment cost (IP)}}{\text{(convective section + waterwall) equipment civil/mechanical/electrical cost (IP)}} = \\ = \frac{\text{(convective section + waterwall) equipment cost (B)}}{\text{(convective section + waterwall) equipment civil/mechanical/electrical cost (B)}}$$

Using the proportion above. Then, knowing the cost of the only radiative section and the cost of the entire boiler, every additional cost have been then re-proportioned to the only radiative section based on the following proportion:

$$\frac{\text{(convective section + waterwall) equipment civil/mechanical/electrical cost (IP)}}{\text{waterwall equipment civil/mechanical/electrical cost (IP)}} = \\ = \frac{\text{(convective section + waterwall) equipment cost (IP)}}{\text{waterwall equipment civil/mechanical/electrical cost (IP)}}$$

Finally, all the costs related to waterwall, comprehensive of the specialised equipment cost, are summed up and added to the final plant total cost.

Obviously, this procedure is far from being accurate, because the scale effect is neglected in this “proportional” method and the costs are hypothesised to be equally subdivided among all the boiler components. Unfortunately, not having other available data, this is the best procedure in order to obtain the required radiative section costs.

Moreover, while radiant waterwall has been calculated taking the Inconel and refractory coverage costs into account, the convective section costs calculated by Thermoflex do not include the Inconel coverage for the first super-heater and re-heater in parallel. Their cost would be calculated starting from the total external exchange area (of both *RH3* and *SH2*), obtain with the following simple formula:

$$\begin{aligned}&\text{Inconel cladded area } [\mathrm{m}^2] = \\ &\qquad = \pi * \text{external tube diameter} * \text{tube length} * \text{number of tube rows} * \\ &\qquad * \text{number of tubes per row}\end{aligned}$$

Where the first two data are known from the hypothesis, both of them are in fact deduced from the project documents of the Turin waste-to-energy plant, while the number of tube rows and tubes per row are calculated by the programme as the main output in heat exchanger dimensioning, visible in Chapter 7.

The cost of Inconel cladding, specified in the hypothesis chapter allows calculating the total Inconel cladding cost. In particular, the following simple formula is adopted:

$$\begin{aligned}&\text{Inconel cladding costs [USD]} = \\ &\qquad = \text{cost of Inconel cladding} \left[\frac{\mathrm{USD}}{\mathrm{m}^2}\right] * \text{Inconel cladded area } [\mathrm{m}^2]\end{aligned}$$

The reference plant presents two super-heaters, so the cladding has been applied only on the first heat exchanger, which is the only parallel one. The specialised equipment cost multiplier equals to 1.05% would then multiply this cost in order to take labour and operations and maintenance costs into account. These costs would be added to the esteemed costs of the overall boiler. The outcome would then be added to the costs of the overall plant calculated by Thermoflex.

Finally, calculations about break-even electricity cost must be computed thanks to the methodologies used in [9]. Knowing the net power produced by every plant (reference and all the four configurations for innovative plant) as well as their costs, the differential cost per MWh produced by innovative plants respect to the reference one can be easily calculated.

The extra power produced can be calculated with the formula below:

$$\begin{aligned}&\text{Extra power [MW]} = \\ &= \text{Power generated with reference plant [MW]} \\ &- \text{Power generated with innovative plant [MW]}\end{aligned}$$

Then, in order to obtain the extra gigawatt per hours produced, an operative time of 7800 h/year are assumed:

$$\text{Extra electricity generated in 1 year [MWh]} = \text{Extra power [MW]} * 7800 \text{ [h]}$$

With the same procedure, the extra capital cost can be calculated:

$$\begin{aligned}&\text{Extra capital cost [MUSD]} = \\ &= \text{Capital cost of reference plant [MUSD]} \\ &- \text{Capital cost of innovative plant [MUSD]}\end{aligned}$$

Later, capital carrying charge and investment for operation and maintenance are assumed to 15% and 4% respectively of the extra capital cost. The assumption derives from the most common values assumed in the practise of costs calculations.

Therefore, the final extra cost would be the following:

$$\text{Extra cost [MUSD]} = \text{Capital capital cost [MUSD]}(\text{Assumed CCC [\%]} + \\ + \text{Assumed O\&M [\%]})$$

Finally, the break-even value of electricity is calculated dividing the extra electricity generated in one year with the extra cost:

$$\text{Break} - \text{even value of electricity } \left[\frac{\text{USD}}{\text{MWh}}\right] = \frac{\text{Extra cost [MUSD]} * \text{1E6}}{\text{Extra electricity generated in 1 year [MWh]}}$$

Chapter 5

Innovative configuration with reheat

This chapter aims to analyse the innovative configuration for the waste-to-energy plant with reheat. In particular, a detailed analysis of plant components would be conducted as well as a focus on the innovative elements of the plant that allows increasing its efficiency without increasing its complexity.

5.1. INTRODUCTION

A waste-to-energy plant with reheat and advanced parameters is represented in **Figure 5. 1**.

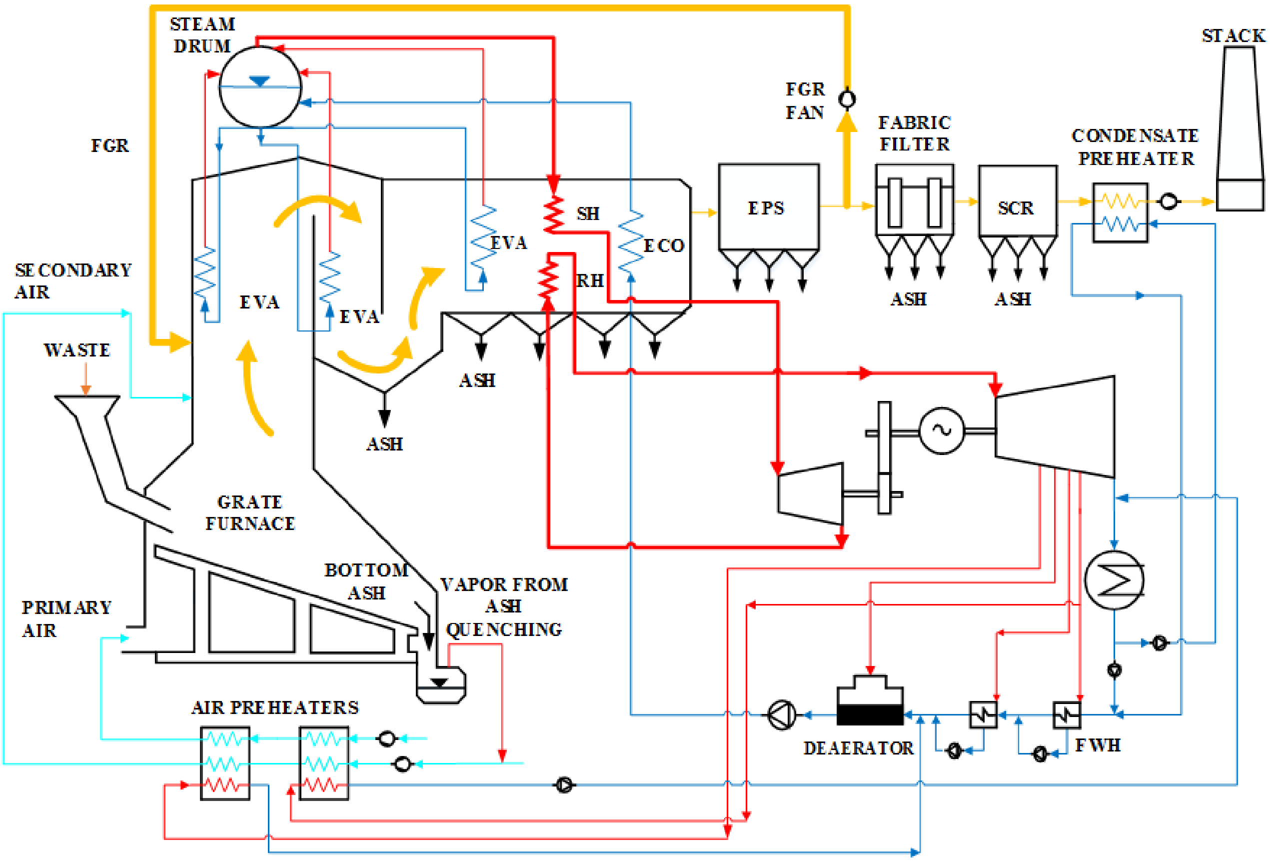


Figure 5. 1 - Advanced waste-to-energy plant with reheat [9]

Yellow lines indicate flue gas, red lines stand for vapour and blue lines for water. Light blue lines indicate air. Sub-cooled water enters the economiser, becoming saturated water. It enters the cylindrical body and is heat up to saturated vapour within the evaporator. Passing trhough the super-heater, it becomes superheated vapour and is transferred to the high-pressure turbine, where it expands to the reheat pressure. Then, it transfers to the re-heater where it is heated up at about superheating temperature. Superheated vapour enters, thus, the low-pressure turbine, where it expands to the condensation pressure. It condensates in the condenser, where it transforms from saturated vapour with high vapour quality, to saturated water. Water is then divided into two flows, at condenser exit: the first flow is heat up to deaerator's temperature by flue gases in the condensate preheater, while the second flow is heat up within regenerators, which extract saturated vapour from the low-pressure turbine body to heat up the water. The most convenient procedure, from the efficiency viewpoint, would be to heat up all the water mass flow with flue gases, but their temperature is not usually high enough to heat up at the sufficient temperature all the water flow, so the remaining water has to be warmed up by spilled vapour. The overall water mass flux enters then the deaerator, where it joins a small part of spilled vapour. It exits as saturated water at the deaerator temperature. A pump raises water pressure and sends it to the economiser. Moreover, a small part of vapour is spilled from the turbine to heat up air before it enters the boiler. Hot primary air enters then the boiler together with waste. The latter is deposited on a grate and burners start the pyrolysis. The real combustion starts over the grate, where secondary air is emitted. Combustion creates flue gases and ash, which can be subdivided in bottom ash, which is discarded at the end of the grate and cooled down before being moved to special treatment areas, and fly ash, which are in suspension within flue gases and would need to be removed in a later stage. Flue gases travel through the boiler, transferring their heat to heat exchangers, warming up the water and/or vapour. Then they enters the ESP (Electro Static Precipitator), which aims to reduce the pollutants quantity (solid or gaseous) within flue gases. A part of flue gases is extracted from the primary flow, to be recirculated within the boiler (FGR – Flue Gas Recirculation). Cooled, relatively inert, recirculated flue gases act as heat sink for the flame and lower peak flame temperatures. Moreover, when mixed with the combustion air, recirculated flue gases lower the average oxygen content of the air, starving the NO_x-forming reaction for one of the ingredients they need. Flue gases pass through other flue gas treatment systems (Fabric Filter and SCR – Selective Catalytic Reduction), which aim to reduce pollutants and ash content under the limits fixed by law. They enters the condensate preheater, heating up the condensate and are then discharged through the stack. However, as explained before, superheating and reheat sections are critical when it comes to corrosion. Consequently, superheating (and reheat) temperature cannot be raised to obtain an efficiency increase without adopting some alternative technologies.

5.2. THE INNOVATIVE CONFIGURATION: PLANT SCHEME

The innovative plant configuration proposed in the present thesis is shown in the **Figure 5. 2** below.

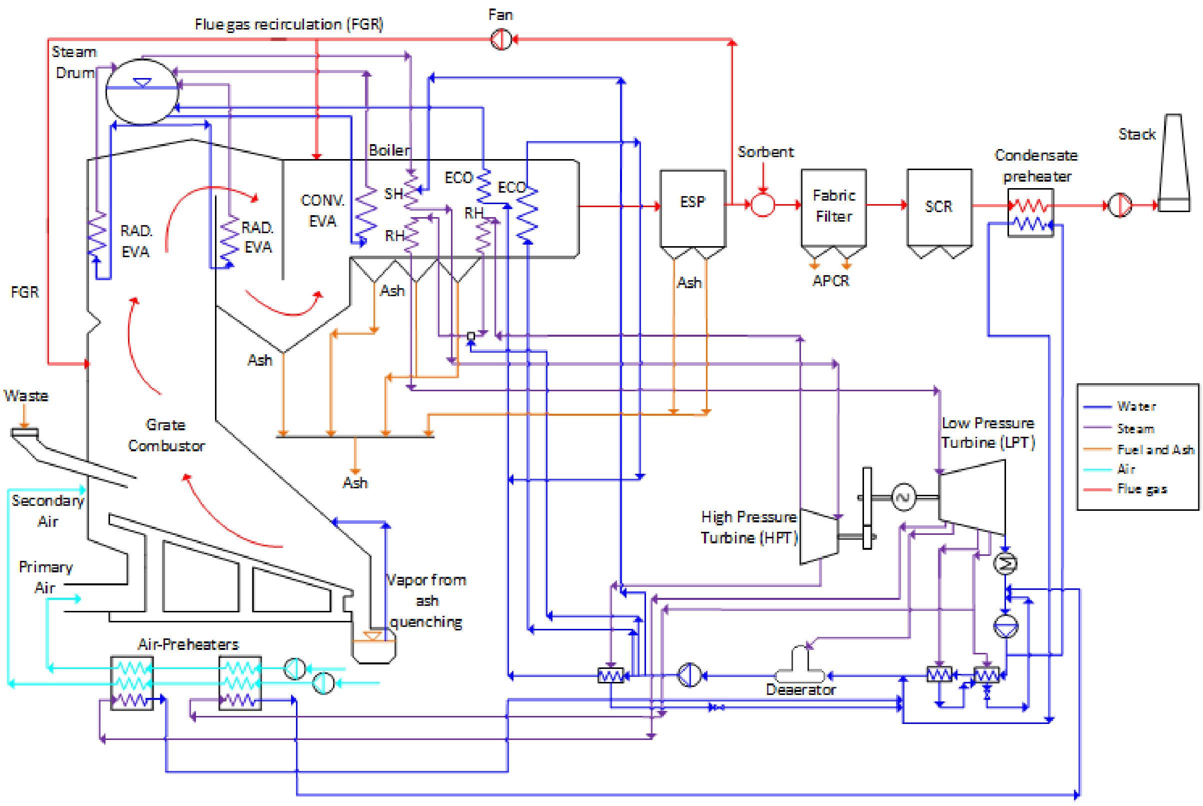


Figure 5. 2 - Innovative waste-to-energy plant configuration

The configuration is pretty similar to the advanced waste-to-energy plant configuration shown before, but with important differences. The flux at the exit of the deaerator, in fact, is split in two fluxes: the first one enters the high-pressure regenerator and the second one enters the boiler through an economiser, which heats up the water to the high-pressure regenerator temperature. This expedient aims to recover more heat from flue gases then the previous configuration, leaving at spilled vapour to heat up the remaining sub-cooled water. The second flow is then reunited with the first one before entering the warmer economiser. However, the greater difference between the two plants is the **gas quench** technology.

5.3.GAS QUENCH TECHNOLOGY

5.3.1. The problem: corrosion

The increase in temperature and pressure of the vapour generated within the boiler assures relevant thermodynamic benefits, but entails superior costs, only justifiable with size increase, due to the necessity to adopt tubes resistant to hot corrosion for the super-heaters (as explained in the previous chapter). This problem is principally attributable to the chlorine presents in chlorinated plastics (PVC) and in sodium chloride (NaCl) contained in urban waste. In fact, it reacts with alkali (Na, K, Ca) and with metals (Fe, Zn, Cd, Pb, Hg), in addition to be present in flue gases as hydrochloric acid (HCl), originating eutectic mixtures which results highly corrosive, being liquid even at low temperature.[1]

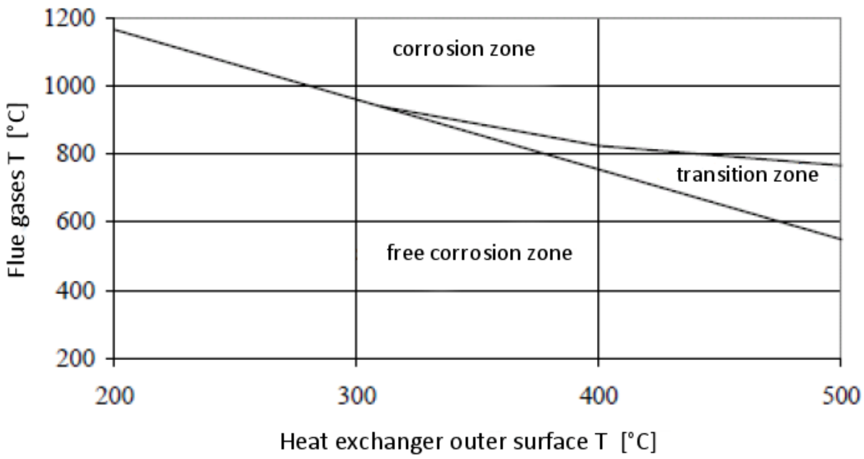


Figure 5. 3 - Hot corrosion as function of heat exchanger's surface temperature (carbon steel) and of flue gases temperature which skim it [1]

Figure 5. 3 shows the dependence of hot acid corrosion on outlet surface temperature of heat exchanger's tubes (which are supposed to be in carbon steel) and on the one of flue gases that skim it. The problem does not interest, generally, the evaporator. In fact, being the evaporation pressure maximum of 170 bar, vapour does never exceed 350°C within the evaporator. Consequently, even considering the optimum heat exchanging coefficients of water in phase transitioning, the outlet tube's surface temperature is under 360°C. Taking into account that flue gases temperature maintains itself under 1000°C, the diagram shows that evaporator would be subjected to corrosion. However, evaporators are often realised in Inconel, a costly nickel-chromium alloy, corrosion resistant, which raises the temperature limit from corrosion and free corrosion zone.

Super-heaters situation, however, is more problematic. In fact, hot vapour flows within them, characterised by small heat exchanging coefficients. Consequently, outlet tube's surface temperature exceed of (50-60) °C the one of the internal fluid, being around 500°C, which, associated to 930°C of flue gases (considering the most critical plant, the one with evaporation pressure 170bar), collocates this heat exchanger in the corrosion zone. Generally, the achievement of vapour temperatures higher than 400°C requires the employment of costly high-content of Nickel and Chromium alloys, which does not consent, anyway, to exceed 450°C.

The problem of hot corrosion negatively influences thermodynamic performances of the plant. Firstly, it represents a limit for maximum superheating temperature. Its decrease entails an average heat exchanging ΔT increase and a consequent increase in exergy destroyed during the process. Moreover, fixed the maximum temperature of superheated vapour, an increase in its pressure makes the curve that represents the expansion in T-s plane, to move left, causing a decrease in the vapour quality of the fluid discharged by the turbine. If it is too low (<0.9), mechanical and fluid-dynamic problems display themselves in the final expansion stages. Thus, fixed the maximum temperature of superheated vapour and the minimum acceptable vapour quality, the maximum superheating pressure is determined and with it, the evaporation pressure.

Summarising, in order to guarantee a vapour vapour quality acceptable to the turbine outlet, the limitation of maximum superheating temperature translates itself in a limitation in the maximum superheating and evaporation pressure, and in an increase in ΔT of heat exchange even during evaporation.

5.3.2. The solution: gas quenching

Gas quench technology is an innovative practice that consents to cool down the flue gases exiting the radiative part of the boiler (in other words the flue gases that have passed the evaporative section). It consists in splitting the recirculated flue gases in two flows: the first one enters the boiler after being united with the vapour derived from bottom ash quenching (as in the more traditional plant configurations). On the other hand, the second one is inserted in the boiler in a later stage, before flue gases enters the convective section. The aim is to reduce flue gases temperature without increasing the waterwall dimensions (which happens in plants without gas-quench innovation because maintaining a relatively low temperature at combustor outlet is equal to require a higher quantity of water within waterwall). In fact, as explained before, flue gases temperature is critical to corrosive mixture creations. The highest temperature within the boiler is registered near the grate and in boiler's radiative section. However, these sections are covered in Inconel, which is an austenitic super-alloy steel. Inconel is an oxidation and corrosion resistant material well suited for service in extreme environments subjected to pressure and heat, as the boiler is. Inconel covering is costly, so it would be preferable to use less material as possible: consequently, only radiative section of the boiler is covered in Inconel. On the other hand, convective boiler section is not covered in Inconel, so it is more sensible to corrosion problems. In particular, the first sections (which are affected to the highest flue gases temperature) are the superheating and reheating sections and are thus the most critical. In particular, superheating (and reheating) temperature should be maintained under 650°C in order not to allow the formation of corrosive mixtures. However, decreasing flue gases superheating temperature do not allow to achieve the highest efficiency, because the minimum temperature difference which is necessary to be maintained between flue gases and water temperatures requires the water-side temperature decreasing (if flue gases temperature decreases). As explained in Chapter 2, higher the superheating temperature, higher the cycle efficiency, so it is important to raise superheating temperature without the risk of corrosion.

Gas quenching technique has exactly this purpose: injecting a part of cooled gases after evaporative section allows reducing the overall temperature of flue gases, bringing them back to the required limit temperature. The recirculated gases mass flow is obviously regulated in relation to the temperature of flue gases exiting the radiative section of the boiler in order to reach the required temperature of 650°C at the exit of radiative section. T-Q diagrams about the same plant producing the same power, with or without reheat, with or without gas quenching are shown in the **Figure 5. 4** below.

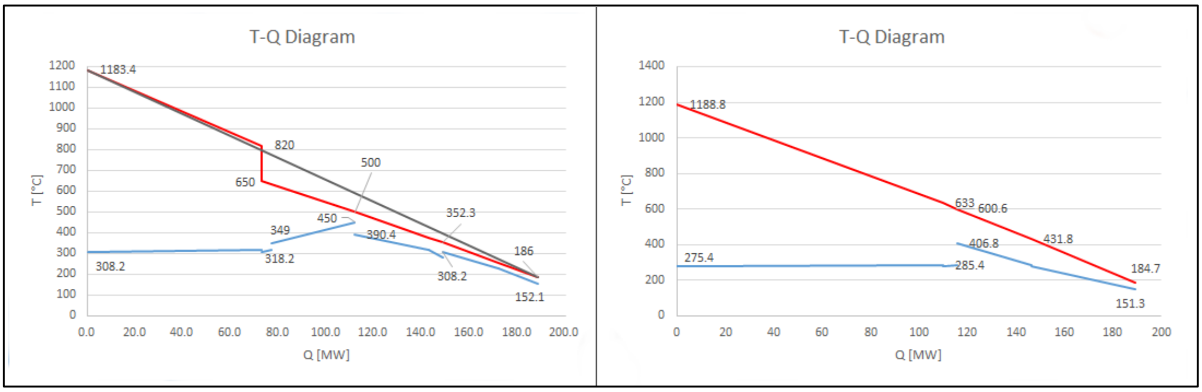


Figure 5. 4 - T-Q diagram of a waste-to-energy plant with reheat (a), showing the differences between flue gases temperature evolution with gas quenching (red line) or without it (black line). (b) shows T-Q diagram of a plant without reheat and gas quenching

Red line and black line indicate in both cases flue gases temperature evolution as a function of heat exchanged (Q), while blue line stands for water temperature evolution. Numbers within the diagrams show temperature values of gases or water in that precise point. Confronting the plant with reheat with and without gas quench, appears clear that, without gas quench, gas temperature at the convective section would be higher (820°C in the example) than the plant provided with gas quench. This would be critical, as explained before, even though a higher temperature difference would be available and so a possibility to increase water temperature and in conclusion the overall efficiency. However, corrosion problems require the flue gases temperature in convective section to be less than 650°C, so gas quench is necessary. Finally, a confrontation between a waste-to-energy plant with reheat and gas quench (red line in **Figure 5. 4 (a)**) and a waste-to-energy plant without reheat and gas quench (**Figure 5. 4 (b)**) is carried out.

If the temperature limit of 650°C at convective boiler section entrance has to be taken into account (**Figure 5. 4 (b)**), a great amount of vapour has to be produced within the boiler (in order to reach the required temperature of about 650°C, starting from a grate temperature of circa 1190°C). As a result, a higher quantity of water would pass through heat exchangers within the boiler. Since the heat exchanged among flue gases and water is the same as in the first case, the net outcome is that superheated vapour cannot be heated up to the required 450°C by flue gases because of its increased mass rate. In **Figure 5. 4 (b)**, in fact, the temperature at the exit of super-heater is shown to be 406.8°C.

In conclusion, gas quench allows to reduce corrosion problems in super-heaters and re-heaters and to reduce the quantity of vapour produced within the boiler (which leads to a reduction in waterwall overall dimensions), which leads to an increase in water temperature that can be achieved, even though water mass quantity decreases. However, it is demonstrable that a plant with reheat and gas quench has higher efficiency than a waste-to-energy plant devoid of reheat and gas quench.

5.4. PLANT DETAILED ANALYSIS

This paragraph aims to analyse the plant modelling using the software Thermoflex. The overall plant can be subdivided in the boiler and steam cycle parts. In the following lines, thus, the detailed final configuration of the plant would be explained, using the graphic of the user-interface of the programme as a guide, in order to better understand the innovative character of the configuration.

5.4.1. Steam Cycle

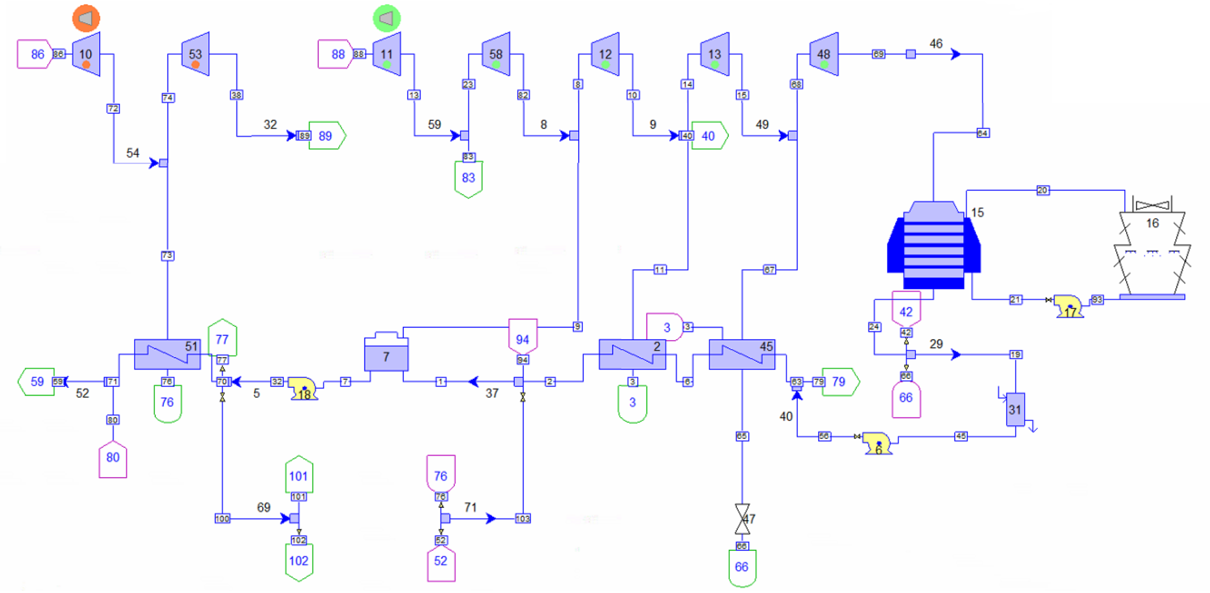


Figure 5. 5 - Steam cycle of the new waste-to-energy plant configuration

The turbines *10* and *53* schematise the high-pressure turbine body. A small part of the vapour flow rate is spilled from it in order to heat up the water in the high-pressure regenerator (*51* in **Figure 5. 5**). Superheated vapour exits the high-pressure turbine body to be reheat within the boiler (flow *89*) and then re-enters the steam cycle (flow *88*), passing through the medium and low-pressure turbine body (*11*, *58*, *12*, *13*, *48*). The first spilled flow exits the turbine in order to pass through the second (and warmer) air preheater. It re-enters in the flow *52*, entering the deaerator together with water from the two low-pressure regenerators and from the condensate pre-heater (flow *94*), as well as the subcooled water from the high-pressure regenerator (flow *78*). The second bleeding is used in the deaerator. The third is furtherly divided in two fluows: the first one goes to the second low-pressure regenerator, while the second one (flow *40*) enters the first air preheater, re-entering the steam cycle in flow *42*, after the principal stream has condensate in the condenser, together with the flow *66*, from the outlet of first low-pressure regenerator. Finally, the last bleeding is used to heat up the subcooled water within the first regenerator. Vapour condenses then within the condenser *15*, which condensates water thanks to the evaporative tower *16*. The device *31* allows maintaining constant the quantity of water in the plant. In fact, in turbine bodies, some water leakages are present, so it is important to restore the original fluid quantity. Then, a pump raises the overall pressure and water is split in two flows: the first one enters the two regenerators, which raise the water temperature to the deaerator one. The second stream (flow *79*) is sent to the condensate preheater, which also raise the water temperature to the deaerator's. This stream reunites to the principal one after the second low-pressure regenerator, before entering the deaerator (flow 1). Afterwards, the pump *18* raises the water pressure to the evaporator's. The flow is then split in three streams: the first one enters the high pressure regenerator and, after being reunited with the second one, enters the boiler (flux *59*), the second one (flow *77*) is sent to one economiser, to be heated up to the same temperature as the first flow, to which it is afterward reunited (flow *80*). As explained before, the necessity to split the stream is dictated by the impossibility, by the only flue gases, to heat up the overall water mass flow rate, which, consequently, has to be heated up partially by spilled vapour from high-pressure turbine. The third flow, on the other hand, is furtherly subdivided in two flows, which would cool down the superheated vapour entering the super-heaters in the attemperators. Finally, water is heat up to become superheated vapour and enters the high-pressure turbine body in flux *86*.

The red and green dots in the turbines indicate the different rotational velocities of the high-pressure body and low-pressure body. Plant with evaporation pressure of 170 bar and 150 bar do not need the high-pressure regenerator, which appears to be absent with this evaporation pressures because flue gases can heat up to the required temperature the overall water mass rate.

The stream cycle has no innovative character in the viewpoint of waste-to-energy plant. The real innovation is about boiler, which would be analysed in the following paragraph.

5.4.2. Boiler

The boiler is the real innovation of the waste-to-energy plant. In fact, it contains the reheat section as well as the gas quench technology, used to decrease flue gases temperature in order to reduce the corrosion risk in the convective section.

Waste (indicated with the orange line) enters the boiler *3* from the tank *4*. Air enters as well the boiler, after being preheated in the air preheaters *14* and *23*, thanks to bleedings from low-pressure turbine body. Air is extracted from an imaginary air tank (*27*, indicated as a red rectangle), even though in the real plant it would be collected from the outside using fans and tubes. Air is divided in two flows, which constitute the primary and secondary air. Red lines indicate gases, so they stand for air as well as flue gases, while blue lines indicate water and water vapour. Finally, a part of flue gases enters as well the boiler (after opportunely passing through the EPS *72*) as FGR (flue gases recirculation). The combustion occurs within the boiler and flue gases exit the top of it. The boiler *3* integrates the radiative evaporator (indicated as a small blue circle in its right corner). Water coming from the second economiser (indicated as *ECO2*) enters the radiative evaporator, after a small part of its mass flow rate has been spilled (flux *36*) and transferred to the convective evaporator (indicated as *EVA*).

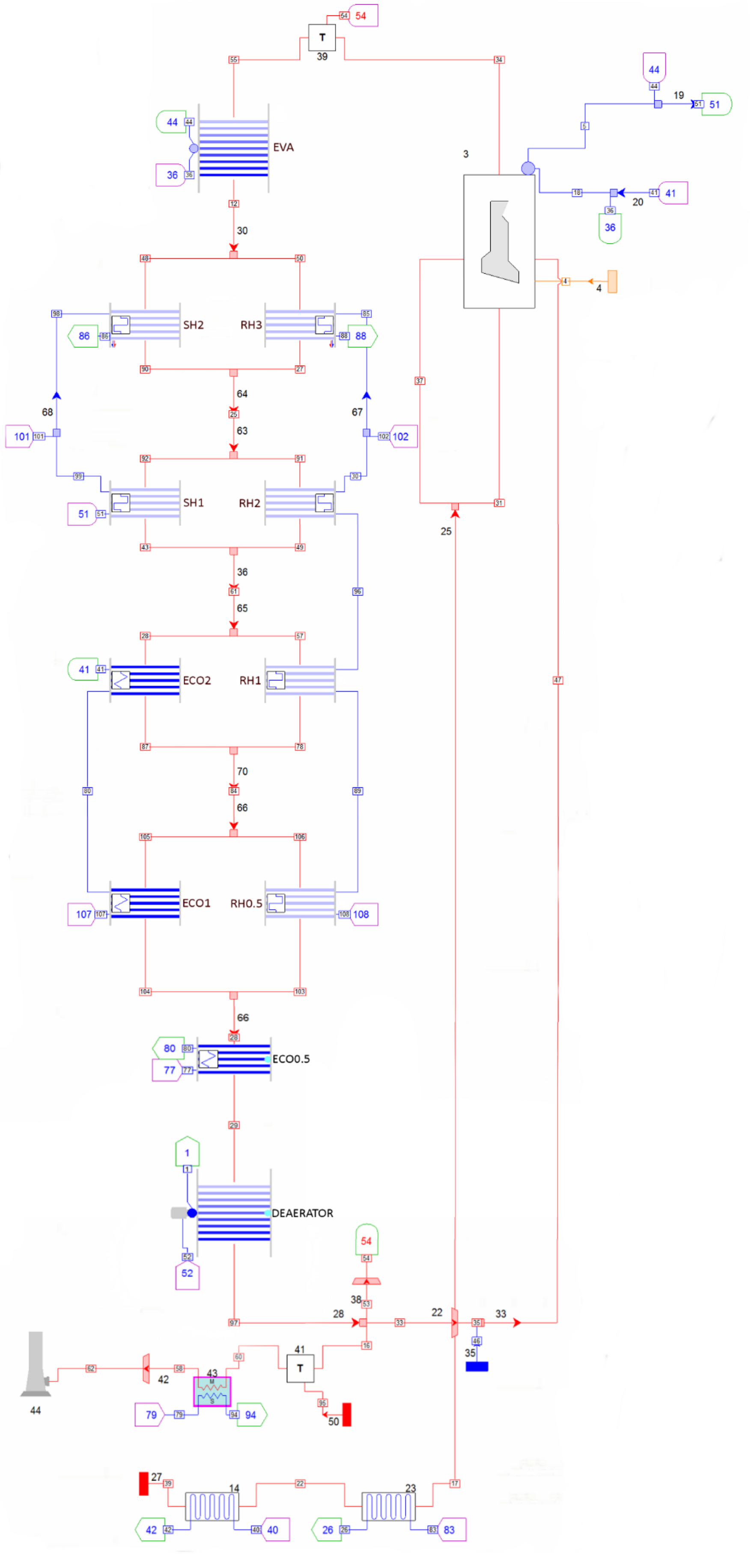


Figure 5. 6 - Boiler in new waste-to-energy plant configuration

Evaporator is actually only one, the subdivision in two section (a radiative and a convective one) is only for a simulation necessity. Evaporator is in fact heated up by the combustion partly thanks to radiation and partly thanks to convection: the schematisation wants to take both aspects into account and the only way to do so is to subdivide the water flux in two evaporators, weighting the subdivision on the quantity of heat received by radiation and convection. Evaporator heats up subcooled water to become saturated water: water from the radiative evaporator is then joined with the small flow from convective evaporator (flux *44*), forming the flow *51*, which would be sent to the superheating section. Flue gases exiting the combustor are cooled down before entering the convective evaporator thanks to the gas quenching technology: flue gases exiting the boiler, after having being filtrated, are partially recirculated (flow *54*) in order to decrease flue gases temperature. The device *39*, in fact, is a temperature stabiliser: the quantity of recirculated flue gases is established by the mass rate and temperature of flue gases exiting the boiler, in order to reach the wanted temperature (650°C in this case).

Cooled flue gases enter then the convective evaporator, exchanging heat with subcooled water to transform it in saturated water. Unlike evaporator, the other heat exchangers receive heat almost only through convection, therefore are schematised as only one device. Flue gases are then split in two flows: one exchanges heat with the second super-heater *SH2* and the second with the third re-heater *RH3*. In this case, also the flue gases subdivision is only for simulation necessity. The two heat exchangers are actually in parallel and the flue gases splitting is weighted on the necessity to guarantee the final flue gases temperature (on both ends of the heat exchangers exits) to be equal – ideally their temperature should be equal in every intermediate point before and after the heat exchangers.

Then, flue gases heat up water and vapour in *SH1* and *RH2*, *ECO2* and *RH1* and finally in *ECO1*, the only heat exchanger which is not in parallel with another one. Afterwards, flue gases are split in three flows: a part of them goes to the boiler as FGR, in a calibrated quantity (see Chapter 7 for further details). A small amount of vapour (imaginary tank *35*) is added to this first flue gases flow, after their pressure have been raised by the fan *22*, in order to simulate the vapour used in gash quenching. Thermoflex doesn't provide a boiler where ashes exits it as a material flux, even if their presence is taken into account: so the simulation requires the presence of vapour which cools down flue gases, after having cooled down ashes (necessary process for correct ashes disposal), in order to better adhere to the reality of a waste-to-energy plant. Fan *42* raises the second flux pressure and send it (flux *54*) to the gas quench device. Again, an actual gas quench device does not exist: flue gases rates is regulated thanks to temperature sensors and valve, its presence is only a simulation necessity. The third part of flue gases and the largest one is sent through the whole filtering system (while the other two streams passes only through the EPS). Filtering system presence is simulated, in addition to a pressure drop, with a temperature decrease, operated in the device *41*, which cools down flue gases mixing them with imaginary air (tank *50*).

Flue gases enters the condensate heat exchanger *43*, heating up part of the water of the steam cycle, from the condensation temperature to the deaerator's one. Finally, their pressure raises thanks to the fan *38* and they are discharged in the atmosphere through the stack.

A detailed analysis of convective heat exchangers is now provided. Subcooled water enters the *ECO1* at the deaerator temperature (flow *77*), to be heated up to the high-pressure regenerator temperature (flow *80*). Then it reunites to the principal flow, heated up within the high-pressure regenerator, entering the *ECO2* (flow *59*). *ECO1* and *ECO2* split the temperature interval from high-pressure regenerator temperature and condensation temperature minus the sub-cooling temperature gap, in order to minimise the approach point delta T (which is the difference between the temperature of cold flue gases and cold fluid within the heat exchanger).

Water exits *ECO2* at condensation temperature minus sub-cooling delta T (which is the difference between the actual temperature of water and its condensation temperature, given the pressure), and enters the evaporator (flow *41*). Then, flow *51* exits the evaporator and enters the first super-heater, *SH1*. Also in this case, the temperature interval is split between *SH1* and *SH2* in order to reduce the approach point delta T in *SH2*. A de-super-heater is present between *SH1* and *SH2*: the stream *101*, spilled after the feed pump, adds to the main flow, cooling it down. This procedure decreases efficiency, because it decreases superheating temperature, but its presence is fundamental when the plant works at partial load. In this case, in fact, water flow mass rate is lower than the full load configuration: consequently, tubes of heat exchangers become warmer and the risk to exceed the maximum consented temperature for mechanical resistance of materials becomes real, being the quantity of waste combusted the same. De super-heaters, however, allow reducing the superheated vapour temperature, decreasing the tube material temperature as well.

De-super-heaters are actually more than two as represented in this simulation, as there are more than two or three super-heaters and re-heaters. Usually these heat exchangers are split in a great number in order to reduce the gap between fluid and flue gases temperatures and every critical section has its own de-super-heater. The simulation is a good compromise between the necessity of obtaining reliable data and creating a model easily modifiable and adaptable.

Superheated vapour exits, then, *SH2*, entering the high-pressure turbine. Superheated vapour enters again the boiler in *RH1*, exiting the high-pressure turbine body. All heat exchangers try to guarantee that not only flue gases, but also water temperatures between two parallel heat exchangers are the same. Temperature interval among the four re-heaters is split in order to observe this principle in addition to the objective of reducing the approach delta T. Another de-super-heater is present between *RH3* and *RH2* (fllow *102*). Finally, flow *88*, which exits *RH3*, re-enters the stream cycle in the low-pressure turbine body.

Chapter 6

Assumptions

This chapter reports the principal calculation hypothesis used in plant modelling: they depend especially on the plant size. Disposal potentiality, in fact, can justify different choices in boiler conformation and in the part of the plant dedicated to energetic recovery. In particular, in small-size plant, construction cost must be limited, so reliability and run continuity are privileged upon electric generation efficiency. When power increases, however, scale economics on machines' costs and efficiencies assure incomes increases derived from the sale of energy produced. This justify the investment of major economic resources for plant complexion and the use of more performant materials, which lead to the increase of run conditions.

6.1. DATA ASSUMPTIONS

In **Table 6. 1** the principal data assumptions are reported. In order to better understand the thermodynamic behaviour of the new configuration for the waste-to-energy plant, the power introduced with fuel has been fixed to 200MW (which means, being fixed the waste LHV, to fix the mass flow rate of waste within the boiler), while the evaporation pressure varies from 110 to 170 bar. Moreover, even the reheat pressure varies from 10 to 50 bar.

The objective is, obviously, to estimate the better combination of evaporation pressure and reheat pressure, to reach the highest efficiency of the given plant.

Parameter	Unit of Measure	Value
Power (waste mass rate * LHV)	MW	200
Evaporation pressure	bar	110, 130, 150, 170
Reheat pressure	bar	10, 15, 20, 30, 40, 50
Condensation pressure	bar	0.07
Gas temperature at convective evaporator inlet	°C	650
Steam temperature at SH outlet	°C	450
Steam temperature at RH outlet	°C	450
Gas temperature at ECO outlet	°C	≈ 185
Gas temperature outlet condensate preheater	°C	130
Gas temperature inlet condensate preheater	°C	175
Temperature of primary/secondary air	°C	165
Deaerator temperature	°C	150
ΔT sub-cooling	°C	10
ΔT approach point	°C	min 50
Flue gases velocity at super-heater/re-heater inlet	m/s	4.5
Number of low-pressure regenerators	-	2
Number of high-pressure regenerators	-	1
Number of air preheaters	-	2
Number of de-super-heaters	-	2

Percentage of total water flux rate to de-super-heaters	%	10
Breakdown of water mass flow among de-super-heaters	%	50
Percentage of total water flow rate to be heated up in the convective evaporator	%	5
Oxygen content at boiler outlet (humid gases)	%	6

Table 6. 1 – New configuration thermodynamic values assumption

The next **Table 6. 2**, on the other hand, shows the principal data assumptions characteristic of the confrontation plant.

Parameter	Unit of Measure	Value
Power (waste mass rate * LHV)	MW	200
Evaporation pressure	bar	70
Condensation pressure	bar	0.07
Gas temperature at convective evaporator inlet	°C	max 650
Steam temperature at SH outlet	°C	450
Gas temperature at ECO outlet	°C	≈ 185
Gas temperature outlet condensate preheater	°C	130
Gas temperature inlet condensate preheater	°C	175
Temperature of primary/secondary air	°C	165
Deaerator temperature	°C	150
ΔT sub-cooling	°C	10
ΔT approach point	°C	min 50
Flue gases velocity at super-heater/re-heater inlet	m/s	4.5
Number of low-pressure regenerators	-	2
Number of air preheaters	-	2
Number of de-super-heaters	-	1
Percentage of total water flux rate to de-super-heaters	%	5
Percentage of total water flow rate to be heated up in the convective evaporator	%	5
Oxygen content at boiler outlet (humid gases)	%	6

Table 6. 2 - Confrontation plant thermodynamic values assumption

The programme establishes blowdown as well as the value of pressure and temperature drop as the sum of every blowdown, pressure and temperature drops in each end every component. In turn, they are inserted in the component's datasheet within the programme, not modifiable by the user. Therefore, they are fixed and non-modifiable.

Moreover, the programme calculates the quantity of flue gases destined to be recirculated for gas quench purposes. In fact, the temperature of flue gases exiting the boiler is fixed by the necessity to guarantee the water required temperatures within the evaporator and the flue gases' at the outlet of the economiser. Since the temperature of flue gases at convective evaporator inlet is a requirement, the quantity of flue gas recirculated in order to cool down flue gases exiting the boiler is known.

Gas temperature at last economiser outlet is fixed to 185°C in order to guarantee the correct operation of flue gas treatment system.

6.1.1. Pressure and temperature of generated vapour

As explained in the previous chapter, pressure and temperature are limited to corrosion problems and turbine fluid-dynamic. Gas quench reduces gas temperature to 650°C (which, according to **Figure 5. 3** of the previous chapter, is the limit to avoid corrosion in the super-heater), consenting to avoid the use of costly alloys for heat exchanger construction. On the other hand, evaporator can take advantage too from gas quench, because the flue gases temperature allow to avoid the Inconel employment, at least in the convective part. Obviously, the reference plant, which is not provided with gas quench, needs to produce flue gases at maximum 650°C in order to avoid hot corrosion in the first heat exchanger. In particular, being the evaporation pressure 70 bar, the combination of the temperature of water and consequently of the outer tube's surface wall and of the flue gases exiting the boiler is too low to incur corrosion. On the other hand, being the superheating temperature the same as the new configuration plant, the reference plant meets the same problems as the new plant.

New configuration plant, other than implementing an innovation to overcome the corrosion problem, presents borderline parameters in terms of materials resistance. The aim, in fact, is to analyse the best performances reachable with the innovative plant, therefore the highest and better value to achieve so must be considered. Obviously, all parameters values consider the material resistance and are plausible for the considered overall power of the plant.

6.1.2. Condenser

In every plant, the condenser is supposed to be cooled with an evaporative tower. This solution, even though entails a minor cooling efficiency respect to a system with flowing water, consents to release the plant location from the presence of an adequate stream.

This entails that the pressure and temperature of condensate results equal for all the plants and parameters combination considered.

The programme considers a scale effect on performances of condenser auxiliaries. Growing the thermal power to dispose of, fans size is hypnotised to grow too. Electric power absorbed by auxiliaries dedicated to condenser refrigeration grows with the fluid flow rate, expressed as percentage of thermal power took away from it.

6.2. THERMODYNAMIC ASSUMPTIONS

6.2.1. Water preheating line

Scale economy associated to plant performances increase allows the water preheating line to result more performant. This implies that only deaerator is present for low electric power produced. It receives vapour spilled from turbine in order to bring the water to about 120°C.[3]

For power higher than 10 MW (which is the case of the plants presented in the thesis), the higher costs associated to the presence of a condensate preheater in parallel with two low-pressure regenerators is present. In the innovative configuration waste-to-energy plant, with evaporation pressure of 110 bar and 130 bar, a high-pressure regenerator is also present, in parallel with an economiser within the boiler. This consents to obtain warmer water (150°C without high-pressure regenerator, 240°C with high-pressure regenerator) in order to reduce the mean ΔT of heat exchanging and, consequently, the exergy losses within the boiler.

The thermal gap between the condenser outlet and boiler inlet is approximately subdivided in two (or three) equal parts, because the deaerator operates only a gap of 5°C, so it cannot be considered as a "part" of the temperature gap division (**Figure 6. 1**). Pressure of the vapour spilled from the turbine for regenerators and deaerators are defined as consequences.

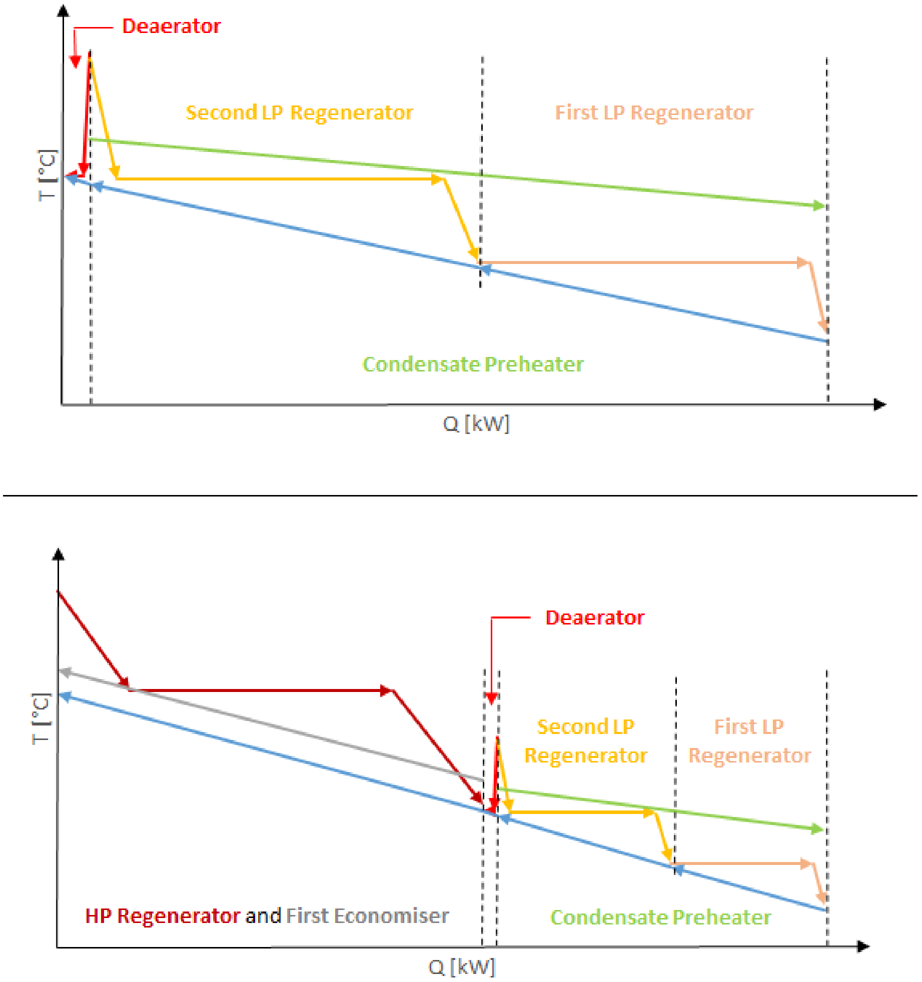


Figure 6. 1 - T-Q diagrams referred to water preheating line

The first diagram shows the situation of the comparative plant as well as the new configuration plant with evaporation pressure of 150 bar and 170 bar, while the second diagram is characteristic of new plant with 110 bar and 130 bar as evaporation pressures.

6.2.2. Flue gases temperature at the stack

In small-size plants, heat recovery downline of the economiser is not considered for costs reasons: flue gases temperature is thus not imposed within the calculation scheme, but it depends on the one required from the post-treatment line. On the other hand, for high-size plants, condensate preheater is used, which permits to exploit a further fraction of thermal energy contained in flue gases for water preheating. In these cases, flue gases temperature at the stack is about 130°C.

6.2.3. Air preheaters

Plant dimensions increase justifies the adoption of a more efficient air preheating line. For this reason, while for smaller sizes the thermal gap is covered in only one heat exchanger, in higher power plants, the exergy destroyed during the process is minimised using two preheaters, alimented by vapour fluxes from the turbine. Bleeding pressure as well as temperature interval distribution are defined in order to assure reasonable temperature differences between the air and the vapour within the heat exchanger and to avoid the risk of intersection between the two curves.

Figure 6. 2 shows the characteristic T-Q diagram of the two preheaters.

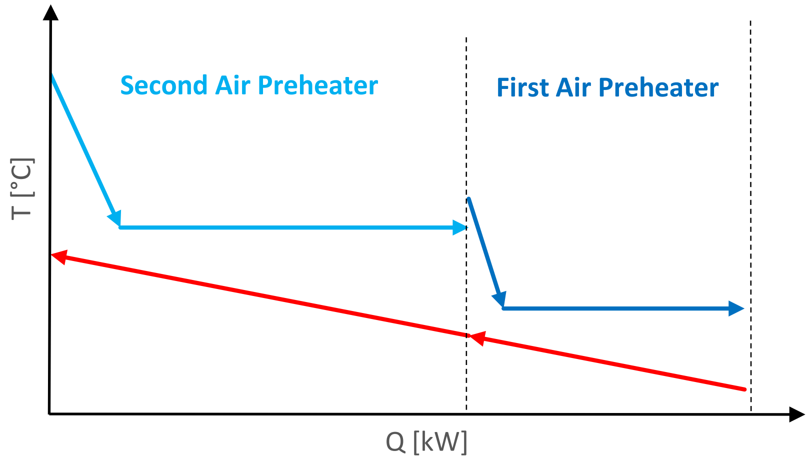


Figure 6. 2 - T-Q diagram related to air preheaters; hot air will then go within the boiler

The minimum temperature difference between air (indicated in red) and water (indicated in blue), registered at the very start of vapour condensation in both cases, is fixed to a value of 30°C.

The adoption of an efficient air preheater line in higher-size plant allow to increment the temperature of the air sent to the combustor, in order to raise the energy contained with flue gases and, consequently, vapour flux rate generated within the boiler.

6.2.4. Quantity of oxygen in flue gases

The presence of oxygen in flue gases is principally due to the necessity to use a major air quantity than the stoichiometry, in order to assure the complete oxidation of fuel within furnace. In high-size plants, a reduction in oxygen percentage can be achieved, because higher costs caused by the increase in furnace complexion are justified.

Quantity of oxygen in flue gases at the stack increases due to seeping of air presents in post-treatment line (or filtering line), due to the slight depression that characterise them.

Value indicated in the **Table 6. 1** and **6. 2** is the typical one for a waste-to-energy plant. Oxygen content in flue gases at the stack is not indicated because, as clarified in the previous chapters, flu gas treatment system is of little interest in the present simulations, so its flue gases parameters has not been taken into account, even though its presence is compulsory for cost calculations reasons.

6.2.5. Heat losses in heat exchangers and combustor

For every heat exchanger presents in the plant Thermoflex assumes 1.5% of heat given to hot fluid to be dispersed in the environment. Thermal loss in combustion chamber, essentially caused by irradiation, are assumed to be 2.5% of the combustion heat, calculated on the LHV.

6.2.6. Slag's extinguishing vapours

Slags are discharged from the grid furnace in an extinguishing bath constituted by a tank full of water, which operate also as a hydraulic seal for the combustor. The mean discharge temperature for the slags is about 350°C. They fall within the water and cause its heating and its partial evaporation. Due to this loss of water, extinguishing bath must be continually reintegrated. The **Figure 6. 3** below shows the scheme of slags extinguishing system.

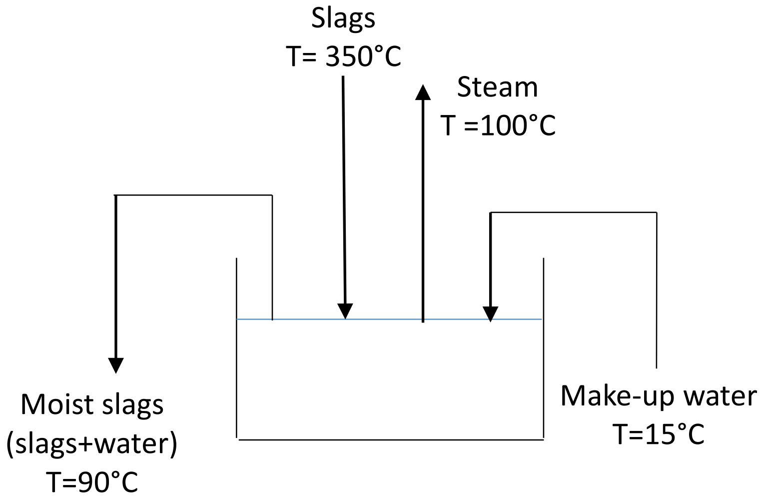


Figure 6. 3 - Slags extinguishing system

Moreover, the extraction of slugs from the bath causes the extraction of a small part of water still in liquid form. This quantity of water loss can be esteem assuming that the material removed from the bath has a humidity percentage of 20% weight.

The quantity of water exiting the furnace within slags can be calculated following the simple equation below.

$$(1 + x)0.20 = x$$

Where x indicates the quantity of water exiting the furnace with slags. (1+x) represents the mass exiting the boiler, where 1 is the kilograms of slags and x the quantity of water. Thus, the final water quantity is obtainable reversing the formula.

$$x = \frac{0.20}{1 - 0.20} = 0.25 \text{ kg}$$

The remaining part of make-up water (constituted by the mass of water removed with the slags and the evaporated water during slags extinguishing) is the one that evaporates, producing vapour, which is normally re-inserted within combustion chamber.

Knowing the water exiting the system with ash, the evaporated make-up water can be calculated with the following energy balance.

$$\begin{aligned} &\dot{m}_{slag} c_{p,slag}\left(T_{in,slag} - T_{ref}\right) = \\ &= \dot{m}_{slag} c_{p,slag}\left(T_{out,slag} - T_{ref}\right) + \dot{m}_{H_2O} c_{p,H_2O}\left(T_{out,slag} - T_{ref}\right) \\ &+ x c_{p,H_2O}\left(T_{out,steam} - T_{ref}\right) + x\Delta h_{vap} \end{aligned}$$

Where the reference temperature has been chosen as 15°C in order to avoid the term constituted with the make-up water quantity. *x* indicated the quantity of vapour vaporised and then added to flue gases recirculated within the combustor. Finally, Δh_{vap} indicated the enthalpy difference between saturated water and vapour at 100°C.

The resultant quantity of vapour added to flue gases before entering the combustion chamber is then 0.069 kg per kilogram of bottom slags discharged. In the Thermoflex plant, this is schematised as a water source (since the software does not provide ashes and slags as physical fluxes), which add vapour at 100°C to flue gases, calibrated on the quantity of bottom slags exiting the boiler.

6.2.7. Flue gasses recirculation in combustion chamber (FGR)

Part of flue gases downstream the boiler are recirculated in the combustor in order to respond to two main purposes:

- Control the combustion chamber temperature to reduce the nitrogen oxide formation and the ash softening/sintering within waste. Usually, where primary and secondary air within combustor are pre-heated (as the case of the present thesis), 15% of flue gas mass flow rate exiting the boiler is recirculated. In presence of flue gases recirculated, however, the quantity of flue gases exiting the boiler depends on the quantity of recirculated gases, which in turn depends on the flue gases exiting the boiler. As a result, the same calculation adopted for water expelled within slags has to be applied.

  $$(1 + x)0.15 = x$$

  Where x indicates the quantity of flue gases to be recirculated and (1+x) the quantity of flue gases exiting the boiler (where 1 represents flue gases produced by the combustion process and x recirculated flue gases).

As a result, reversing the equation, gives that the quantity of flue gases recirculated is equal to 0.176 kg/s per one kilograms per second of flue gases produced by combustion process. The Thermoflex schematisation considers this requirement at the splitter *28*.

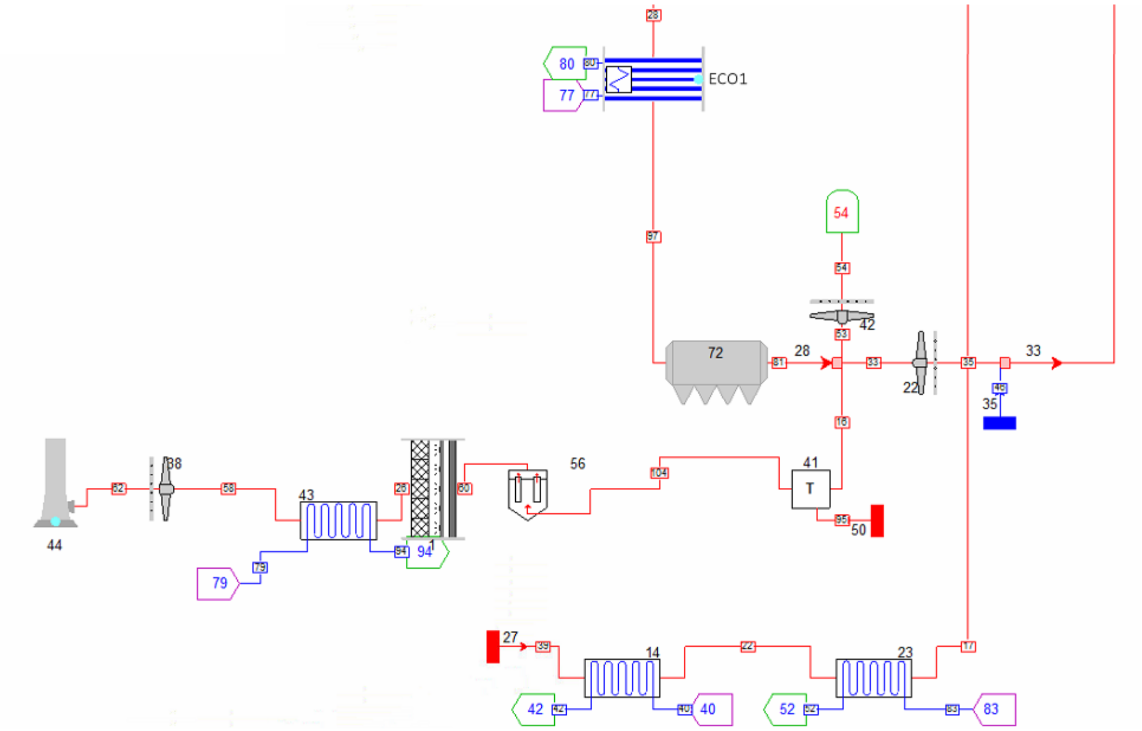


Figure 6. 4 - Flue gases splitting

Flow exiting the mixer *33* has to be the 15% of flow exiting the boiler (which is different for flue gases entering the splitter *28*). Therefore, the flow *33* has to be the 15% of flue gases exiting the boiler minus the quantity of vapour added for slags extinguishing purposes. Flow *53* (and *54*) is the required one for diminishing flue gases temperature from their temperature at combustor outlet to 650°C as required by data hypothesis. The remaining flue gases quantity (flow *16*) is the sent to heat up water in the heat exchanger *43* and then discharged through the stack.

Therefore, supposing 1 kilogram of flue gases produced by the combustion process, 0.176 kilograms minus the quantity of vapour for slags extinguishing is the quantity of flue gases at flux *33*, while they become 0.176 kilograms at the exit of mixer *33*. One kilogram is discharged in the air through the stack, plus the small quantity correspondents to the quantity of vapour emitted in recirculated flue gases, coming from slags extinguishing bath and x is the quantity of flue gases for gas quenching purposes. Flue gases entering the splitter *22* is (1.176+x). The scheme below clarify the situation explained.

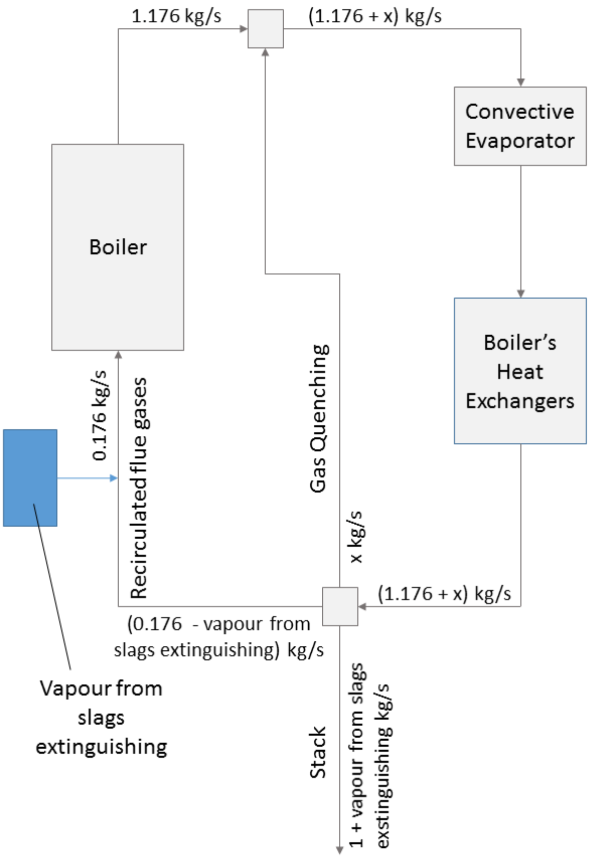


Figure 6. 5 - Flue gases recirculation scheme

- Provide the gas quench at convective section entrance. It is a second flue gases recirculation finalised to decrease flue gases temperature to 650°C, retained compatible with the hot corrosion resistance provided by alloys usually adopted (Inconel 625).

6.2.8. Fuel composition

Fuel combustion is assumed from the paper "Assessing the performances of energy recovery from waste" by Prof. Eng. Stefano Consonni and Eng. Federico Viganò, presented on the 9th edition of SIDISA (International Symposium of Sanitary and Environmental Engineering) [10]. It considers the URW_{REF}, which is the urban residual waste taken as reference for further comparisons. The waste composition hypothesised for the plant analysed in the thesis is the URW_{REF} one and is presented in the **Table 6. 3** below.

Component	**Value [weight %]***
Total Moisture	33.37
Ash	16.46
Carbon C	27.59
Hydrogen H	4.23
Nitrogen N	0.67
Chlorine Cl	0.26
Sulphur S	0.04
Oxygen O	17.38

Total	100.00

*The original composition included also 0.004% of F, but the Thermoflex fuel tank interface don't contemplate its presence

Table 6. 3 - Fuel ultimate analysis [10]

Next table, on the other hand, shows the proximate analysis of the fuel.

Component	Value [weight %]*
Total Moisture	33.37
Ash	16.46
Volatile Matter	27.59
Fixed Carbon	4.23
Total	100.00

Table 6. 4 - Fuel proximate analysis [10]

Fuel LHV is 10,340 kJ/kg, while fuel HHV (higher heating value) is 12,078 kJ/kg. Other properties of the fuel are the specific heat (c_p) at 25°C of dry fuel of 1.13 kJ/(kg°C) and the specific heat at 300°C of dry fuel of 1.926 kJ/(kg°C). Finally, its bulk density is supposed to be of 288.3 kg/m^3.

Fuel analysis and composition determines the quantity and quality of bottom and fly ashes created within the boiler as well as the amount of fuel flow mass rate within the boiler to produce the required power of 200 MW.

6.2.9. Ash composition

The software Thermoflex provides a wide range of fuel datasheets implemented within the programme itself. The fuels "Municipal Solid Waste" defined in Thermoflex have been used as references for ash composition, showed in the **Table 6. 5** below. Since all the five type of MSW presents the same ash composition, the choice among one of them to be taken as reference has no point.

Component	Value [weight %]
Silica, SiO_2	40.0
Aluminium Oxide, Al_2O_3	24.0
Titanium Dioxide, TiO_2	1.3
Iron Oxide, Fe_2O_3	16.8
Calcium Oxide, Ca_2O	5.8
Magnesium Oxide, MgO	2.0
Sodium Oxide, Na_2O	0.8
Potassium Oxide, K_2O	2.4
Phosphorous Pentoxide, P_2O_5	0.1
Sulphur Trioxide, SO_3	5.3
Other	1.5
Total	100.00

Table 6. 5 - Ash analysis

6.2.10. Air composition

Air composition adopted in the present plant is the default composition given by Thermoflex for humid air. Since this data is in agreement with the real mean humid air composition, it has not been changed and is reported in the table below.

Component	**Value [mole %]**
Oxygen O_2	20.814
Carbon dioxide CO_2	0.030
Moisture H_2O	0.649
Nitrogen N_2	77.573
Argon Ar	0.934
Total	100

Table 6. 6 - Air composition

6.2.11. Water pressure drop

The programme includes reasonable pressure drops for each and every component, while pressure drops in ducts, at turbine inlet and in the deaerator are neglected for the sake of simplicity. As regards the boiler, pressure drops are concentrated in economisers and super-heaters/re-heaters tubes.

Referring to the vapour cycle, reasonable values has been fixed at condenser and regenerators inlet (for the last ones, for both water and vapour side) by the programme.

Component	**Pressure drop**	**Unit of Measure**	**Value**
Condenser	Water heat to condensate outlet	[m]	1
1st LP Regenerator*	Water-side Δp/p	%	5
	Vapour-side Δp/p	%	1
2nd LP Regenerator*	Water-side Δp/p	%	5
	Vapour-side Δp/p	%	1
1st HP Regenerator (if present)**	Water-side Δp/p	%	1
	Vapour-side Δp/p	%	1
Economisers	Overall Δp/p	%	8
Super-heaters/Re-heaters	Overall Δp/p	%	8

*the overall pressure drop of the two low-pressure regenerators is obviously the same as the water-side pressure drop in the condensate preheater since they are in parallel

** the overall pressure drop of the first high pressure regenerator is obviously the same as the water-side pressure drop in the first economiser since they are in parallel

Table 6. 7 - Hypothesised pressure drops in water cycle

6.2.12. Gases pressure drop

Similarly to the water cycle, even the gases cycle (which comprehend flue gases as well as air) has pressure drops, which occurs in correspondence to the passage within each and every heat exchanger within the boiler, starting from the convective evaporator. Again, reasonable values has been fixed by Thermoflex at inlet and outlet of every heat exchanger.

Pressure drop	**Unit of Measure**	**Value**
Convective Evaporator Δp/p	%	0
Super-heater SH2/Re-heater RH3 Δp/p	%	0.02
Super-heater SH1/Re-heater RH2 Δp/p	%	0.02

Economiser ECO2/Re-heater RH1 Δp/p	%	0.08
Economiser ECO1.5/Re-heater RH0.5 Δp/p (if present)	%	0.4
Economiser ECO1 Δp/p	%	1
Condensate preheater Δp/p	%	2
1st air preheater air-side pressure drop	[mbar]	0.7
2nd air preheater air-side pressure drop	[mbar]	0.6

Table 6. 8 - Hypothesised pressure drops in gases cycle

6.2.13. Turbine thermodynamic assumptions

Turbine thermodynamic assumptions consider both the high-pressure and low-pressure turbine sub-systems and the turbine body as a whole.

Some thermodynamic hypothesis have been made according to values suggested by the programme itself, which reflect the most common values for state-of-the-art components.

The **Table 6. 9** below shows the principal assumptions specifically made for the turbine body.

Parameter	Unit of Measure	Value
Low-pressure body shaft speed*	rpm	3000
Mechanical efficiency	%	99.75
Condensation quality (Wilson line)	[-]	0.97
Moisture efficiency penalty (Baumann coefficient)	[-]	0.72

*High-pressure body shaft speed within Thermoflex programme does not influence any output parameter (such as power produces) so its definition results to be unnecessary.

Table 6. 9 - Turbine body thermodynamic assumptions

Low-pressure body shaft speed has been chosen in order to maximise the turbine efficiency without exceeding in not-real values.

The programme estimates leakages and overall efficiency. Moreover, total exhaust loss within the turbine body is corrected thanks to a correlation provided by the programme, which associates the overall dry exhaust loss with the exhaust volume flow (**Figure 6. 6**). The graphic is peculiar for every turbine segment.

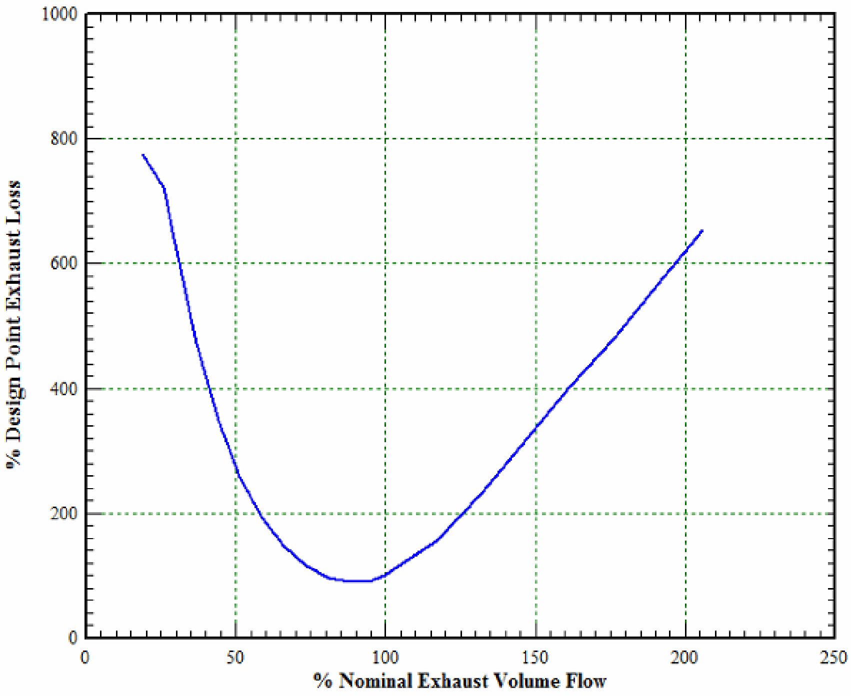


Figure 6. 6 - Exhaust loss curve (non-dimensional)

This loss contributes to form the actual leaving enthalpy of the low-pressure turbine steam and includes losses associated with the turbine exhaust hood, the steam leaving velocity (kinetic energy) and the restriction loss of turn up loss.

The corrected exhaust loss is then obtained with the following formula also provided by Thermoflex:

$$\mathrm{CEL} = \mathrm{ELdry} * 0.87 * (1 - \mathrm{y}) * (1 - 0.65\mathrm{y})$$

Where *ELdry* indicates exhaust dry loss obtained in the curve shown in the **Figure 6. 6** above and *y* the exhaust moisture fraction.

The condensation quality coefficient is associated to the Wilson line.

Steam expanding is a turbine invariably goes from a superheated state to a saturated mixture state. When crossing the saturated vapour line rapidly, the steam does not condense immediately, but remains briefly in a state of metastable equilibrium. In this state, it follows the laws governing superheated steam, until a lower pressure is reached. At some point condensation suddenly takes place and the fluid is once again in thermal equilibrium and subject to the laws of saturated mixture. The points in which this condensation occurs vary depending upon conditions and are scattered in a band somewhat below the saturation line as shown in **Figure 6. 7**.

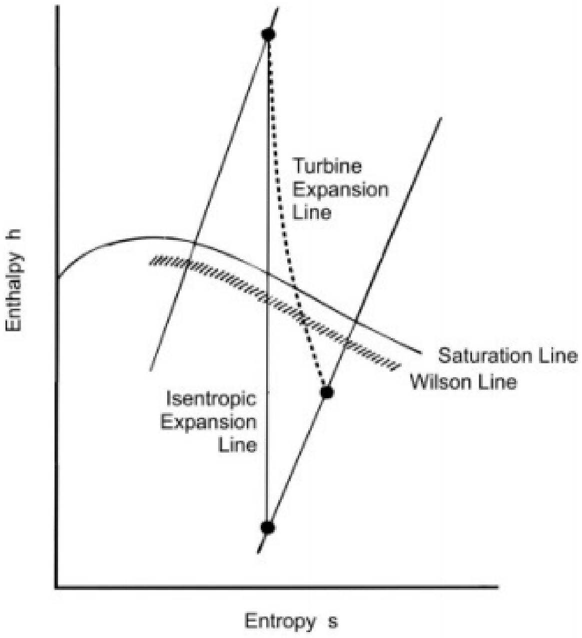


Figure 6. 7 – Wilson line relative to saturation line

This band is known as the Wilson line. The phenomenon of metastable equilibrium is associated with the surface effects on very small drops and bubbles, which are associated in turn with additional losses, counted within the coefficient shown in the **Table 6. 9** above.[19]

Finally, the Baumann coefficient corrects the theoretical isentropic turbine efficiency due to wetness within turbine stages with slightly superheated steam at the entrance. The loss due to wetness corresponds to the sub-cooling loss. This coefficient, moreover, takes into account not only the previously mentioned energy losses, but also working-fluid mass losses, because the steam converted into water no longer perform work in the considered and subsequent turbine stages.[20]

6.2.14. Combustor thermodynamic assumptions

Assumptions about the furnace layout are related to heat transfer both in the combustion reaction and from flue gases to water within the radiant evaporator. The following **Table 6. 10** summarises the value suggested by the programme and which represents the state-of-the-art data for waste-to-energy combustors.

Parameter	Unit of Measure	Value
Carbon to soot conversion rate	%	0.3
Fly ash recirculation ratio	-	1
Fly ash as percentage of total ash	%	10
Unburned carbon loss as percentage of carbon in fuel	%	1.5
Distribution of unburnt carbon in fly ash and bottom ash	-	0.5
Heat transfer non-uniformity correction factor	-	0.55
Fuel chlorine converted to HCl	%	80
Fuel mercury leaving with flue gases	%	80
Flue gas SO_2 to SO_3 conversion rate	%	0.5

Table 6. 10 - Furnace heat transfer properties

The following **Table 6. 11**, on the other hand, shows the combustor heat transfer assumptions related to the waterwall.

Parameter	Unit of Measure	Value
Wall thermal conductivity at 260°C	W/(m°C)	46.73
Wall thickness	mm	12.7
Water side fouling factor	m^2°C/W	$1.761*10^{-4}$
Gas side fouling factor	m^2°C/W	0.022
Primary waterwall protection	-	Pin-stud SiC refractory
Primary protection coverage	%	40
Primary coverage thermal resistance	m^2°C/W	0.0044
Secondary coverage by Inconel	%	60

Table 6. 11 - Waterwall heat transfer assumptions

6.2.15. Convective section thermodynamic assumptions

This section of the paragraph aims to describe the thermodynamic choices made to better distribute the temperature range that compete the various heat exchangers. In particular, the analysis must reach two objectives:

- Maintain the same flue gases and water temperature before and after every heat exchanger in parallel (i.e. flue gases temperature at inlet and outlet of *SH2* and *RH1* must be the same)
- Reduce as much as possible the gap between flue gases and water/vapour temperature (respecting the limit imposed in the **Table 6. 1** for approach point)

These two purposes can be achieved through the correct management of two parameters:

- Correctly distributing the temperature interval among the heat exchangers allows to optimise the temperature gap between fluid and gases
- Changing the quantity of flue gases split between the two heat exchangers in parallel allows to regulate the temperature of flue gases exiting each heat exchanger

In fact, defining Q_{HE} the heat exchanged between fluid and flue gases within the tubes of a generic super-heater (but it would be the same, with some adaptation, for evaporators and/or economisers):

$$Q_{HE} = \dot{m}_{H2O}\left(h_{H2O,out} - h_{H2O,in}\right) = \dot{m}_{gas}c_{p,gas}\left(T_{gas,in} - T_{gas,out}\right)$$

Indicating the mass flux rate of water and flue gases respectively with $\dot{m}_{H2O}$ and $\dot{m}_{gas}$. The terms with *h* indicated the enthalpy of the vapour while the $c_{p,gas}$ indicated the specific heat of flue gases and the terms with *T* stand for the temperature of flue gases before and after the heat exchanger. Water parameters are fixed by the plant mass and energy balances (mass flux rate) and the requirements and data assumptions (evaporation pressure and superheating temperature stabilise the water enthalpy). The enthalpy of vapour at the inlet of the heat exchanger could be varied modifying the temperature of the heat exchanger inlet. However, the necessity of maintaining the same fluid temperature at inlet and outlet of every heat exchanger prevent this possibility.

On the other hand, the overall flue gases mass flow rate is fixed by the quantity of fuel and air introduced within the boiler. Fuel is fixed by the required power of the overall plant, while the air excess fixes the air quantity, which in turn leads to a request on the oxygen percentage in flue gases. The specific heat of flue gases weakly depends on their temperature and pressure and can be considered, on first approximation, as a constant.

Finally, flue gases temperatures are fixed by the necessity to be the same as the other heat exchanger in parallel with the first one, the necessity to decrease the temperature gap between water and flue gases as much as possible, without exceeding the limit of 50°C and to guarantee the required vapour temperature achievement.

The only degree of freedom remained is thus the mass flow rate of flue gases. Adjusting these parameters allows to respect all the requirements listed above, in particular to maintain the same flue gases temperature at every heat exchangers in parallel outlet.

The following lines would be spent to evaluate the specific assumptions made for every couple of parallel convective heat exchangers. Initial pressure within the boiler (which in turn depends on fuel, FGR and air pressure and combustion process) fixes flue gases pressure, which resents of pressure drop within every heat exchanger.

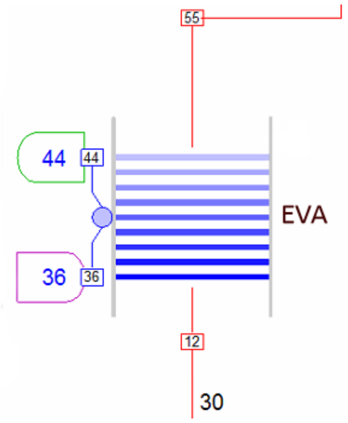


Figure 6. 8 - Convective evaporator

Flue gases temperature before the convective evaporator is the one obtained by gas quenching, 650°C, while at evaporator outlet it is calculated by the programme in order to guarantee the water heating up. Water enters at evaporation pressure with the evaporation temperature minus the sub-cooling ΔT and exits at condensing temperature.

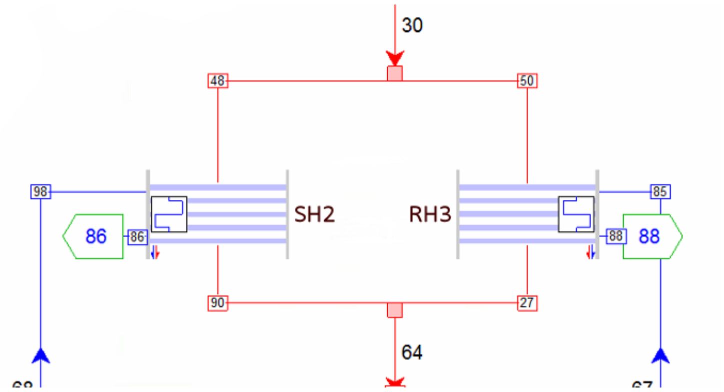


Figure 6. 9 - Second super-heater in parallel with the third re-heater

In order to increase the overall efficiency, the two first super-heater/re-heater are in parallel with flue gases. This means that the hotter flue gases encounter the colder vapour, while the colder flue gases meet the hotter superheated vapour.

The previous convective evaporator dictates the flue gases temperature at super-heater/re-heater inlet. The outlet vapour temperature is fixed to 450°C, which is the highest consented value for superheated vapour temperature without risk of hot corrosion. Inlet vapour temperature is cooled down by the two de-super-heaters, which add the cold water from the deaerator outlet to the superheated vapour from *SH1/RH2*. The outlet temperature from the first super-heater and second re-heater is regulated in order to obtain the required flue gases temperature at heat exchangers exit.

Outlet flue gases temperature is subjected to two constraints: it has to be the same at both heat exchanger's outlets and it is regulated in order to reduce as more as possible the gap with the superheated vapour. To this purpose, the regulation techniques adopted could follow two paths:

- **1st Strategy.** Reduce the gap between colder flue gases and hotter superheated vapour temperature to the minimum consented (minimum approach point is equal to 50°C). This means that, being the outlet superheated vapour temperature fixed, even the outlet flue gases temperature is fixed to 500°C and superheated vapour inlet temperature is consequently adjusted.
- **2nd Strategy.** Try to regulate inlet superheated vapour temperature in order to better distribute the temperature gap between the two couple of super-heater/re-heater. This strategy increase the gap in the *SH2/RH3* approach point, but guarantee the *SH1/RH2* vapour T-Q diagram to be more close to the flue gases' one, increasing its heat exchanging efficiency.

The **Figure 6. 10** shows the difference between the two strategies. The second strategy decreases the vapour inlet temperature in *SH2/RH3*, consequently decreasing the temperature gap elaborated by these heat exchangers. The flue gases temperature at the first two heat exchangers is thus higher than the first strategy, but the second vapour T-Q diagram results more close to the flue gases because of the parallel configuration of the first two heat exchangers respect to the flue gases direction. Positioning as parallel the first heat exchangers, in fact, permits to maintain within imposed limits the tubes' material temperature.

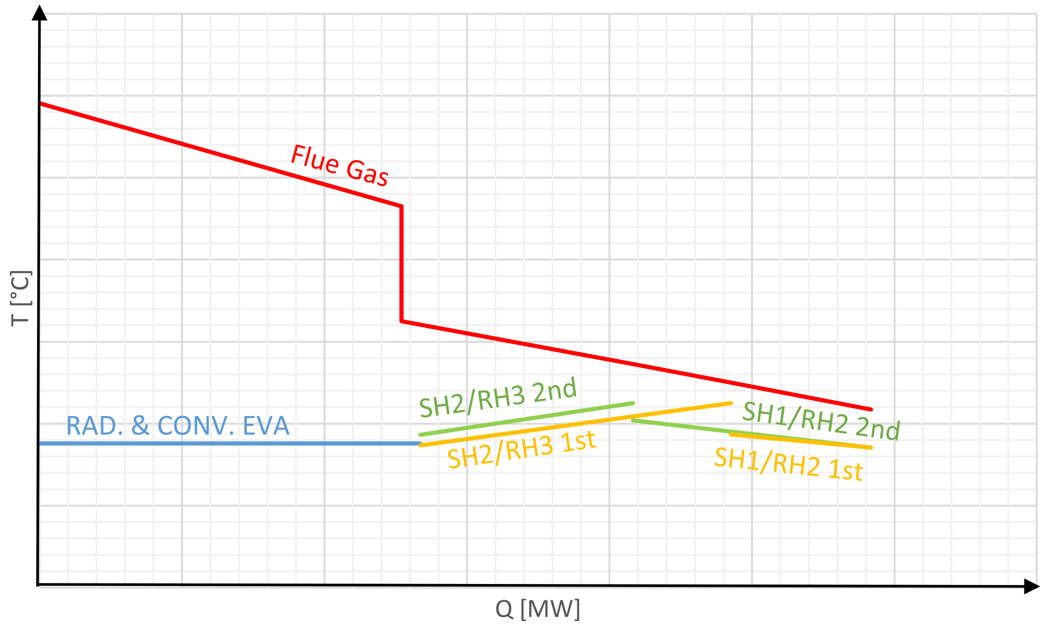


Figure 6. 10 - T-Q diagram of the first section of waste-to-energy boiler, representing the differences between the two strategies, indicated as "1st" and "2nd"

The second strategy is proven to guarantee a higher overall plant efficiency and thus is the one adopted in the present plant. However, this last strategy introduces two more constraints about the maximum temperature to which the materials must reach in order not to run into hot corrosion or decay of materials mechanical properties (**Table 6. 12**).

Material temperature constraint	Unit of Measure	Value
SH2-RH3	°C	475
SH1-RH2	°C	420

Table 6. 12 - Material temperature constraints in super-heaters and re-heaters

Flue gases exits temperature is adjusted in the optic of respecting the constraints showed in the previous table as well as the ones showed before. For the reasons explained before, the flue gases exiting the first super-heater/re-heater would be higher than 50°C more than the superheated vapour temperature.

In the reference plant case, the strategy adopted is exactly the same, even though the re-heat components are obviously absent.

In conclusion, the superheated vapour exit temperature is fixed to 450°C and the heat absorbed within the convective evaporator determines the flue gases inlet temperature. On the other hand, superheated vapour inlet temperature depends on the mass flow rate and temperature of water in the de-super-heaters (which are fixed respectively by the data assumptions and the temperature after the feed pump in the steam cycle) and on the flue gases exit temperature. The last, in turn, depends on the constraints imposed, in particular aims to approach as closely as possible, the material temperature constraints indicated in **Figure 6. 1**.

The presence of de-super-heaters do not allow the respect of the requirement of same temperature values both at *SH1/RH2* outlet and *SH2/RH3* inlet. In fact, since the quantity of water added to the super-heaters section is the same as the re-heaters section as well as its property, the effect that it causes on the two fluxes is obviously different, for they have the same temperature but different pressures (which means their transformation is located in a different part of the entropy diagram). Consequently, if the superheated vapour temperature is the same at *SH1/RH2* outlet, it would not be the same at *SH2/RH3* inlet and vice-versa.

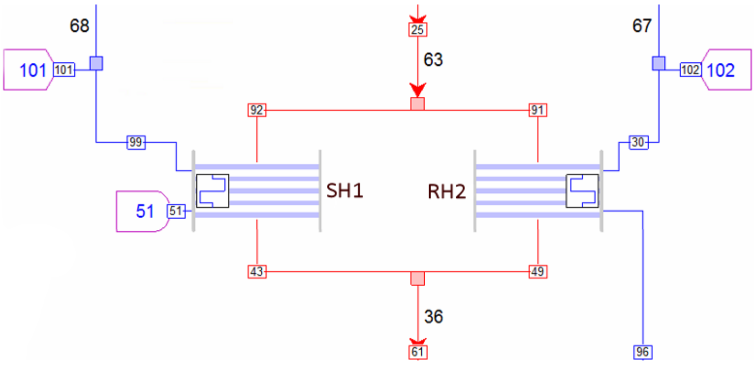


Figure 6. 11 - First super-heater in parallel with the second re-heater

Flux *51* exits the evaporator (it is the merge of radiant and convective evaporations fluxes) before entering the first super-heater. Consequently, its conditions are fixed: its pressure is the evaporation pressure and temperature is the condensation temperature at that evaporation pressure. Thus, even superheated vapour inlet temperature in *RH2* is fixed, because of the necessity to be the same as the other heat exchanger in parallel. Superheated vapour outlet temperature is determined as explained in the *SH2/RH3* section.

The heat absorbed within the previous heat exchangers determines the flue gases inlet temperature, while their outlet temperature is fixed by the constraints respect requirement and the necessity to guarantee the same temperature at *SH1* and *RH2* exit.

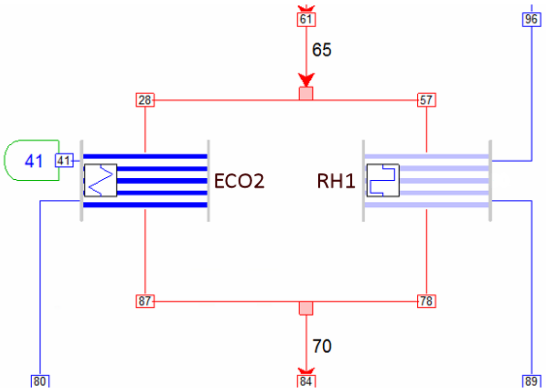


Figure 6. 12 - Second economiser in parallel with the first re-heater

Flue gases temperature is dictated, as in the previous heat exchangers cases, by the heat absorbed in the previous and present heat exchangers and the constraint to maintain the same flue gases temperature at heat exchangers exits. Economiser dictates the outlet water/vapour temperature, because flow *41* enters the evaporator. This means that its temperature is the condensation temperature at evaporation pressure minus the sub-cooling temperature gap, fixed to 10°C. The re-heaters imposes the inlet temperature of water/vapour, since flow *89* comes directly from the high-pressure turbine outlet. Data assumptions determines its pressure and turbine efficiency, computed by the programme, fixes its temperature.

In case of 110 bar or 130 bar as evaporation pressures, the water entering the economiser, comes directly from the high-pressure regenerator (absent in configuration with evaporation pressures 150 bar or 170 bar). Re-heat pressure determines the vapour spilled pressure from high-pressure turbine that enters the high-pressure regenerator, because its pressure fixes, as well as the mass flow rate subdivision between regenerator and first economiser, the final temperature of regenerator outlet.

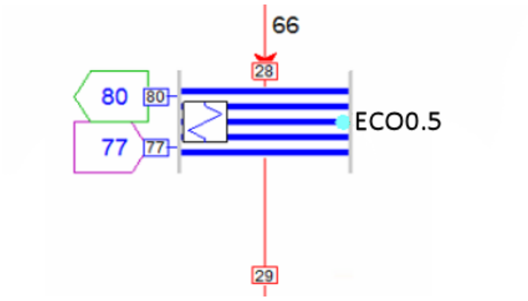


Figure 6. 13 - First Economiser

Finally, the first economiser receives water from the deaerator (or the high-pressure regenerator where presents) and flow *80* exits it in order to enter the second economiser (150 bar or 170 bar as evaporation pressures) or to join the water coming from high-pressure regenerator and then enters the second economiser (110 bar or 130 bar as evaporation pressures).

Heat absorbed from previous heat exchangers defines the flue gases inlet temperature, while the necessity to guarantee the correct mode of operation of flue gas treatment system dictates the flue gases outlet temperature, stabilising it around 185°C.

6.3.COST ASSUMPTIONS

As mentioned before, Thermoflex can operate cost calculations of the overall plant, excluded the radiative section.

As a result, data from a similar plant has been used in order to provide a rough estimation of the waterwall costs, as extensively explained in Chapter 4. The following **Figure 6. 13** provide costs related to the radiative section and boiler, extracted from the similar plant calculations, divided by evaporation pressures. The proportion explained in the previous chapter has been than operated and the final results are consultable in the next chapter.

Cost parameter – 70 bar (ref plant)	Value [USD]
Specialised Equipment – Boiler	102’470’000
Specialised Equipment – Waterwall	160’395’008
Civil, Excavation and Backfill – Boiler	1’288’400

Civil, Concrete – Boiler	20’479’201
Mechanical, Equipment erection and Assembly – Boiler	63’022’000
Electrical Assembly and Wiring, Controls – Boiler	63’022’000

Table 6. 13 - Cost parameters for waterall costs estimations - reference plant

Cost parameter – 110 bar*	Value [USD]
Specialised Equipment – Boiler	*
Specialised Equipment – Waterwall	*
Civil, Excavation and Backfill – Boiler	*
Civil, Concrete – Boiler	*
Mechanical, Equipment erection and Assembly – Boiler	*
Electrical Assembly and Wiring, Controls – Boiler	*

*Costs for the plant with 110 bar as evaporation pressure were not present in the datasheet provided for the similar plant, so they have been calculated as linear interpolation through a trend line from final values, resulted from the proportions explained in Chapter 4

Table 6. 14 - Cost parameters for waterall costs estimations - evaporation pressure 110 bar

Cost parameter – 130 bar	Value [USD]
Specialised Equipment – Boiler	136’163’500
Specialised Equipment – Waterwall	70’622’000
Civil, Excavation and Backfill – Boiler	780’530
Civil, Concrete – Boiler	12’608’600
Mechanical, Equipment erection and Assembly – Boiler	45’287’100
Electrical Assembly and Wiring, Controls – Boiler	3’652’350

Table 6. 15 - Cost parameters for waterall costs estimations - evaporation pressure 130 bar

Cost parameter – 150 bar	Value [USD]
Specialised Equipment – Boiler	134’628’500
Specialised Equipment – Waterwall	64’997’000
Civil, Excavation and Backfill – Boiler	796’350
Civil, Concrete – Boiler	12’872’901
Mechanical, Equipment erection and Assembly – Boiler	42’295’200
Electrical Assembly and Wiring, Controls – Boiler	3’410’400

Table 6. 16 - Cost parameters for waterall costs estimations - evaporation pressure 150 bar

Cost parameter – 170 bar	Value [USD]

Specialised Equipment – Boiler	135’116’650
Specialised Equipment – Waterwall	59’311’000
Civil, Excavation and Backfill – Boiler	708’070
Civil, Concrete – Boiler	11’481’301
Mechanical, Equipment erection and Assembly – Boiler	38’549’700
Electrical Assembly and Wiring, Controls – Boiler	3’107’900

Table 6. 17 - Cost parameters for waterall costs estimations - evaporation pressure 170 bar

High values for waterwall costs (which constitutes roughly half of the entire boiler cost) can be explained adding that they comprehend also the costs for casing and piping of the whole boiler body.

When it comes to overall plant costs, however, Thermoflex calculates them with a series of multiplicative coefficients listed in the tables below. They are provided directly by the software and are so maintained as the most reasonable data for a high-power plant.

All the costs are provided in USD, which are United States Dollars, since Thermoflex is an American program. The software would provide regional costs and conversion from dollars to euros, but since the program maintains the exchange rate constant and not continuously updated, the analysis with the default unit of measure results preferable in reliability terms.

The next table presents the principal contractor’s soft costs.

Parameter	Unit of Measure	Value
Contingency for labour	%	15
Contingency for specialised equipment	%	2
Contingency for other equipment	%	3
Contingency for commodities	%	5
Profit margin on labour	%	20
Profit margin on specialised equipment	%	5
Profit margin on other equipment	%	5
Profit margin on commodities	%	5
Permits, licenses, fees and miscellaneous	%	0
Bonds and insurance	%	1
Contractor’s fee	%	3

Table 6. 18 – Contractor’s soft costs

The next table presents the principal owner’s soft costs.

Parameter	Unit of Measure	Value
Permits, licenses, fees and miscellaneous	%	2
Land cost	%	0
Utility connection cost	%	0
Legal and financial costs	%	2
Escalation and interest during construction	%	4
Spare parts and materials	%	0
Project administration and developer’s fees	%	1

Table 6. 19 - Owner's soft costs

Moreover, since the cost for Inconel cladding must be calculated, some hypothesis have to be clarified before showing the results. The first hypothesis is related to the area to cover in Inconel. As explained before, the critical section for hot corrosion problems is only the *SH2/RH3* one. Consequently, only this area would have the protective coverage. Secondly, the Inconel coverage costs has been assumed to 2500 €/m^2, in agreement to the data provided.

6.4. DIMENSIONING ASSUMPTIONS

This paragraph aims to report all the dimensioning assumptions made in order to design the final waste-to-energy plant. Design of some component such as condenser, deaerator, regenerators and auxiliaries have been made according to values suggested by the programme itself, which reflect the most common values for state-of-the-art components.

The program automatically dimensions the elements that are not mentioned in the following sub-paragraphs, therefore dimensioning assumptions are not necessaries for them.

6.4.1. Condenser

The first component's dimensioning hypothesis are related to the condenser. The **Figure 6. 14** below shows its elevation view (for schematisation purposes only).

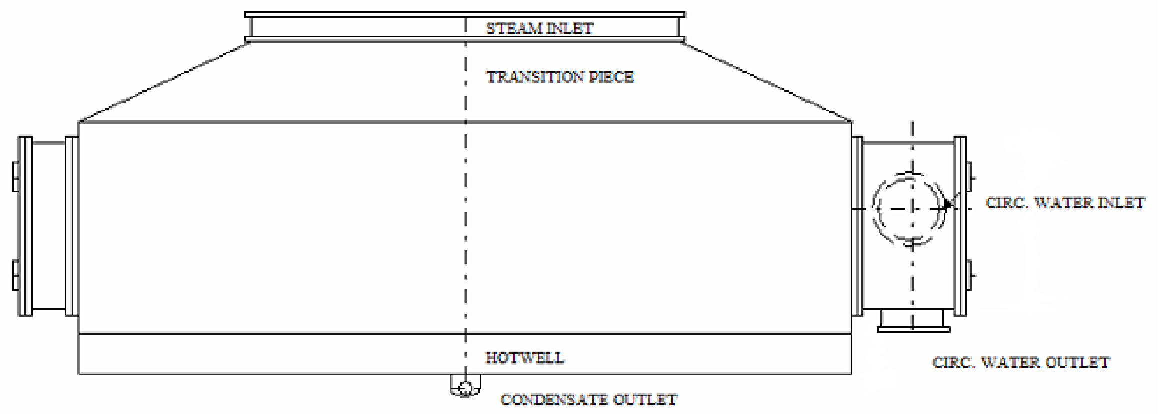


Figure 6. 14 - Condenser elevation view

The **Table 6. 22** below presents the main condenser hardware inputs.

Parameter	Unit of Measure	Value
Tube material	-	Stainless steel
Tube type	-	Seam welded
Cleanliness factor	%	90
Tube outer diameter	mm	19.05
Tube thickness	mm	0.7112
Tube pitch/outside diameter	-	1.6
Tube metal conductivity	W/(m°C)	14.88
Tube water velocity	m/s	1.829
Number of condenser passes	-	2
Aspect ratio of uniformly-spaced tube bundle (height/width)	-	1

Condenser cross section/uniformly-spaced tube bundle cross section	-	1.56
Cooling water pipe water velocity	m/s	2.591

Table 6. 20 - Condenser hardware inputs

6.4.2. Regenerator

The two (or three) regenerators present within the plant can be schematised as follows (**Figure 6. 15**).

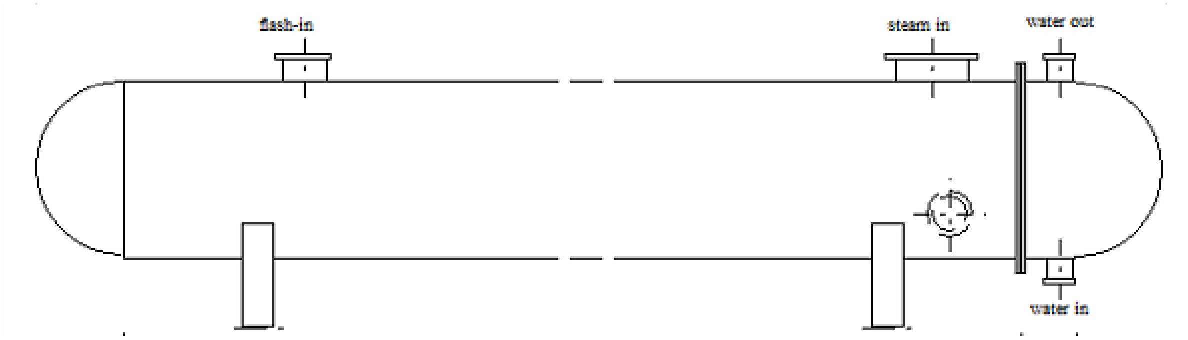


Figure 6. 15 - Regenerator elevation view

As in the condenser case, regenerators are dimensioned following the most common state-of-the-art procedures and assumptions, summarised in the **Table 6. 23** below.

Parameter	Unit of Measure	Value
Tube water velocity	m/s	1.448
Tube inside fouling resistance	$m^2°C/W$	$5.812*10^{-5}$
Tube outside fouling resistance	$m^2°C/W$	0
Number of feedwater passes	-	2
Minimum pinch temperature difference	°C	3
Tube material	-	Stainless steel
Tube outer diameter	mm	15.88
Tube thickness	mm	1.245
Tube pitch/outside diameter	-	1.65
Tube metal conductivity at 150°C	W/(m°C)	16.61
Tube metal conductivity slope	$W/(m°C^2)$	0.0125
Shell inner diameter/uniformly-spaced tube bundle diameter	-	1.35

Table 6. 21 - Regenerator hardware inputs

6.4.3. Heat exchanger

All the heat exchanger present in the flue gases cycle can be schematised as in the **Figure 6. 16** below.

All the heat exchangers which are in contact with flue gases exiting the furnace (so all the evaporators, economisers, re-heaters and super-heaters) are not provided of fins on external surface of tubes, because of the high fouling factor of flue gases generated by waste combustion. As a result, heat exchanging surface would be more extended than the same surface provided by fins. Ultimately, this would cause an increase in material costs. On the other hand, air pre-heater are only in contact with external air, so can be provided with fins.

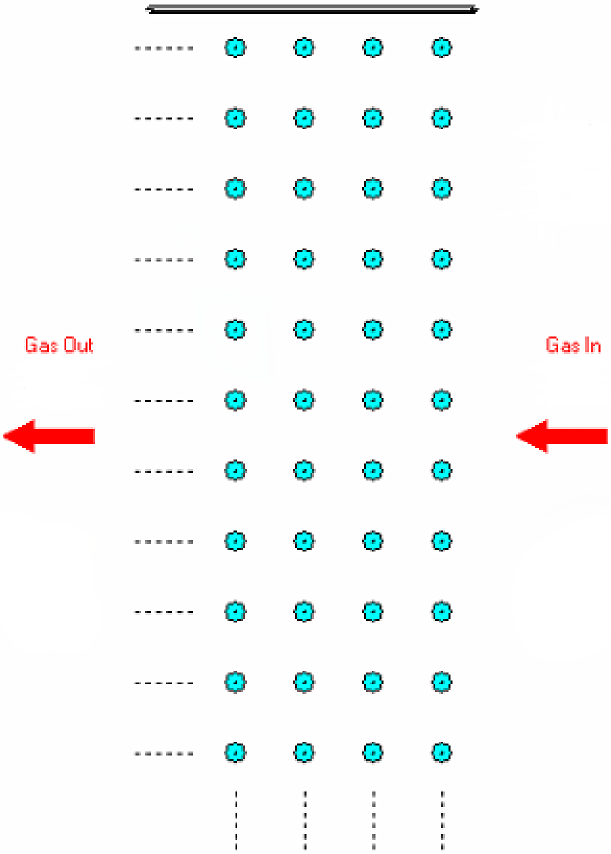


Figure 6. 16 - Heat exchanger scheme

Torino waste-to-energy plant has served as model for boiler heat exchangers' length, tube diameter and tubes' spacing in order to adopt reasonable measures for heat exchangers dimensioning. Moreover, Thermoflex sets up all the tubes' wall thicknesses to the minimum in order to support the pressure of water/steam that passes within them. The **Table 6. 24** below presents the main dimensioning hypothesis adopted for each and every heat exchanger within the whole plant.

Parameter	Unit of Measure	Value
Boiler water/stream side fouling factor	m^2°C/W	$1.761*10^{-4}$
Boiler gas/air side fouling factor	m^2°C/W	$8.806*10^{-4}$
Air pre-heater water/stream side fouling factor	m^2°C/W	$8.806*10^{-5}$
Air pre-heater gas/air side fouling factor	m^2°C/W	$1.761*10^{-4}$
Super-heater/re-heater longitudinal row pitch	mm	120
Super-heater/re-heater transverse row pitch	mm	120
Evaporator longitudinal row pitch	mm	120
Evaporator transverse row pitch	mm	240
Economiser longitudinal row pitch	mm	80
Economiser transverse row pitch	mm	80
Air pre-heater longitudinal row pitch	mm	63.5

Air pre-heater transverse row pitch	mm	63.5
Boiler heat exchangers' length	m	5.8
Super-heater/re-heater outer diameter	mm	50.8
Evaporator outer diameter	mm	44.5
Economiser outer diameter	mm	38.1
Air pre-heater outer diameter	mm	25.4
Air pre-heater tube length/heat exchanger width	-	1
Air pre-heater fin thickness	mm	0.6096
Air pre-heater fin spacing	mm	3.023
Air pre-heater number of fins	m^{-1}	275.3
Air pre-heater fin height	mm	12.7

Table 6. 22 - Heat exchangers hardware inputs

6.4.4. Auxiliaries

The auxiliaries hypothesis refers to the amount of power dedicated to miscellaneous auxiliaries. They are composed by a fixed part and a percentage of the gross power produced by the plant. According to the graduation thesis of F. Begnis, "*Bilancio energetico di sistemi con co-combustione di combustibile derivato dai rifiuti (CDR) e combustibili fossili"* [1], the percentage variable with the gross power produced by the plant has been left constant and equal to 1%, while che fixed power absorbed has been raised to 900kW.

Chapter 7

Performances evaluation

This chapter is about performance of the innovative waste-to-energy plant configuration. The following paragraphs would show the principal outcome resulting from the model computation by Thermoflex of the principal components of the plant. The first choice made in order to reduce degrees of freedom of the plant is about reheat pressure, illustrated in the following paragraph.

7.1. REHEAT PRESSURE CHOICE

The choice of re-heat pressure has been made before realising the final plant, in order to reduce degrees of freedom. The configuration of the plant used to obtain data regarding the different re-heat pressures is simpler than the final plant simulated in Thermoflex, but the results and conclusions are the same as the final plant. Choosing reheat pressure before further complication in plant configuration allows obtaining significant results without increasing calculation time. Since all the plants share the same overall input heat introduced by waste (which is 200 MW), the efficiency of the plant, defined as follows:

$$\eta = \frac{\text{net power produced}}{\text{heat introduced}}$$

depends only on net power produced and their trend as functions of reheat pressure is the same.

Hence, the following graphic (**Figure 7. 1**) shows the evolution of efficiency in the simplified configuration plant, with an evaporation pressure of 70 bar. It is demonstrable that the same results and the same conclusions can be deducted by the more complex plant model with the evaporation pressures required by data assumptions.

The diagram shows that between 10 bar and 20 bar the net power produced by the overall plant is maximum, while decreasing for higher reheat pressures. Its trend confirms the one presented in **Chapter 2**, showing the evolution of plant efficiency as function of reheat pressure. In this case, in fact, the maximum corresponds to the maximum in the efficiency graphic as well. Obviously, the calculated reheat pressures do not cover the entire interval from condensation pressure to evaporation pressure, so the form of the present diagram could mislead the reader about its true tendency.

Observing the data in the diagram, it is clear that there is no point in enquiring the reheat pressures over 50 bar, since it is known from the **Figure 2. 6**, that it would continue to diminish.

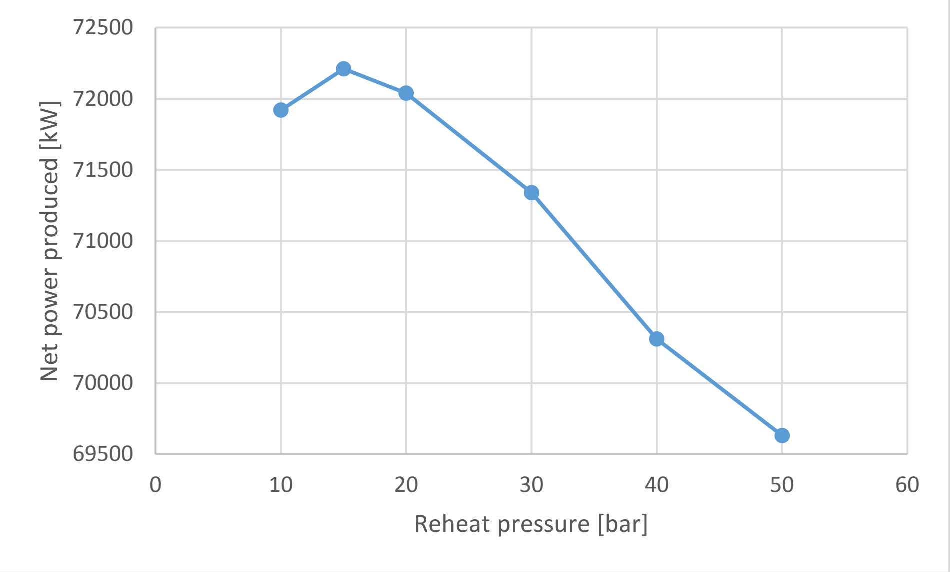


Figure 7. 1 - Net power produced by the plant as function of reheat pressure

However, especially for high evaporation pressures, being the superheating and reheating temperature constant and equal to 450°C, a mixture of vapour and water (condensed vapour with high vapour quality) could constitute the fluid exiting the high-pressure turbine and entering within the reheat heat exchangers, if the reheat pressure becomes too low. Therefore, in the new configuration plant, a reheat pressure of 15 bar can not be adopted since the fluid exiting high-pressure turbine must be superheated vapour, in order not to incur in fluid-dynamic losses and turbine blades damages. The **Figure 7. 1** above shows that, beyond the point of 15 bar, the power produced (and for what explained before, the overall efficiency) decreases, so it is important to maintain the reheat pressure lower than possible. The pressure that guarantees the exit of slight superheated vapour from high-pressure turbine for all evaporation pressures considered is **25 bar**, so this pressure would be fixed for furtherly plant calculations and outcome evaluations.

7.2.GENERAL PERFORMANCES

This chapter aims to provide the general performances of the plant, while the output of every model principal component would be presented in the specific paragraphs. The only variable left is the evaporation pressure, so the data in this and further paragraphs would be presented subdivided by this parameter.

	Evaporation Pressure [bar]				
	70 (cfr plant)	**110**	**130**	**150**	**170**
Gross power [kW]	65'124	70'277	71'128	71'560	71'736
Net power [kW]	58'119	62'868	63'518	63'529	63'444
Total Auxiliaries [kW]	7'004	7'408	7'610	8'031	8'293
Gross electric efficiency [%] **gross power/power introduced with fuel**	32.57	35.14	35.57	35.78	35.83
Net electric efficiency [%] **net power/power introduced with fuel**	29.06	31.44	31.76	31.77	31.65

Flue gases exiting the furnace [kg/s]	152.00	152.00	152.00	152.00	152.00
Steam flow at high-pressure turbine inlet [kg/s]	69.46	61.60	60.52	59.20	58.73

Table 7. 1 - General performances

The plant without reheat shows worse performances than any other configuration with reheat plant, confirming the superiority of the innovative configuration respect to the more traditional one. As shown in the **Table 7. 1** above, the gross and net power increases with the evaporation pressure, which is predictable according to the theory of steam cycles presented in the **Chapter 2**. Therefore, the increase in power becomes slighter with evaporation pressure increasing, becoming even negative in the case of net power for 170 bar as evaporation pressure. This table would suggest that the plant with 150 bar as evaporation pressure seems to be the best respect to the others, but performances are only half of the part of plant evaluation. The better evaporation pressure choice is the one that combines the best efficiency in relation to the best plant costs so, in conclusion, the lower cost per kilowatt produced.

7.3.STEAM CYCLE

This paragraph aims to analyse the steam cycle performances, showing the detailed outcome about the condenser, water preheating line, and turbines. Graphics, tables and figures would be provided for each and every component of the steam cycle. In particular, every flux follows a colour code: blue indicates water/steam fluxes while red air or flue gases. Tables with thermodynamic properties of the fluxes report the correspondent colour, while tables related to dimensioning and heat exchanged would be shown separately.

7.3.1. Condenser

The flow exiting the low-pressure turbine enters the condenser to be cooled down (not considering the cooling water), as shown in **Figure 7. 2**.

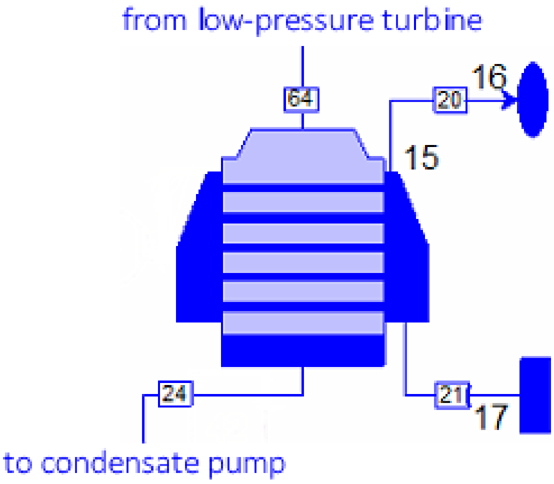


Figure 7. 2 - Condenser fluxes

The next **Table 7. 2** reports the principal thermodynamic data of every flux of the condenser, for every evaporation pressure.

Flux 64					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	0.07	0.07	0.07	0.07	0.07
T [°C]	39.03	39.03	39.03	39.03	39.03
x [-]	0.854	0.895	0.895	0.895	0.895
h [kJ/kg]	2220.02	2319.76	2319.76	2319.76	2319.76
m [kg/s]	52.57	47.40	47.01	46.93	46.77

Flux 24					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	0.1681	0.1681	0.1681	0.1681	0.1681
T [°C]	39.02	39.02	39.02	39.02	39.02
Sub [°C]	17.4	17.4	17.4	17.4	17.4
h [kJ/kg]	163.37	163.37	163.37	163.37	163.37
m [kg/s]	52.57	47.40	47.01	46.93	46.77

Table 7. 2 - Condenser thermodynamic output

The terms “Sub” indicates the difference between the evaporation temperature and the sub-cooled water temperature, at the same pressure. Dimensioning data are provided in the **Table 7. 3** below.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Effective surface area [m^2]	4567	4302	4267	4260	4137
Number of condenser passes [-]	2	2	2	2	2
Tube material [-]	Stainless steel	Stainless steel	Stainless steel	Stainless steel	Stainless steel
Number of tubes in condenser [-]	7107	6670	6615	6604	9791
Tube length [m]	10.7	10.8	10.8	10.8	8.5
Tube outside diameter [mm]	19.05	19.05	19.05	19.05	15.88
Tube inside diameter [mm]	17.63	17.63	17.63	17.63	14.45
Tube wall thickness [mm]	0.7112	0.7112	0.7112	0.7112	0.7112
Tube weight dry [kg]	25’100	23’650	23’460	23’420	22’570
Overall length [m]	12.7	12.7	12.7	12.7	10.4
Overall width [m]	3.0	2.9	2.9	2.9	2.9
Overall height [m]	5.0	4.8	4.8	4.8	4.9

Table 7. 3 - Condenser dimensioning output

Finally, the last **Table 7. 4** below shows the heat exchanged within the condenser.

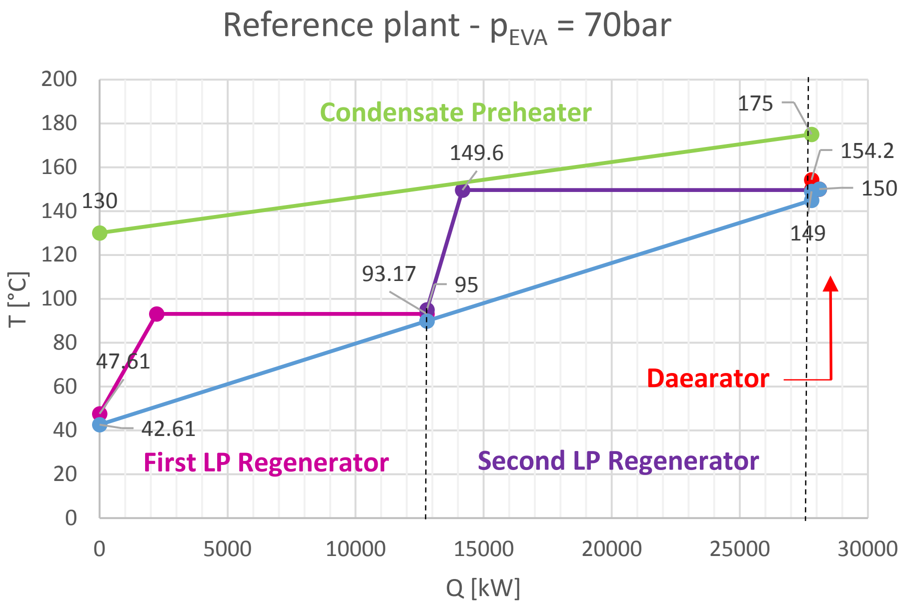
	Evaporation Pressure [bar]

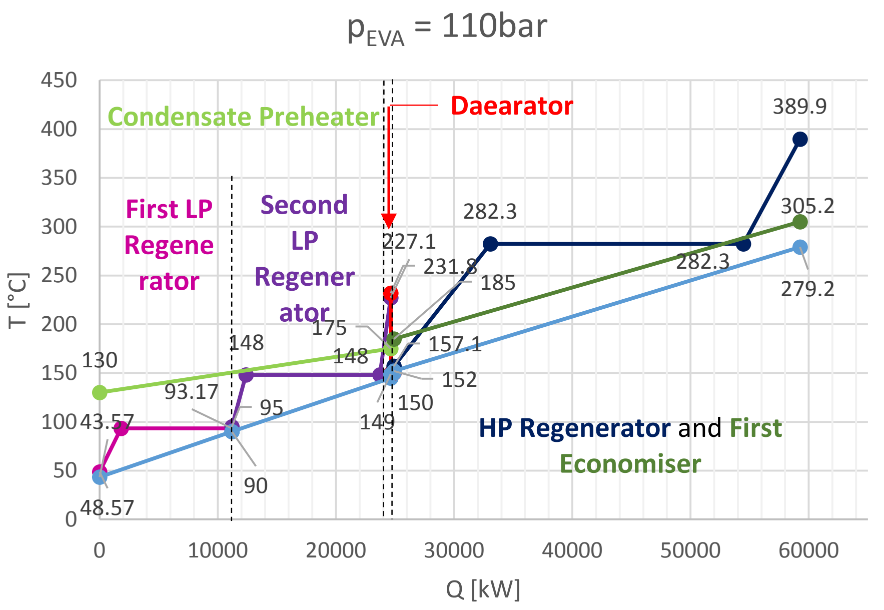
	70 (cfr plant)	110	130	150	170
Heat rejection [kW]	108'118	102'210	101'375	101'209	100'854

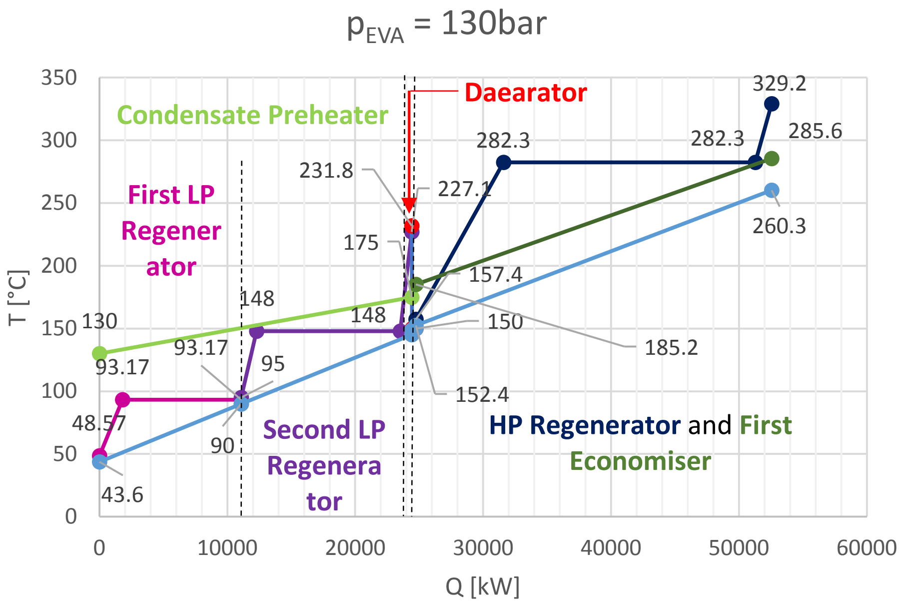
Table 7. 4 - Heat exchanged within the condenser

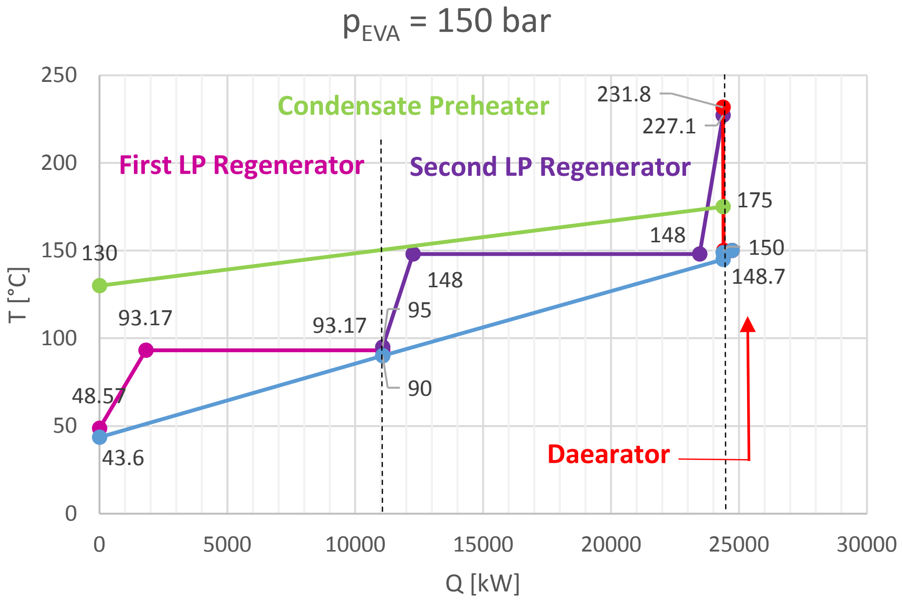
7.3.2. Water preheating line

This term comprehends all the devices used to increase the water temperature before having it entering the boiler. Therefore, the two low-pressure regenerators, the condensate pre-heater, the deaerator as well as the eventual high-pressure regenerator are part of this set. The following **Figure 7. 3** would show the T-Q diagram of the confronting plant as well as the four innovative configuration plants cases (in other words, the plant with the four different evaporation pressures).






p_EVA = 130bar
Condensate Preheater
Daearator
First LP Regenerator
Second LP Regenerator
HP Regenerator and First Economiser
329.2
285.6
282.3
282.3
260.3
231.8
227.1
175
157.4
150
148
148
185.2
152.4
130
93.17
93.17
95
90
48.57
43.6
T [°C]
Q [kW]
0
50
100
150
200
250
300
350
0
10000
20000
30000
40000
50000
60000


p_EVA = 150 bar
Condensate Preheater
First LP Regenerator
Second LP Regenerator
Daearator
231.8
227.1
175
150
148
148
148.7
130
93.17
93.17
95
90
48.57
43.6
T [°C]
Q [kW]
0
50
100
150
200
250
0
5000
10000
15000
20000
25000
30000

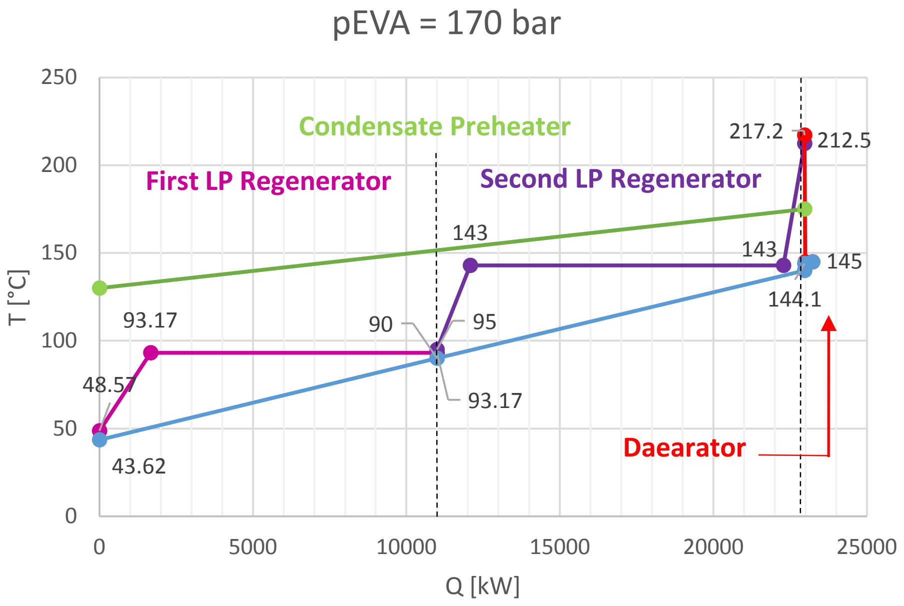


Figure 7. 3 - T-Q diagram of the water preheating line

As explained before, the heat exchanged within the deaerator is almost negligible compared to the one exchanged in regenerators or in the first economiser, because this device's objective is not to heat up the water, but to eliminate insoluble gases from the cycle.

The plant with 170 bar ad evaporation pressure fails to guarantee all the temperature of 185°C at boiler outlet because of the minimum temperature gap imposed on the economisers heat exchangers. To solve the problem, deaerator temperature has been diminished to 145°C and the second regenerator outlet temperature to 140°C (and the condensate pre-heater outlet temperature as well).

Finally, the T-Q diagrams do not seem to be influenced by the evaporation pressure and that is legit since the condensation pressure remains the same as well as the bleedings pressure. The only variable is the presence of high-pressure regenerator in parallel with the first economiser (in 150 bar and 170 bar plants) or its absence (in 110 bar and 130 bar plants), as well as the water/vapour quantity in the measure of which its temperature can or cannot be increased to the required one.

7.3.3. Turbine body

The high-pressure turbine comprehend the turbine sections that expand the vapour from super-heater outlet to re-heater inlet, eventually with the vapour bleeding for high-pressure regenerator. The low-pressure turbine, therefore, receives the reheated vapour and expands it until it reaches the condensation pressure. The following figures show the entire vapour expansion in the high-pressure turbine (**Figure 7. 4**) and low-pressure turbine (**Figure 7. 5**), in the h-s diagram for the plant with 110 bar as evaporation pressure. The other plants with different evaporation pressures show almost the same expansion diagrams, but the plant with evaporation pressures of 150 bar and 170 bar do not present the bleeding in the high-pressure turbine because of the absence of high-pressure regenerator. Moreover, the confrontation plant do not resent of the differentiation between high and low-pressure turbine, since the reheat is absent.

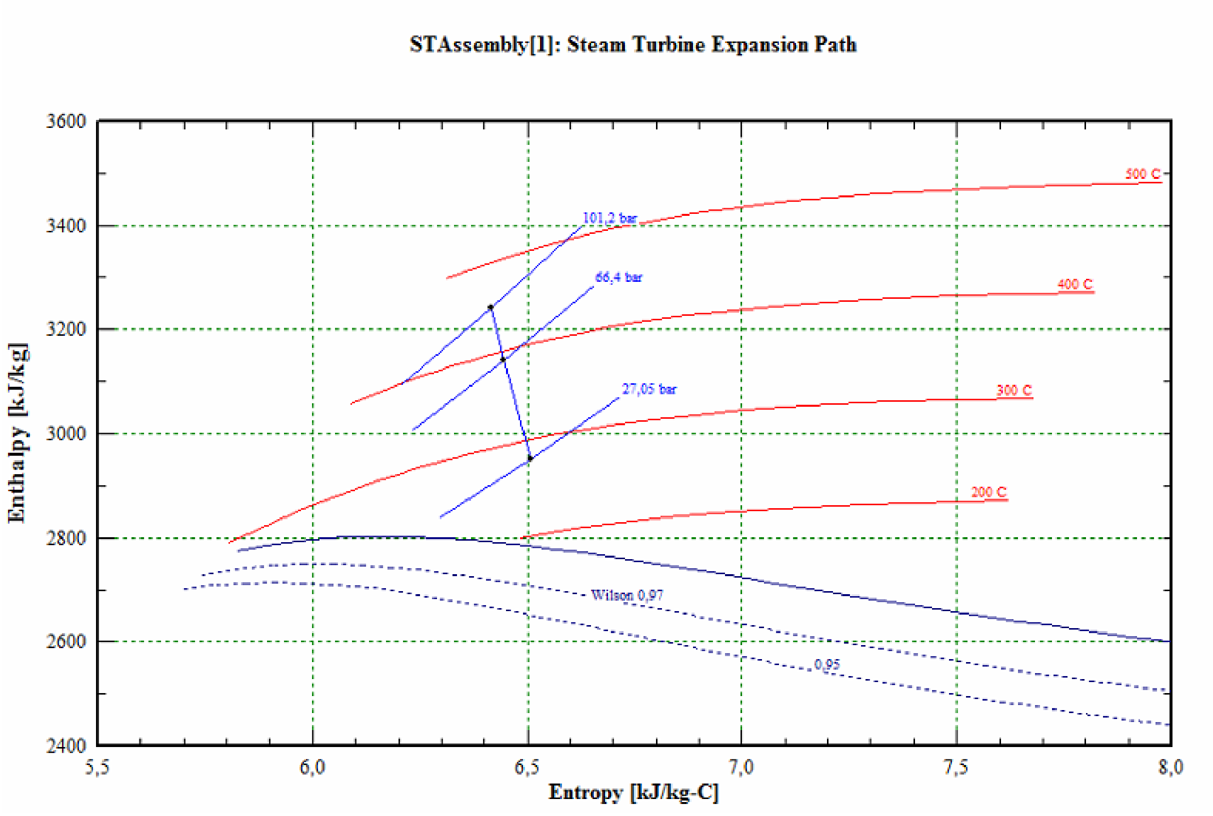


Figure 7. 4 - High-pressure turbine expansion h-s diagram

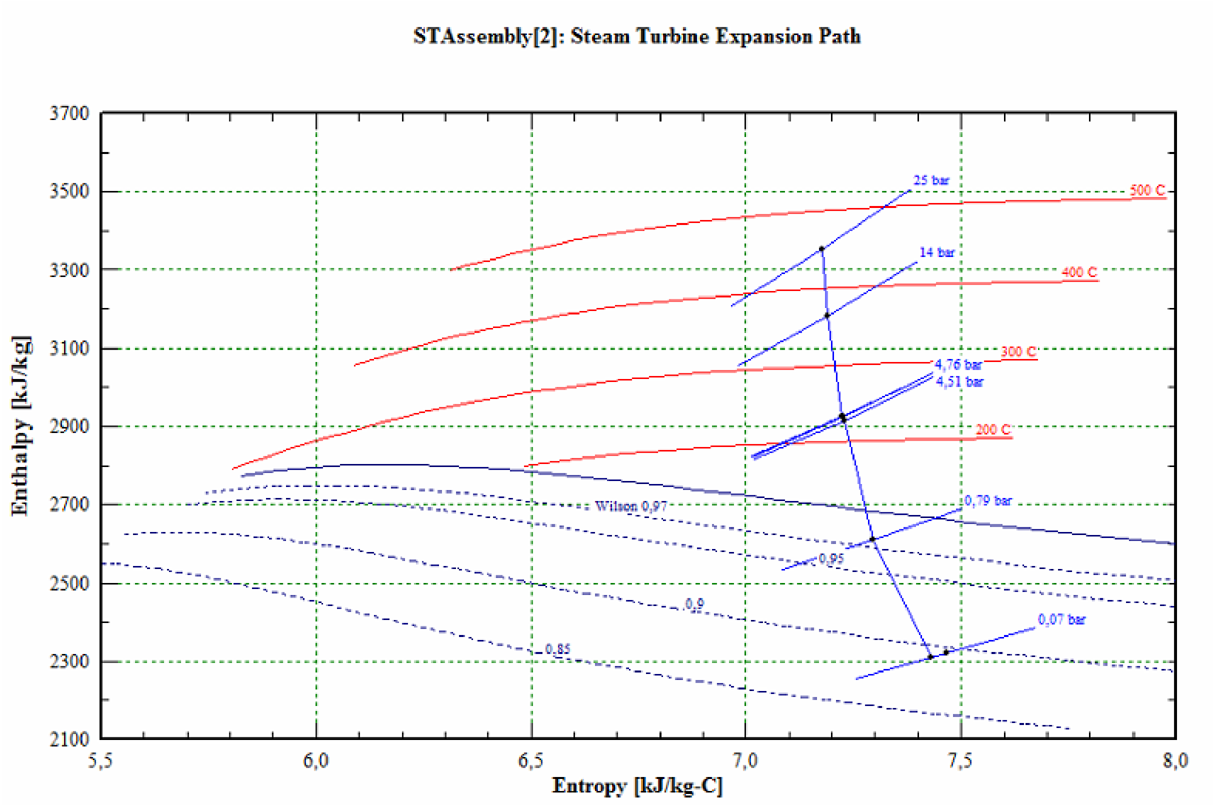


Figure 7. 5 - Low-pressure turbine expansion h-s diagram

The following images report the turbine bodies in order to introduce the main flows properties for all the plants considered.

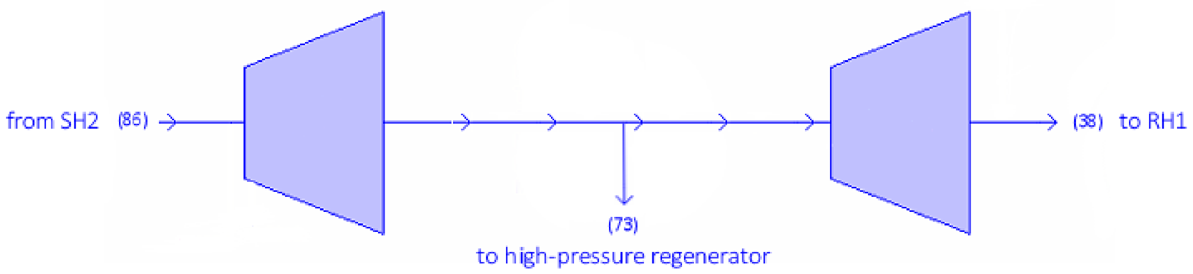


Figure 7. 6 – High-pressure turbine

The following **Table 7. 5** indicates its fluxes main thermodynamic properties.

Flux 86					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	64.40	101.20	119.60	138.00	156.40
T [°C]	450.0	450.0	450.0	450.0	450.0
Sup [°C]	169.8	138.2	125.6	114.5	104.5
h [kJ/kg]	3296.96	3241.54	3211.81	3180.65	3147.99
m [kg/s]	69.46	61.60	60.52	59.20	58.73

Flux 73		
	Evaporation Pressure [bar]	
	110	130
p [bar]	66.40	49.50
T [°C]	389.9	329.2
Sup [°C]	107.6	65.9
h [kJ/kg]	3141.16	3015.08
m [kg/s]	2.608	1.287

Flux 38					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	14.00	27.05	27.05	27.05	27.05
T [°C]	257.4	279.3	260.5	245.2	232.0
Sup [°C]	62.3	51.1	32.3	17.0	3.8
h [kJ/kg]	2944.68	2950.88	2899.62	2855.03	2814.07
m [kg/s]	68.89	58.99	59.23	59.20	58.73

Table 7. 5 – High-pressure turbine thermodynamic output

The following **Table 7. 6** reports data about energy obtained by the high-pressure turbine and its efficiency.

	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
Generator power [MW]	65.13	16.62	17.95	18.46	18.79

Generator efficiency [%]	98.33	97.41	97.46	97.48	97.49
Gearbox efficiency [%]	98.5	98.5	98.5	98.5	98.5
Shaft Power [MW]	66.24	17.37	18.70	19.23	19.56
Group overall efficiency [%]	86.08	85.45	85.75	85.52	84.19

Table 7. 6 – Energy obtained from the high-pressure turbine

Finally, a **Table 7. 7** about the main high-pressure turbine dimensions is presented below.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Overall ST and Generator Length [m]	18.33	12.56	12.74	12.83	12.85
Overall ST and Generator Width [m]	4.878	2.590	2.626	2.640	2.648
Overall ST and Generator Weight [kg]	260'200	63'450	66'550	67'900	68'500
Foundation Length [m]	20.65	13.89	14.08	14.32	14.33
Foundation Width [m]	5.853	3.108	3.152	3.168	3.178

Table 7. 7 - Main high-pressure turbine dimensions

The reference plant dimensioning data presented in the high-pressure turbine Table 7.7 refers to the whole turbine body since its subdivision is only fictional. Thus, in the low-pressure turbine data about the principal dimensioning parameters, only innovative plant values would be presented.

The **Figure 7. 7** below, however, shows the low-pressure turbine and all the bleedings for low-pressure regenerators and air pre-heaters.

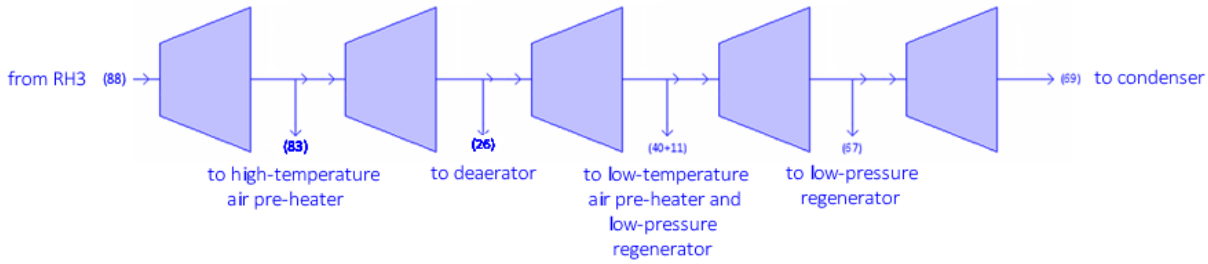


Figure 7. 7 - Low-pressure turbine

Flux 88					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	14.00	25.00	25.00	25.00	25.00
T [°C]	257.4	450.0	450.0	450.0	450.0
Sup [°C]	62.3	226.1	226.1	226.1	226.1
h [kJ/kg]	2944.68	3351.05	3351.05	3351.05	3351.05
m [kg/s]	63.34	61.58	61.12	61.05	60.77

Flux 83

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	14.00	14.00	14.00	14.00	14.00
T [°C]	257.1	365.2	365.3	365.3	365.3
Sup [°C]	62.3	170.2	170.2	170.2	170.2
h [kJ/kg]	2944.12	3183.34	3183.37	3183.38	3183.39
m [kg/s]	5.553	4.421	4.421	4.421	4.421

Flux 26

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	4.76	4.76	4.76	4.76	4.76
T [°C]	154.2	231.8	231.8	231.8	231.8
Sup [°C]	4.2	81.8	81.8	81.8	81.8
h [kJ/kg]	2754.87	2923.59	2923.59	2923.59	2923.67
m [kg/s]	0.133	0.119	0.133	0.152	0.149

Flux 40 + 11

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	4.51	4.51	4.51	4.51	4.51
T [°C]	149.6	227.1	227.1	227.1	227.1
Sup [°C]	1.6	79.1	79.1	79.1	79.1
h [kJ/kg]	2746.42	2914.77	2914.83	2914.84	2914.87
m [kg/s]	(only 11) 4.797	6.607	6.560	6.552	6.490

Flux 67

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	0.79	0.79	0.79	0.79	0.79
T [°C]	93.17	93.17	93.17	93.17	93.17
x [-]	0.925	0.977	0.977	0.977	0.977
h [kJ/kg]	2494.60	2611.90	2611.95	2611.97	2612.03
m [kg/s]	(67+40) 6.369	3.028	2.989	2.982	2.933

Flux 69

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	0.07	0.07	0.07	0.07	0.07
T [°C]	39.03	39.03	39.03	39.03	39.03
x [-]	0.854	0.895	0.895	0.895	0.895
h [kJ/kg]	2220.02	2319.74	2319.78	2319.77	2319.78
m [kg/s]	52.59	47.40	47.01	46.93	46.77

Table 7. 8 - Low-pressure turbine thermodynamic output

The values reported in the row “Sup” indicate the difference between the superheated fluid temperature and the evaporation temperature, at the same pressure.

In the case of reference plant, the subdivision between high and low pressure turbine is only fictional and the high pressure turbine is assume to be the one which elaborates the pressure gap from 70 bar to 40 bar.

Finally, the following tables show the main energy obtainable by the low-pressure group and its principal dimensioning data. The data analysed so far confirm the general trend of power produced increase (generator power) with the increase of evaporation pressure, even if its increase diminish for every evaporation pressure raising.

	Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Generator power [MW]	53.61	53.18	53.10	52.86
Generator efficiency [%]	98.20	98.19	98.19	98.19
Shaft Power [MW]	54.60	54.15	54.07	53.84
Group overall efficiency [%]	91.57	91.56	91.56	91.56

Table 7. 9 - Energy obtained from the low-pressure turbine

	Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Overall ST and Generator Length [m]	16.30	16.27	16.26	16.24
Overall ST and Generator Width [m]	4.793	4.781	4.779	4.774
Overall ST and Generator Weight [kg]	228'550	227'150	226'900	226'150
Foundation Length [m]	18.73	18.69	18.68	18.66
Foundation Width [m]	5.752	5.737	5.735	5.729

Table 7. 10 - Main low-pressure turbine dimensions

7.4. FLUE GASES CYCLE

The flue gases cycle comprehends all the heat exchangers within the boiler as well as the air pre-heaters. The tables with red colour would indicate the fluxes related to air or flue gases, while waste is the only flow with the colour orange.

7.4.1. Combustor

The combustor is intended as the part where air and fuel react to create flue gases, which exit at the top of it, entering the main convective section. The radiative evaporator or waterwall is also located within the combustor and in here, the 95% of the water vaporises, becoming saturated vapour.

The following **Figure 7. 8** shows the furnace and fluxes entering and exiting it. The further **Table 7. 11** would illustrate their main thermodynamic properties.

Where the bottom and left fluxes are related to the pre-heated air, flow *47* represents the recirculated flue gases. Finally, flow on the top of the furnace is obviously related to flue gases entering the main convective section.

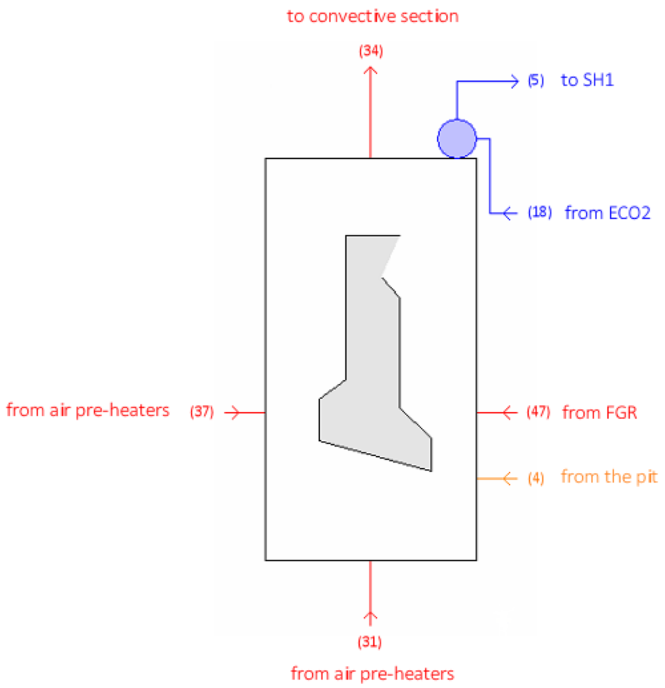


Figure 7. 8 – Furnace

Flux 18					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	69.99	110.65	129.60	149.54	169.36
T [°C]	264.3	308.5	320.6	331.9	342.0
Sub [°C]	21.5	10.0	10.0	10.0	10.0
h [kJ/kg]	1155.81	1391.23	1462.44	1532.53	1598.95
m [kg/s]	63.25	55.34	53.82	52.64	51.76

Flux 5					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	69.99	110.65	129.64	149.54	169.36
T [°C]	285.8	318.5	330.6	341.9	352.0
Sup [°C]	0.0	0.0	0.0	0.0	0.0
h [kJ/kg]	2773.29	2707.87	2667.65	2616.17	2553.72
m [kg/s]	62.63	54.79	53.28	52.12	51.25

Flux 4					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	1.0132	1.0132	1.0132	1.0132	1.0132
T [°C]	25.0	25.0	25.0	25.0	25.0
m [kg/s]	19.34	19.34	19.34	19.34	19.34

Flux 31					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	1.075	1.075	1.075	1.075	1.075
T [°C]	165.0	165.0	165.0	165.0	165.0
m [kg/s]	68.01	68.01	68.01	68.01	68.01

Flux 37					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	1.075	1.075	1.075	1.075	1.075
T [°C]	165.0	165.0	165.0	165.0	165.0
m [kg/s]	45.34	45.34	45.34	45.34	45.34

Flux 47					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	1.066	1.066	1.066	1.066	1.066
T [°C]	196.9	196.9	192.2	196.9	196.5
m [kg/s]	22.61	22.61	22.61	22.61	22.61

Flux 34					
	Evaporation Pressure [bar]				
	70 (ref plant)	110	130	150	170
p [bar]	1.0126	1.0126	1.0126	1.0126	1.0126
T [°C]	650.0	799.5	839.0	878.0	916.0
m [kg/s]	152.0	152.0	152.0	152.0	152.0

Table 7. 11 - Furnace thermodynamic output

Obviously, the water quantity decreases with the flue gases outlet temperature increasing and so with the evaporation pressure. Air, flue gases recirculated and produced quantities are constant because they all depends on the fuel mass flow, which is constant in order to maintain the power introduced fixed to 200 MW and the required air excess (or oxygen content in flue gases), which is a data assumption.

The following **Table 7. 12** shows the results about ashes produced within the combustor. Ashes are partly composed by ashes already present within the waste and partly by unburned carbon, considered separately.

Ashes					
	Evaporation Pressure [bar]				
	70 (cfr plant)	110	130	150	170
Bottom ash flow [kg/s]	2.937	2.937	2.937	2.937	2.937
Unburnt carbon flow in bottom ash [kg/s]	0.0758	0.0758	0.0758	0.0758	0.0758
Fly ash flow [kg/s]	0.3267	0.3267	0.3267	0.3267	0.3267
Unburnt carbon flow in fly ash [kg/s]	0.0042	0.0042	0.0042	0.0042	0.0042
Total unburnt carbon flow in ash [kg/s]	0.08	0.08	0.08	0.08	0.08
Combustion efficiency [%]	98.64	98.64	98.64	98.64	98.64

Table 7. 12 - Furnace's ashes composition

Even in this case, ashes quantity and composition does not depend on the evaporation pressure. The **Table 7. 13** above shows the energy balance operated by Thermoflex within the component combustor.

Energy balance					
	Evaporation Pressure [bar]				
	70 (cfr plant)	110	130	150	170
Fuel Input [kW]	199'979	199'979	199'979	199'979	199'979
Air Input [kW]	16'120	16'120	16'120	16'120	16'120
Recirculation gas into furnace [kW]	4'340	4'341	4'345	4'341	4'445
Total Energy in [kW]	**220'439**	**220'440**	**220'445**	**220'441**	**220'544**
Flue Gas [kW]	111'652	140'964	148'857	156'702	164'388
Heat transfer to waterwall [kW]	101'364	72'173	64'327	56'516	48'983
Heat losses [kW]	2'533.9	1'804.1	1'608.0	1'412.7	1'224.2
Unburnt carbon in ash [kW]	2'699.3	2'699.3	2'699.3	2'699.3	2'699.3
Bottom ash sensible heat [kW]	1'963.9	2'512.6	2'655.8	2'795.6	2'930.0
Fly ash sensible heat [kW]	218.2	279.2	295.1	310.6	325.6
Total energy out [kW]	**220'431**	**220'432**	**220'443**	**220'436**	**220'550**
Heat Balance Error [%]	**0.0037**	**0.0036**	**0.0008**	**0.0021**	**-0.0029**

Table 7. 13 - Furnace energy balance

The *Table 7. 14* about the combustor aims to provide the mole composition of flue gases and confronting them with the composition calculated with the method explained in Chapter 2, showing that the calculation with chemical equilibrium allows to obtain reasonably right results.

Flue gases composition						
	Evaporation Pressure [bar]					Composition calculated
	70 (cfr plant)	110	130	150	170	
Mole percent N_2 [%]	66.16	66.16	66.16	66.16	66.16	66.18217
Mole percent O_2 [%]	6.023	6.023	6.023	6.023	6.023	5.974081
Mole percent CO_2 [%]	9.556	9.556	9.556	9.556	9.556	9.652456
Mole percent H_2O [%]	17.42	17.42	17.42	17.42	17.42	17.38963
Mole percent SO_2 [%]	0.0053	0.0053	0.0053	0.0053	0.0053	0.00524
Mole percent Ar [%]	0.7955	0.7955	0.7955	0.7955	0.7955	0.795673
SO3 in flue gas [ppm]	2.434	2.434	2.434	2.434	2.434	-
HCl in exit gas [kg/h]	148.9	148.9	148.9	148.9	148.9	-

Table 7. 14 - Flue gases composition obtained by the program and calculated

The higher error is visible on CO_2 and O_2 percentage calculations due to the presence of unburned carbon within ashes. Thus, not all the carbon becomes carbon dioxide, so the free oxygen increases while CO_2 decreases.

Finally, the last **Table 7. 15** shows the principal dimensioning output obtained for the only furnace and the whole boiler.

Furnace and Boiler					
	Evaporation Pressure [bar]				
	70 (cfr plant)	**110**	**130**	**150**	**170**
Furnace width[m]	25.81	19.29	17.82	16.39	15.00
Furnace depth [m]	12.900	9.646	8.912	8.195	7.501
Furnace height [m]	77.43	57.88	53.47	49.17	45.01
Furnace volume [m3]	30'369.00	12'684.00	10'005.00	7'779.00	7'501.00
Furnace dry weight [kg]	3'920'000.00	2'323'000.00	2'037'000.00	1'770'000.00	1'523'000.00
Boiler length [m]	69.39	76.46	69.37	71.72	123.50
Boiler width [m]	17.57	22.16	22.50	24.86	26.26
Boiler height [m]	6.996	6.966	7.996	6.975	6.981
Boiler dry weight [kg]	3'054'000	6'036'000	4'330'000	4'455'000	5'907'000

Table 7. 15 - Furnace and Boiler main dimensioning parameters

The boiler overall dimensions results higher than the traditional waste-to-energy plant. The reason lays in the simulation programme Thermoflex, which uses the carbon boiler models to calculate and dimension the waste-to-energy boiler. The programme, in fact, simulate a horizontal HRSG for the convective section of the boiler and the three passages between radiative and convective section are designed in vertical, which explains the height in the table above.

7.4.2. Convective section

The convective section comprehends all the heat exchangers within the flue gases cycle except for the air pre-heaters.

The **Table 7. 16** below shows the synthesis output of the boiler simulation, considering the overall exchange area subdivided by the type of heat exchangers within the boiler.

Heat transfer surface area					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Overall surface area [m^2]	23'598	44'880	44'830	44'784	47'659
Economiser [m^2]	9'818	24'332	25'649	25'993	27'220
Waterwall [m^2]	7'381	4'131	3'527	2'982	2'498
Convective Evaporator [m^2]	284.7	213.8	162.0	167.7	148.5
Superheaters [m^2]	6'114	7'745	6'669	6'150	6'410
Reheaters[m^2]	-	8'458	8'823	9'492	11'382

Table 7. 16 - Boiler heat transfer area

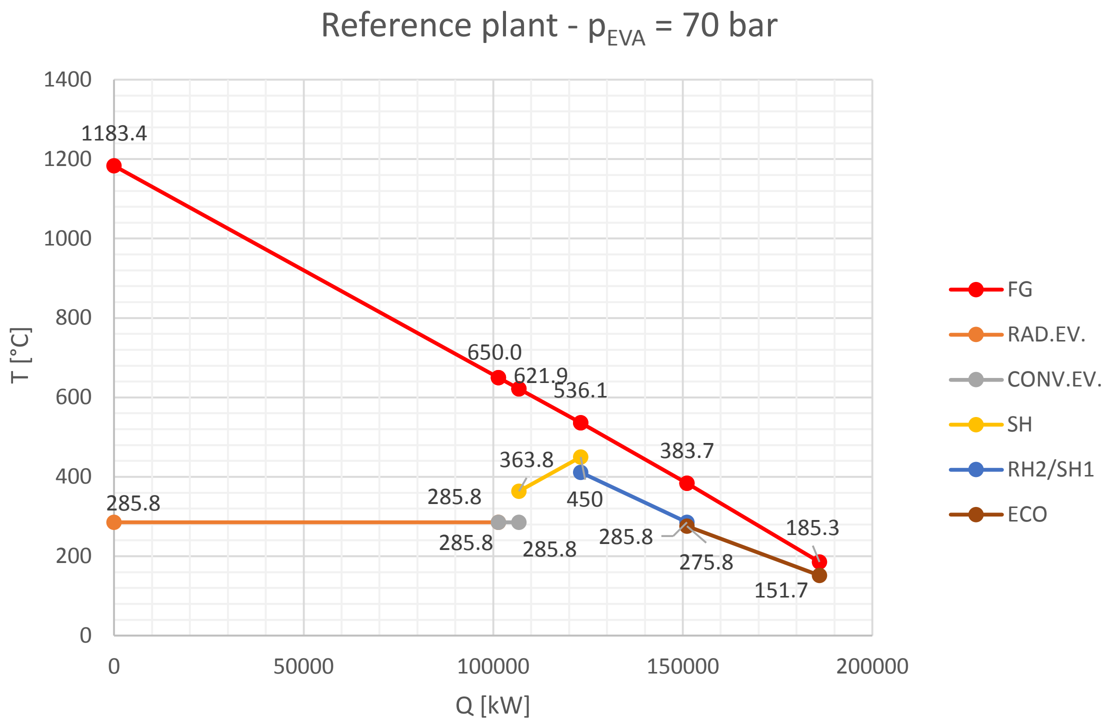
While the next **Table 7. 17** shows the heat exchanged associated to the heat exchangers mentioned above.

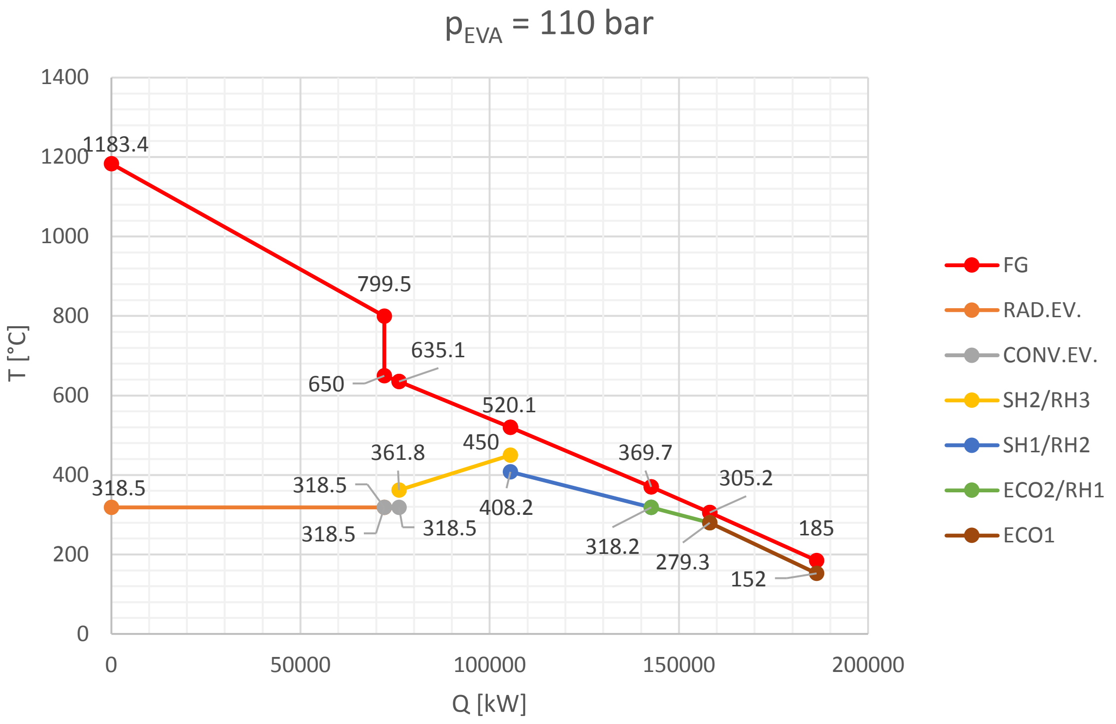
Heat transfer					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Overall heat transferred [kW]	186'013	180'406	175'635	172'700	169'849
Economiser [kW]	34'881	37'752	43'932	49'719	53'523

Waterwall [kW]	101'364	72'173	64'327	56'516	48'983
Convective Evaporator [kW]	5'415.0	3'867.0	3'435.0	3'020.0	2'613.1
Superheaters [kW]	44'353	41'535	42'478	42'526	44'617
Reheaters[kW]	-	25'079	21'463	20'919	20'113

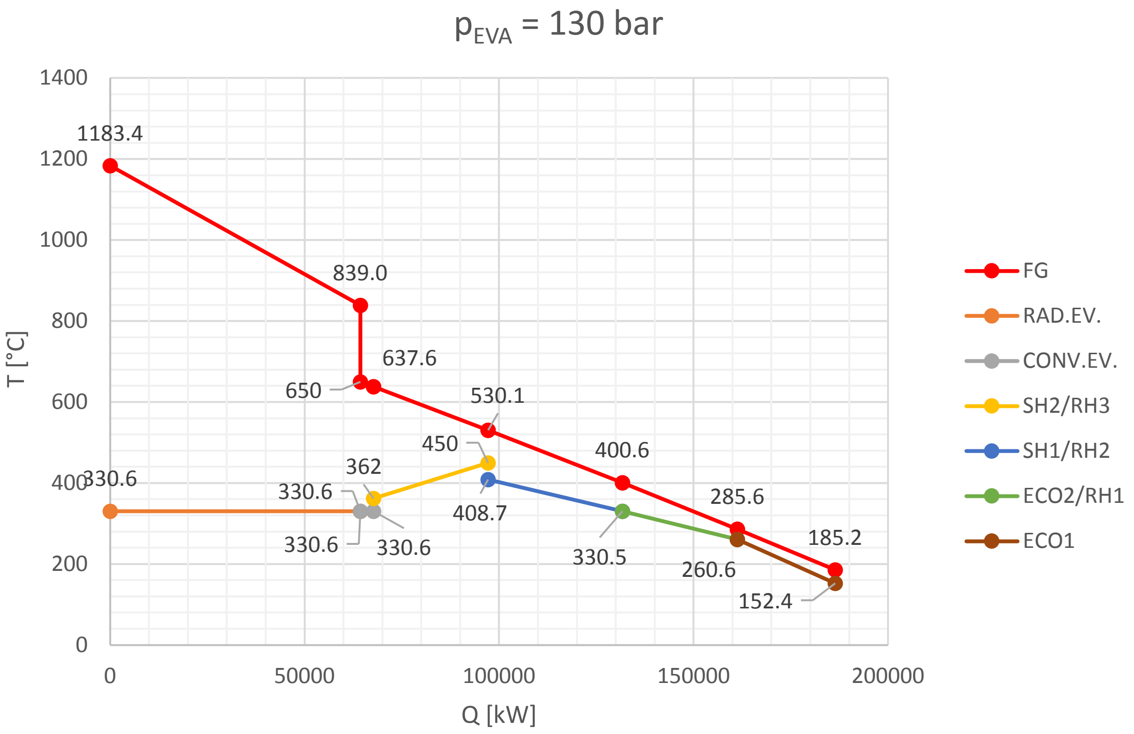
Table 7. 17 - Heat transferred within the boiler

The following **Figure 7. 9** shows the five boiler T-Q diagrams, where flue gases exchange heat with vapour/water. For completeness reasons, radiative evaporator (*RAD.EV.*) have been added too.





p_{EVA} = 110 bar



p_{EVA} = 130 bar

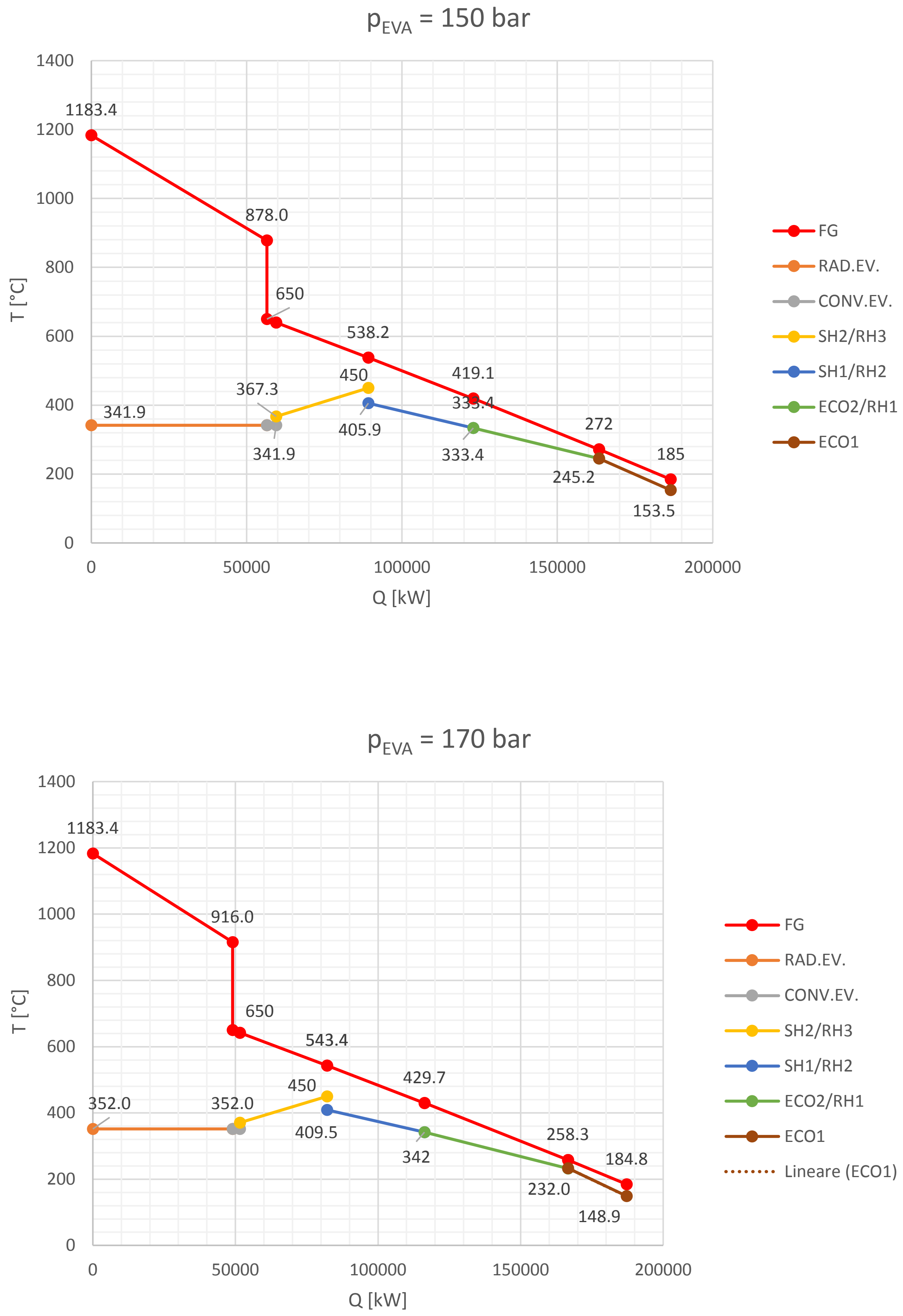


Figure 7. 9 - T-Q diagram of the boiler

The term *FG* indicates the flue gases temperatures, while other lines shows water/vapour evolution. In order to maintain the lower gap possible between flue gases and water/vapour, the heat exchanged within *ECO1* reduces progressively with evaporation pressure increase and increases the heat exchanged in the previous heat exchangers in parallel.

Moreover, the difference between vapour temperature in heat exchangers in parallel (especially in *SH1/RH2* outlet and *SH2/RH3* inlet due to the presence of de-super-heaters) is of 10°C maximum, so they have not been reported in order not to add negligible results to yet crowded diagrams.

The reference plant shows its incapacity to reach the required temperature of flue gases at boiler outlet of 185°C because of the vapour quantity produced and the constraints of flue gases temperature. The problem lays in the gas quenching absence in the confrontation plant. This requires the limit of 650°C as flue gases temperature exiting the furnace. However, lower their temperature, higher the total steam quantity produced within the furnace and so higher the heat exchanged quantity necessary to heat the steam/water up. The result is a too great quantity of water to be heated up by a too modest quantity of flue gases, which fails to make the water reach the required evaporation temperature minus the sub-cooling temperature gap. The solution is to consent the evaporator to act as an economiser, effectuating part of the condensation process within the evaporator (both radiative and convective).

The further tables shows the dimensioning outcome of every heat exchanger within the convective section.

	EVA - Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Tube material [-]	Carbon Steel	T91	T91	T91	T91
Number of tube rows [-]	6	3	2	2	2
Number of tubes per row [-]	60	88	100	104	92
Number of rows per waterside flow pass [-]	6	3	2	2	2
Gas path transverse width [m]	15.45	20.76	22.26	23.7	25.18

Table 7. 18 - EVA dimensioning output

	SH2 - Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Tube material [-]	T91	T22	T91	T22	T22
Number of tube rows [-]	14	20	18	16	16
Number of tubes per row [-]	134	106	116	130	136
Number of rows per waterside flow pass [-]	1	1	1	1	1
Gas path transverse width [m]	15.45	12.18	14.03	15.05	16.24

Table 7. 19 - SH2 dimensioning output

	RH3 - Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Tube material [-]	T22	T22	T22	T22
Number of tube rows [-]	24	18	18	18
Number of tubes per row [-]	68	72	76	74
Number of rows per waterside flow pass [-]	4	3	6	6

Gas path transverse width [m]	8.639	8.237	8.649	8.940

Table 7. 20 - RH3 dimensioning output

	SH1 - Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Tube material [-]	T22	T91	T91	T91	T91
Number of tube rows [-]	48	54	40	32	32
Number of tubes per row [-]	132	116	128	142	150
Number of rows per waterside flow pass [-]	1	1	1	1	1
Gas path transverse width [m]	15.45	13.64	15.45	16.59	18.26

Table 7. 21 - SH1 dimenioning output

	RH2 - Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Tube material [-]	T91	T91	T91	T91
Number of tube rows [-]	60	40	32	28
Number of tubes per row [-]	58	58	60	58
Number of rows per waterside flow pass [-]	3	4	4	2
Gas path transverse width [m]	7.182	6.813	7.109	6.926

Table 7. 22 - RH2 dimensioning output

	ECO2 - Evaporation Pressure [bar]				
	70 (ref plant) (ECO)	**110**	**130**	**150**	**170**
Tube material [-]	T22	T22	T22	T22	T22
Number of tube rows [-]	72	40	60	72	240
Number of tubes per row [-]	196	156	176	192	78
Number of rows per waterside flow pass [-]	0.5	1	1	1	2
Gas path transverse width [m]	15.45	12.91	14.16	15.59	16.55

Table 7. 23 - ECO2 dimensioning output

	RH1 - Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Tube material [-]	Carbon Steel	Carbon Steel	Carbon Steel	Carbon Steel
Number of tube rows [-]	78	90	104	168
Number of tubes per row [-]	66	46	68	72
Number of rows per waterside flow pass [-]	13	18	8	21
Gas path transverse width [m]	7.911	4.333	8.104	8.638

Table 7. 24 - RH1 dimensioning output

	ECO1 - Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Tube material [-]	T22	T22	Carbon Steel	Carbon Steel
Number of tube rows [-]	112	96	78	66
Number of tubes per row [-]	256	276	302	308

Number of rows per waterside flow pass [-]	1	1	0.5	1
Gas path transverse width [m]	20.82	22.26	23.70	25.18

Table 7. 25 - ECO1 dimensioning output

Finally, the following tables show the results about heat exchangers dimensioning. Calculations refers to a previous plant configuration, but since the procedure is proven effective, the proof that the programme executes a correct dimensioning of heat exchangers becomes independent on the configuration of the value of evaporation pressure of the plant. The results are about the *RH3* heat exchanger and the following **Table 7. 29** shows the water side principal data.

RH3 water side properties						
Flux	$T_{H2O,in}$	$p_{H2O,in}$	$T_{H2O,out}$	$p_{H2O,out}$	T_{wall}	T_B **[°C]**
[-]	[°C]	[bar]	[°C]	[bar]	[°C]	Average temperature
[-]	[-]	[-]	[-]	[-]	[-]	$(T_{H2O,in}+T_{H2O,out})/2$
direct current	348.6	25.5	450	25	408.6	399.3

Table 7. 26 - RH3 dimensioning, water side properties

RH3 water side calculation parameters	
$T_{calc,H2O}$	$p_{calc,H2O}$
[°C]	[bar]
403.95	25.25

Table 7. 27 - RH3 dimensioning, water side calculation parameters

μ_{H2O}	cp_{H2O}	λ_{H2O}	ρ_{H2O}	w_{H2O}	D_{in}
[Pa*s]	[J/(kg*°C)]	[W/(m*°C)]	[kg/m^3]	[m/s]	[m]
dynamic viscosity	specific heat	thermal conductivity	density	mean velocity	internal diameter
0.000013484	1995.456	0.0270106	13.60402467	9.48	0.034434

Table 7. 28 - RH3 dimensioning, water side data

The **Table 7. 29** below shows the results obtained from water side data and properties as well as the convective heat exchanger coefficient obtained by Thermoflex.

Pr_{H2O}	Re_{H2O}	Nu_{H2O}	h_{H2O} **[W/(m^2*°C)]**
Prandtl number	Reynolds number	Nusselt number	Convective coefficient water side
0.996154425	329339.9987	594.852488	466.6121453
0.9547	112879	249.6	411.5

Table 7. 29 - RH3 dimensioning, water side characteristic numbers and convective coefficient

The underlined numbers indicates the one calculated by Thermoflex. The Reynolds and Nusselt numbers appears to be completely different from the one calculated by the software, but the convective coefficient seems more similar.

The same procedure can be applied to the gas side.

RH3 gas side properties				
Flux	**$T_{FG,in}$**	**$p_{FG,in}$**	**$T_{FG,out}$**	**$p_{FG,out}$**
[-]	[°C]	[bar]	[°C]	[bar]
[-]	[-]	[-]	[-]	[-]
direct current	635	1.011	500	1.006

Table 7. 30 - RH3 dimensioning, gas side properties

RH3 gas side calculation parameters	
$T_{calc,FG}$	**$p_{calc,FG}$**
[°C]	[bar]
567.5	1.0085

Table 7. 31 - RH3 dimensioning, gas side calculation parameters

μ_{FG}	**cp_{FG}**	**λ_{FG}**	**η_{FG}**	**w_{FG}**	**D_{out}**
[Pa*s]	[J/(kg*°C)]	[W/(m*°C)]	[m^2/s]	[m/s]	[m]
dynamic viscosity	specific heat	thermal conductivity	kinematic viscosity	mean velocity	outer diameter
0.000027125	1171	0.0416	0.000048422	9.734	0.03865

Table 7. 32 - RH3 dimensioning, gas side data

s_1	**s_2**	**a**	**b**	**ψ**	**w_m**	**L**
[m]	[m]	[-]	[-]	[-]	[m/s]	[m]
0.0902	0.079	2.333764554	2.043984	0.663463	14.6715	0.060711278

Table 7. 33 - RH3 dimensioning, gas side additional properties

Pr_{FG}	**Re_{FG}**	**$Nu_{lam,FG}$**	**$Nu_{turb,FG}$**	**$N_{singletube,FG}$**	**f_{corr}**	**Nu_{FG}**	**h_{FG} [W/(m^2*°C)]**
0.7635	18395.1	82.312	85.8419	119.2289	1.32616	158.12	108.3
0.7329	7450					72.98	106.8

Table 7. 34 - RH3 dimensioning, gas side characteristic numbers and convective coefficient

Even in this case, the Prandtl number and the convective coefficient seems to be roughly the same, while the other two characteristic numbers differ greatly. This would depend on the different calculation approach, which thus can not be known since it is part of the intellectual property and know-how of the firm producer of the software.

Now the overall heat transfer coefficient can be calculated.

$Foul_{FG}$	**$Foul_{H2O}$**	**$Q_{exchanged}$**	**λ_{steel}**	**L_t**
[m^2*°C/W]	[m^2*°C)/W]	[W]	[W/(m*°C)]	[m]
0.015	8.806E-05	14250000	36.86	8.992

Table 7. 35 - RH3 dimensioning, general data

$\Delta T_{ml,DC}$ [K]	**U_{ext} [W/(m^2*°C)]**	**A_{ext} [m^2]**	**N_{tubes}**
135.4443376	37.32063327	2819.064458	2581.958973
	-	-	2520

Table 7. 36 - RH3 dimensioning, number of tubes

The value of tube numbers calculated is included in the tolerance interval, since Thermoflex uses a different methodology for the calculation and takes into account other factors which can not been known for the reasons explained before.

7.4.3. Flue gases splitting

Flue gases exiting the deaerator enters the splitter represented in the **Figure 7. 10**.

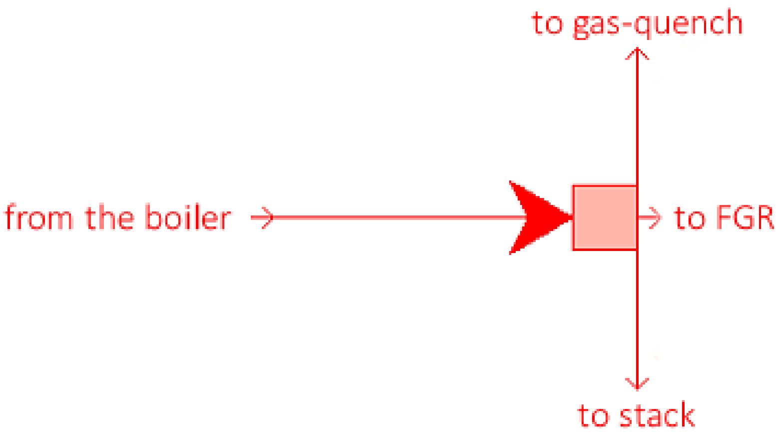


Figure 7. 10 - Flue gases splitter

This device operates the splitting of flue gases exiting the first economiser in flue gases designated to gas quenching, to flue gases recirculation within the combustor and to be expelled through the stack, after having heated up the condensate.

The following **Table 7. 37** shows the outcome of flux splitting, which do not depend on the evaporation pressure, since the flue gases temperature at deaerator outlet is roughly the same. The only variable that depends on the evaporation pressure is the gas destined to gas quenching because the temperature of flue gases at furnace outlet increases with the evaporation pressure.

from the Boiler					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	1.0019	1.0005	1.0007	1.0006	1.0003
T [°C]	185.3	185.0	185.2	185.0	184.6
m [kg/s]	152.0	204.9	219.2	233.3	247.1

Gas-Quench					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	1.0019	1.0005	1.0007	1.0006	1.0003
T [°C]	185.3	185.0	185.2	185.0	184.6
m [kg/s]	0.0	52.89	67.15	81.28	95.08

FGR					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**

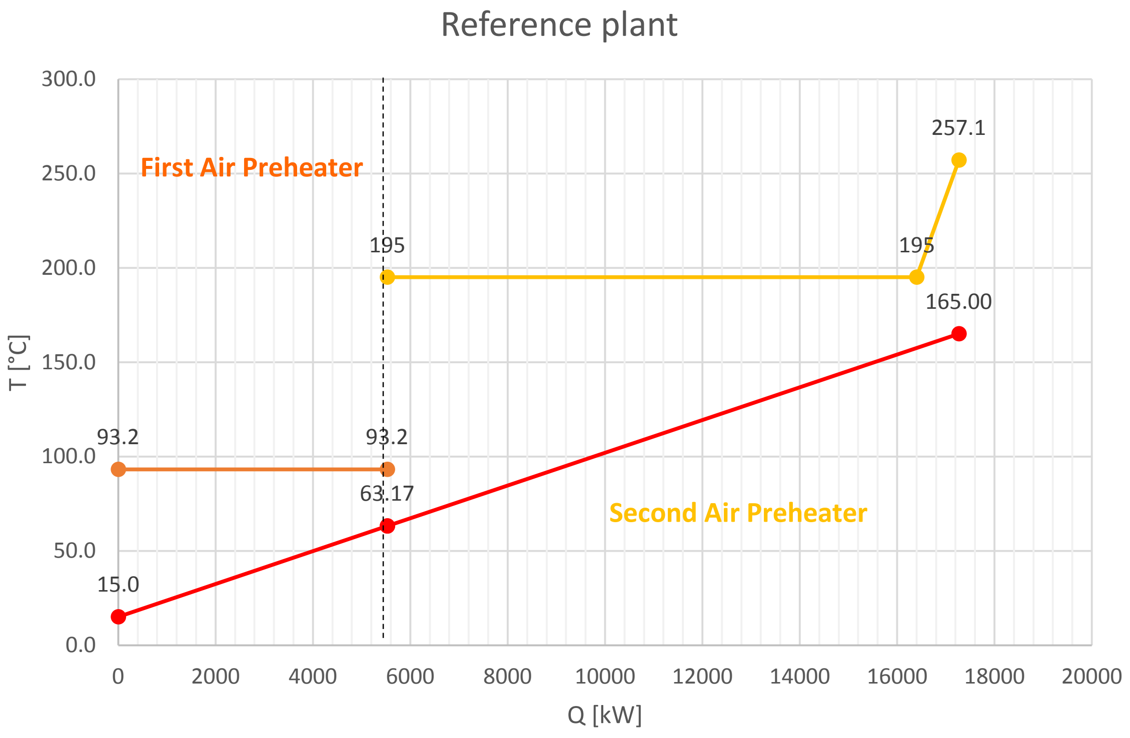
p [bar]	1.0019	1.0005	1.0007	1.0006	1.0003
T [°C]	185.3	185.0	185.2	185.0	184.6
m [kg/s]	22.40	22.40	22.40	22.40	22.40

Stack					
	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
p [bar]	1.0019	1.0005	1.0007	1.0006	1.0003
T [°C]	185.3	185.0	185.2	185.0	184.6
m [kg/s]	129.6	129.6	129.6	129.6	129.6

Table 7. 37 - Splitter thermodynamic output

7.4.4. Air pre-heaters

Finally, the two air pre-heaters are presented. Again, the heat exchange within these two heat exchangers do not depend on the evaporation pressure, so the following diagrams are represented only for the confronting and innovative configuration plant.



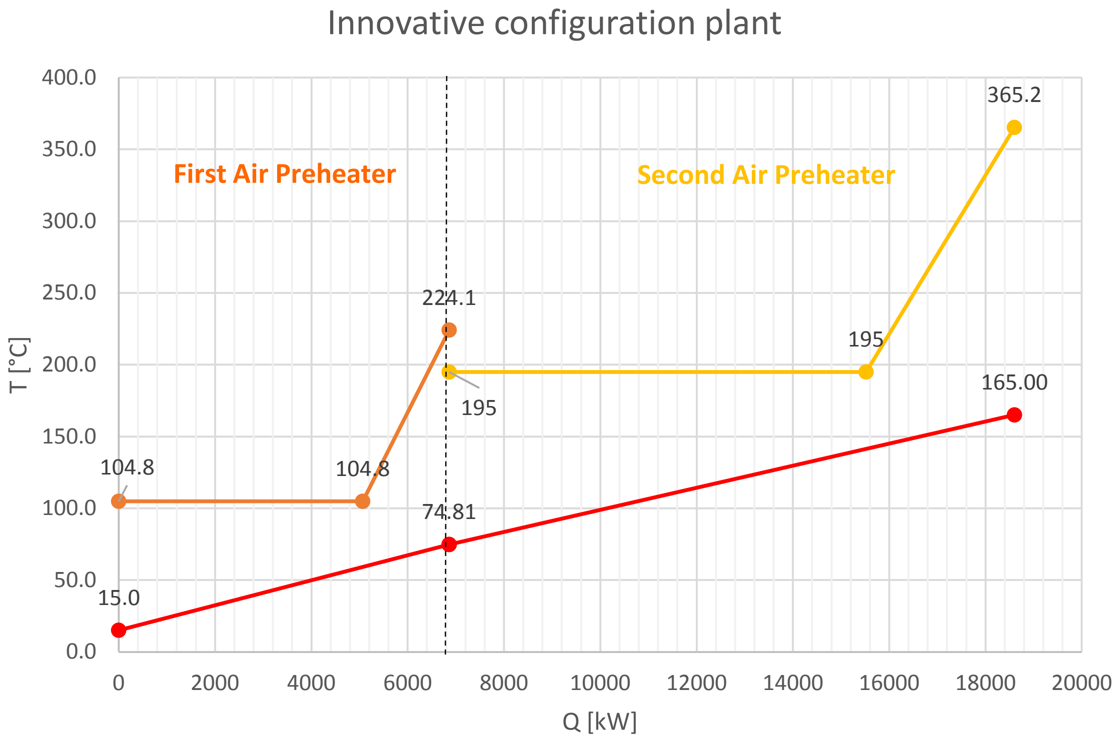


Figure 7. 11 - T-Q diagram of the air preheating line

7.5.AUXILIARIES

The **Table 7. 38** below shows the overall power required from auxiliaries in every plant. Moreover, the following figures present the auxiliaries percentage consumption.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
High-pressure Turbine [kW]	239.30	61.24	65.97	67.85	68.76
Low-pressure Turbine [kW]		197.00	195.40	195.10	194.90
HRSG Assembly [kW]	76.53	76.41	76.50	76.41	76.19
Furnace [kW]	383.70	383.70	383.70	383.70	383.70
Electrostatic Precipitator [kW]	362.70	434.30	452.10	468.50	483.30
Fabric Filter [kW]	127.70	127.70	127.80	127.70	127.30
Wet-Dry cooling tower [kW]	523.30	491.00	487.00	486.20	489.00
Condensate pump [kW]	176.50	156.00	153.70	153.50	137.50
Feed pump [kW]	870.70	1'054.60	1'211.00	1'572.30	1'771.00
Cooling water pump [kW]	1'089.70	1'002.10	990.30	987.60	987.00
Gas-quench fan [kW]	-	166.90	207.70	253.50	302.20
FGR fan [kW]	361.20	369.10	368.10	368.80	369.80
Stack fan [kW]	1'241.60	1'282.50	1'277.80	1'276.30	1'284.70
Miscellaneous [kW]	1'551.20	1'602.80	1'611.30	1'615.60	1'617.40
Total [kW]	7004.13	7405.35	7608.37	8033.06	8292.75

Table 7. 38 - Total auxiliaries power consumption

Fan *42* represents the fan dedicated to flue gases discharged through the stack while fan *22* is the one dedicated to FGR. Fan *38*, on the other hands, increase the pressure of flue gases recirculated for gas quench purposes. Moreover, pump *6* is the one dedicated to condensate pressure increasing, while the feed pump *18* is the one that fulfil the highest pressure gap and so absorb the highest quantity of power. Finally, pump *17* is dedicated to cooling fluid movement within the wet-dry evaporative tower.

ST assembly indicates the turbine body subdivided in ST Assembly [1], which corresponds to high-pressure turbine and ST Assembly [2], which is the low-pressure turbine.

Values indicates beside every “piece” of the diagrams, other than the percentage of the total power absorbed, indicate its absolute value, expressed in kW.

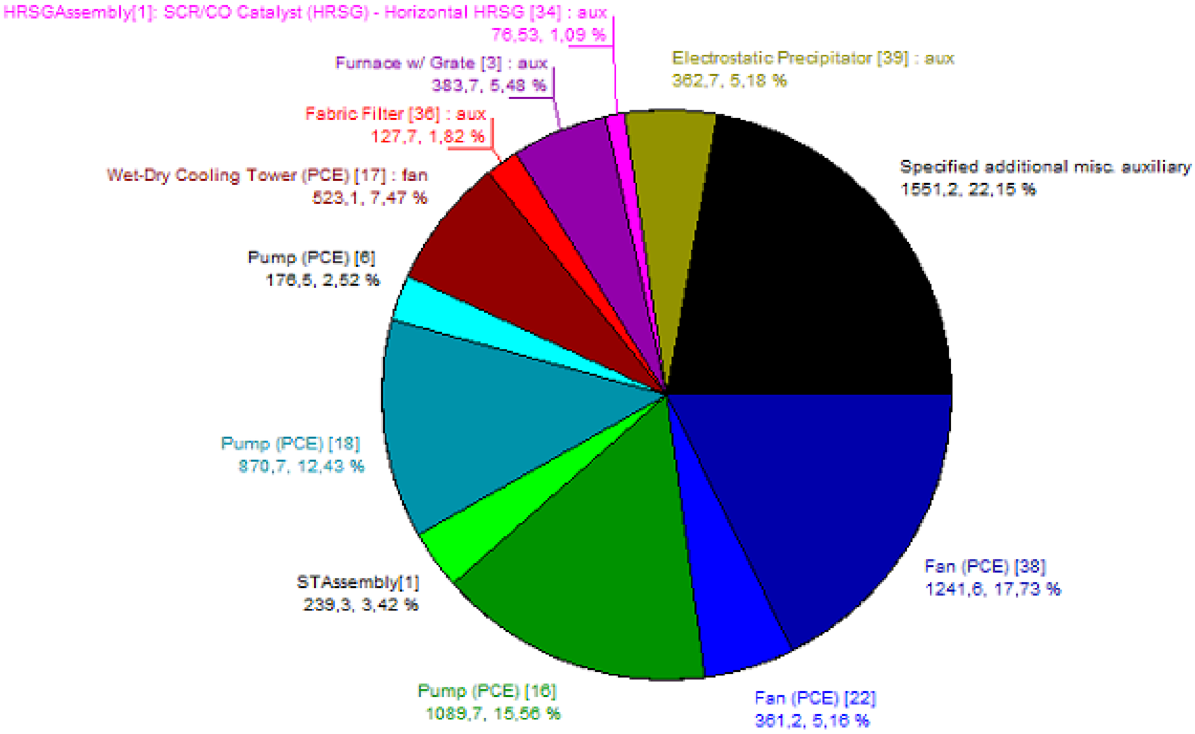


Figure 7. 12 - Reference plant, power absorbed by auxiliaries in percentage

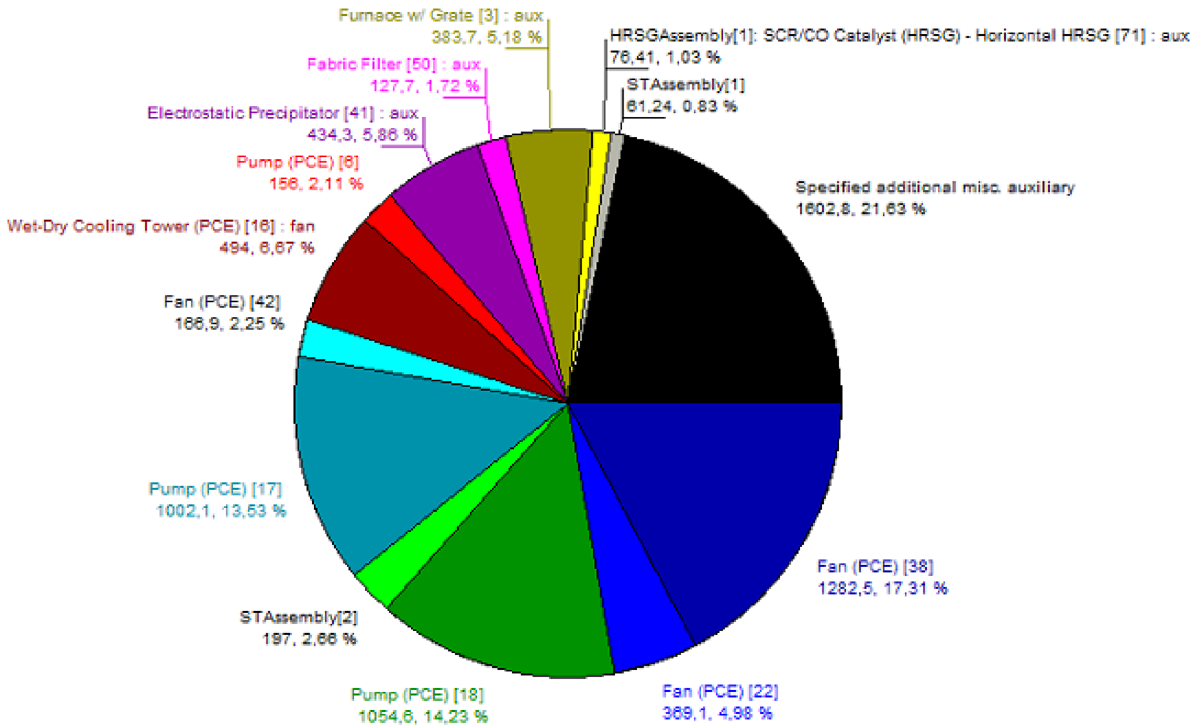


Figure 7. 13 - p_{EVA} = 110 bar, power absorbed by auxiliaries in percentage

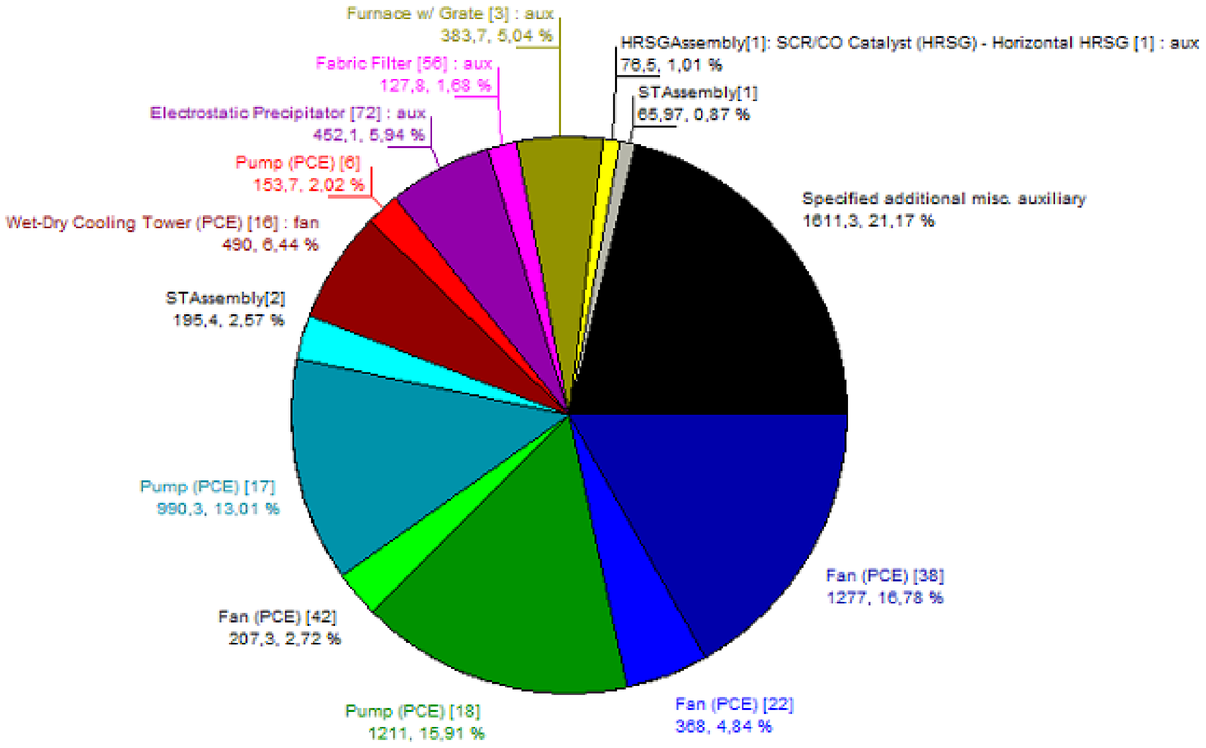


Figure 7. 14 - p_{EVA} = 130 bar, power absorbed by auxiliaries in percentage

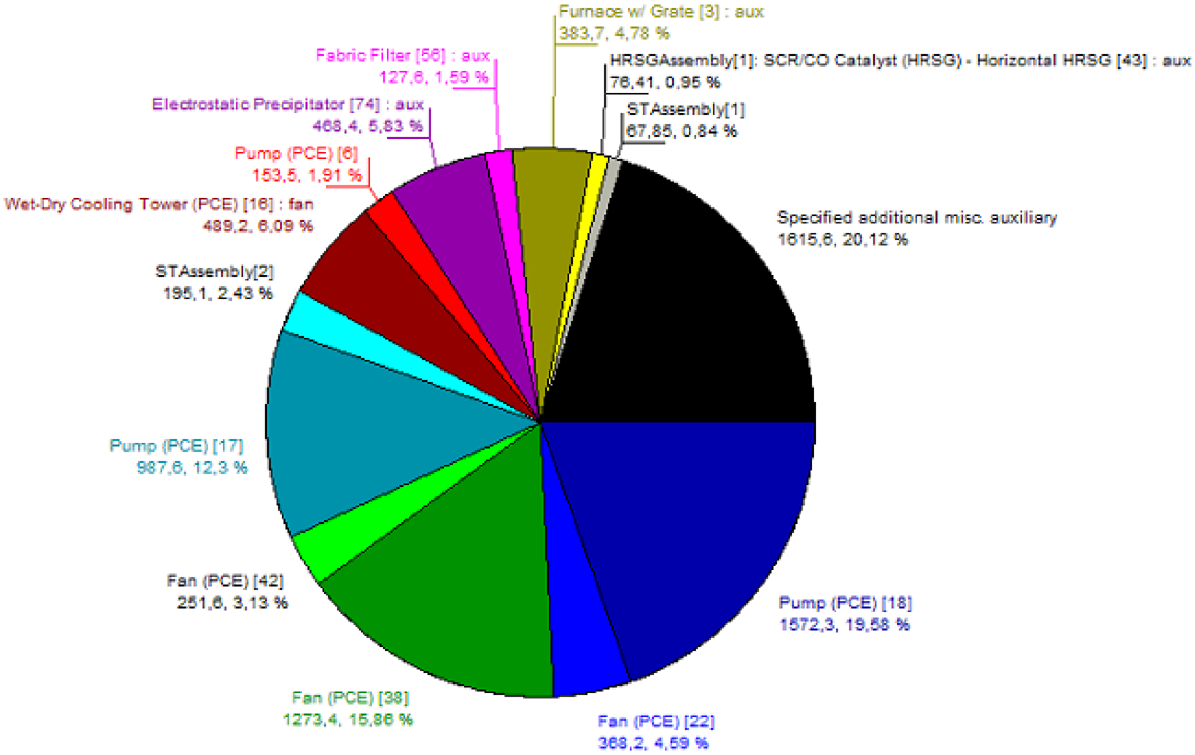


Figure 7. 15 - p_{EVA} = 150 bar, power absorbed by auxiliaries in percentage

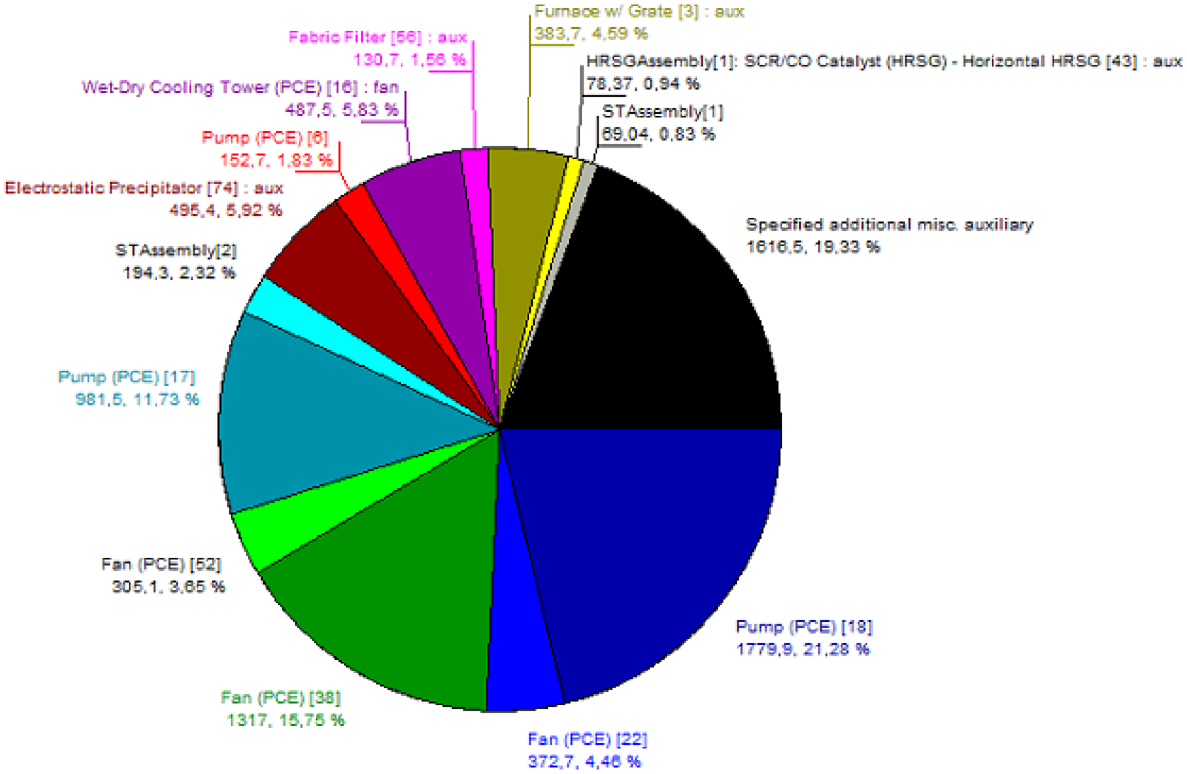


Figure 7. 16 - p_{EVA} = 170 bar, power absorbed by auxiliaries in percentage

The power absorbed by feed pump increases in accordance to evaporation pressure, since it has to increase saturated water pressure of a growing quantity of bar.

Chapter 8

Cost estimates

The present chapter aims to provide the main costs for all the components of the plant, grouping them on few sub-sets in order to give a clearer idea on the expenditure order of magnitude.

8.1.STEAM CYCLE

In this paragraph, the costs related to the steam cycle would be analysed.

8.1.1. Condenser

The condenser costs comprehend, other than the condenser itself, also the wet-dry cooling tower, necessary to cool down the saturated vapour that enters the condenser. In fact, when the plant is not built near a great water source capable of satisfying the required high quantity of water, such as a river or the see, an evaporative tower is needed. The plant is supposed to have the more general characteristics as possible, so the presence of the evaporative tower is taken into account. This device requires, obviously, a pump to guarantee the water movement, but its cost impact is furtherly analysed in the paragraph dedicated to auxiliaries' costs.

	Evaporation Pressure [bar]				
	70 (cfr plant)	**110**	**130**	**150**	**170**
Reference Material, Equipment, and Installation Costs					
Equipment					
Condenser cost [USD]	1'376'154	1'322'216	1'315'304	1'313'925	1'274'724
Mechanical					
Mechanical labor [h]	3'390	3'270	3'260	3'250	3'130
Mechanical labor cost [USD]	145'843	140'726	140'075	139'945	134'416
Civil					
Foundation Concrete Volume [m^3]	174.9	164.5	163.3	163.0	150.4
Foundation material & equipment cost [USD]	135'183	127'211	126'282	126'097	115'979
Excavation/backfill volume [m^3]	546.0	506.5	502.0	501.1	540.9
Civil Labor [h]	19'727	18'314	18'154	18'122	19'380
Civil Labor cost [USD]	4'749	4'462	4'428	4'422	4'131
Total civil cost [USD]	180'449	169'537	168'271	168'019	156'959
Cost Summary					
Total reference installed cost [USD]	1'857'355	1'778'004	1'768'086	1'766'108	1'701'458
Total estimated installed cost [USD]	**2'015'480**	**1'928'957**	**1'918'159**	**1'916'006**	**1'844'806**

Table 8. 1 - Condenser's costs

The costs related to the condenser comprehend, other than the main component cost, also the mechanical and civil labour costs as well as the foundation construction, materials and equipment. The **Table 8. 1** above, moreover, shows a difference between the total reference cost and total estimated cost. These two costs differ from the corrective coefficients showed in Chapter 6, related to, among others, operations and maintenance.

The lower the evaporation pressure, the higher the cost of the condenser. In fact, as explained before, a high evaporation pressure requires a higher temperature of flue gases exiting the boiler, so a lower quantity of vapour produced in the waterwall and finally a lower quantity of water entering the condenser. This would lead to lower diameters for condenser's tubes and lower overall condenser dimensions, which obviously reduces its costs. A minor quantity of saturated vapour requires a lower mass flux cooling fluid (water, again) within the condenser, so the evaporative tower would present higher costs for lower evaporation pressures.

On the other hand, the following **Table 8. 2** shows the synthesis data about condenser and wet dry cooling evaporative tower.

		Cost Breakdown	**Water-cooled Condenser**	**Wet-Dry Cooling Tower**	**Sum of costs**
Evaporation pressure [bar]	**70 (ref plant)**	**Est. Cost [USD]**	2'016'036	3'455'075	**5'471'111**
	110	**Est. Cost [USD]**	1'928'957	3'293'585	**5'222'542**
	130	**Est. Cost [USD]**	1'918'159	3'277'017	**5'195'176**
	150	**Est. Cost [USD]**	1'916'006	3'269'665	**5'185'671**
	170	**Est. Cost [USD]**	1'844'806	3'262'651	**5'107'457**

Table 8. 2 - Condenser and Evaporative Tower cost breakdown

As foresaw in the previous lines, even the wet-dry cooling tower costs increase with evaporation pressure decrease.

8.1.2. Regenerators

For the same reason explained for the condenser case, even the regenerators follows the same rule for costs variation as functions of quantity of water within the plant (and so the plant's evaporation pressure). In fact, the bleeding pressures are always the same, independently from the evaporation pressure, as well as the temperature and pressure of fluids entering and exiting these devices.

Therefore, the costs for regenerators increase as the evaporation pressure decreases. The only exception is for the plant with 110 bar and 130 bar as evaporation pressures, since this plants comprehend the presence of an additional high-pressure regenerator, which, of course, increment the overall regenerators' costs. Even in this case, though, the whole regenerators' costs follows the inverse proportionality law with the evaporation pressure.

		Cost Breakdown	**Regenerators**	**First low-pressure regenerator**	**Second low-pressure regenerator**	**High-pressure regenerator**

Evaporation Pressure [bar]	**70 (cfr plant)**	**Ref. Cost [USD]**	604'018	262'595	341'424	
		Est. Cost [USD]	644'039			
	110	**Ref. Cost [USD]**	833'270	2'415'190	317'738	274'013
		Est. Cost [USD]	888'372			
	130	**Ref. Cost [USD]**	737'599	243'267	286'117	208'215
		Est. Cost [USD]	786'107			
	150	**Ref. Cost [USD]**	528'603	242'909	285'694	
		Est. Cost [USD]	563'779			
	170	**Ref. Cost [USD]**	524'688	241'422	283'266	
		Est. Cost [USD]	559'621			

Table 8. 3 - Regenerators cost breakdown

8.1.3. Turbine

The turbine body comprehends the high and low pressure turbine. The high-pressure turbine resents on the presence of a high-pressure bleeding in the plant with 110 bar and 130 bar as evaporations pressures, so the cost of this two parts of the whole turbine body would increase due to this factor. On the other hand, the low-pressure body presents the same characteristics for all the innovative plant configurations: superheated vapour enters it with a temperature of 450°C and pressure of 25 bar and saturated vapour exits it with 0.07 bar as pressure and the correspondent evaporation temperature. The only difference among them is the vapour quantity dictated, again, by the evaporation pressure and the quantity of vapour spilled which can slight vary again due to the mass flux difference.

While this difference among low-pressure turbine bodies is present, it does not seem to affect the costs, which remains roughly the same at evaporation pressure variation.

The reference plant presents data only in the table related to high-pressure turbine because, again, there is only one turbine body, lacking it of the reheat.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Reference Material, Equipment, and Installation Costs					
Equipment					
Steam Turbine Package Cost [USD]	17'197'000	3'894'000	4'043'000	3'994'000	4'042'000
Mechanical					
Mechanical labor [h]	18'230	5'600	5'920	6'060	6'130
Mechanical labor cost [USD]	783'700	240'650	254'650	260'800	263'500

Transportation & Rigging					
On-site Transportation & Rigging [USD]	207'400	58'200	60'800	61'900	62'400
Civil (ST and Laydown Pads)					
Foundation concrete volume [m^3]	392.00	76.92	80.58	82.35	83.06
Foundation material & equipment cost [USD]	472'400	78'350	81'600	83'250	83'850
Excavation/backfill volume [m^3]	649	102	107	109	110
Excavation/backfill material and equipment cost [USD]	25'120	5'350	5'370	5'390	5'390
Civil labor [h]	14'800	2'210	2'300	2'350	2'370
Civil labor cost [USD]	562'500	84'150	87'550	89'250	89'950
Cost Summary					
Total Reference Installed Cost [USD]	19'248'000	4'361'000	4'533'000	4'495'000	4'548'000
Total Estimated Installed Cost [USD]	**20'479'000**	**4'644'000**	**4'828'200**	**4'789'000**	**4'846'000**

Table 8. 4 - High-pressure turbine's costs

	Evaporation Pressure [bar]			
	110	**130**	**150**	**170**
Reference Material, Equipment, and Installation Costs				
Equipment				
Steam Turbine Package Cost [USD]	12'246'000	12'189'000	12'179'000	12'150'000
Mechanical				
Mechanical labor [h]	16'690	16'700	16'690	16'650
Mechanical labor cost [USD]	720'900	718'000	717'500	716'000
Transportation & Rigging				
On-site Transportation & Rigging [USD]	184'700	183'700	183'500	183'000
Civil (ST and Laydown Pads)				
Foundation concrete volume [m^3]	343	341	341	340
Foundation material & equipment cost [USD]	414'200	411'650	411'200	409'950
Excavation/backfill volume [m^3]	571.00	567.00	567.00	565.00
Excavation/backfill material and equipment cost [USD]	22'110	21'990	21'960	21'900
Civil labor [h]	12'980	12'900	12'890	12'850
Civil labor cost [USD]	493'250	490'250	489'700	488'200
Cost Summary				
Total Reference Installed Cost [USD]	14'081'000	14'015'000	14'003'000	13'969'000
Total Estimated Installed Cost [USD]	**15'028'000**	**14'957'000**	**14'944'000**	**14'908'000**

Table 8. 5 - Low-pressure turbine's costs

The turbine package comprehends, other than the turbine itself, the generator cost as well as the gearbox, exhaust system and electrical/control/instrumentation package. They also take into account the lube oil package, main auxiliaries and emergency pumps, as well as the transportation to site.

Finally, the **Table 8. 6** below shows the breakdown costs for the whole turbine body. The reference plant turbine, even if lacks of the reheat and the expansion starts from a lower pressure, appears to present the higher cost.

		Cost Breakdown	High-pressure turbine body	Low-pressure turbine body	Sum of Costs
Evaporation pressure [bar]	**70 (ref plant)**	**Est. Cost [USD]**	20'479'000		**20'479'000**
	110	**Est. Cost [USD]**	4'644'000	15'028'000	**19'672'000**
	130	**Est. Cost [USD]**	4'828'200	14'957'000	**19'785'200**
	150	**Est. Cost [USD]**	4'789'000	14'944'000	**19'733'000**
	170	**Est. Cost [USD]**	4'846'000	14'908'000	**19'754'000**

Table 8. 6 - Turbine body's cost breakdown

Costs for turbine body comprehend the two assemblies, as well as the gearbox and the alternator. **Table 8. 6** seems to suggest that the costs derived from the presence of two turbine assemblies rather than one and the higher consequent turbine body complexion are more than compensated by the lower water mass flow rate present in the innovative plant. These observation seems slight optimistic at a further analysis, but more detailed study should be conducted in order to evaluate the reliability of data, maybe using a turbine's producers expert advice.

8.2. FLUE GASES CYCLE

The flue gases cycle comprehends the furnace, which costs have been calculated with the formula in Chapter 4 and the data in Chapter 6, the convective section, which comprehend the heat exchangers from the convective evaporator to the first economiser, the stack, the CO catalyst and SCR. Finally, the additional heat exchangers not included in the boiler (the two air pre-heaters as well as the condensate pre-heater) are part of the flue gases cycle as well.

The combustor and convective section costs depend on two factors, which have the opposite effect: the evaporation pressure and the water mass fllow.

With the evaporation pressure increase, in fact, there is the necessity to increase the tube thickness too, in order to sustain the major stresses created within the tubes and pipes materials. Therefore, with evaporation pressure increase, costs should increase consequently. However, the water/vapour mass flow reduces with evaporation pressure increase, so the heat exchanger can be designed for lower dimensions, reducing its cost and so the waterwall/convective section's overall costs.

These two parameters in antithesis would sign a minimum in waterwall and convective section costs, which is the optimum point between their combined values.

8.2.1. Waterwall

The **Table 8. 7** below shows the outcome of the calculation explained in Chapter 4 for the waterwall esteemed costs.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Reference Material, Equipment, and Installation Costs					
Equipment					
Waterwall esteemed cost [USD]	53'277'139.69	53'671'066.88	53'868'030.51	48'439'690.65	52'799'286.24
Civil					
Excavation and Backfill [USD]	447'410.24	364'351.94	322'823.50	299'552.88	289'268.09
Concrete [USD]	7'111'614.67	5'847'109.17	5'214'857.12	4'842'235.85	4'690'460.04
Mechanical					
Equipment erection and Assembly [USD]	22'836'565.84	20'642'122.21	19'544'900.47	16'601'372.63	16'433'449.94
Electrical assembly and Wiring					
Controls [USD]	1'919'047.55	1'734'316.44	1'641'950.68	1'394'398.80	1'380'077.73
Cost Summary					
Total esteemed installed cost [USD]	**85'591'778**	**82'258'967**	**80'592'562**	**71'577'251**	**75'592'542**

Table 8. 7 – Waterwall's costs

The optimum between the two parameters mentioned above shows a minimum in correspondence of 150 bar as evaporation pressure.

8.2.2. Convective section

As explained in Chapter 4, the Thermoflex costs calculations do not take the Inconel cladding of the first section of *SH2/RH3* heat exchangers into account, so the **Table 8. 8** below shows the values calculated with the method explained in Chapter 4. These costs are then added to the overall convective section costs.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Inconel cladded area SH2 [m^2]	1'749.0	1'944.3	1'941.0	1'944.8	2'010.0
Reference cost for Inconel cladding SH2 [USD]	5'908'821.6	6'568'623.1	6'557'474.4	6'570'312.3	6'790'584.0
Esteemed cost for Inconel cladding SH2 [USD]	6'204'262.7	6'897'054.3	6'885'348.1	6'898'827.9	7'130'113.2
Inconel cladded area RH3 [m^2]	-	1'526.20	1'185.30	1'244.00	1'239.40

Reference cost for Inconel cladding RH3 [USD]	-	5'156'114.08	4'004'417.52	4'202'729.60	4'187'188.96
Esteemed cost for Inconel cladding RH3 [USD]	-	5'413'919.78	4'204'638.40	4'412'866.08	4'396'548.41
Total additional reference cost for Inconel cladding [USD]	5'908'821.60	11'724'737.20	10'561'891.92	10'773'041.92	10'977'772.96
Total additional esteemed cost for Inconel cladding [USD]	**6'204'262.68**	**12'310'974.06**	**11'089'986.52**	**11'311'694.02**	**11'526'661.61**

Table 8. 8 - SH2/RH3 cladding costs

In the case of Inconel cladding, the reference plant registers the lower costs because of the lack of re-heat heat exchangers, so its cost is roughly half of the cost for heat exchangers cladding in any other innovative plant configuration.

The **Table 8. 9** below presents the principal costs value about the overall boiler as presented by Thermoflex, so without considering the cladding coverage as well as the waterwall.

	Evaporation Pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Equipment					
Overall Equipment cost [USD]	27'379'000	54'774'000	45'448'000	47'176'000	67'483'000
Main stack cost [USD]	2'572'000	2'572'000	2'572'000	2'572'000	2'587'000
SCR & Aqueous Ammonia System [USD]	760'700	760'000	760'600	760'100	771'700
CO Catalyst [USD]	422'650	422'250	422'550	422'250	428'700
Mechanical					
Mechanical labor [h]	131'300	256'500	186'150	191'500	253'950
Mechanical labor cost [USD]	5'646'000	11'158'000	8'005'000	8'235'000	10'920'000
Transportation & Rigging					
On-site transportation and rigging [USD]	2'663'000	4'740'000	3'583'000	3'670'000	4'656'000
Civil					
Foundation Concrete Volume [m^3]	5'600.00	10'530.00	8'470.00	10'250.00	27'240.00
Civil Labor [h]	124'150	233'450	187'550	227'050	604'400
Civil Labor cost [USD]	4'717'000	8'872'000	7'128'000	8'628'000	22'969'000
Total civil cost [USD]	9'006'000	16'934'000	13'603'000	16'465'000	43'834'000
Cost Summary					

Total reference installed cost [USD]	44'694'000	87'606'000	70'639'000	75'546'000	126'892'000
Total estimated installed cost [USD]	**49'001'000**	**95'992'000**	**77'198'000**	**82'696'000**	**140'015'000**

Table 8. 9 – Convective Section's costs

Even in this case, the lower costs are for the innovative plant with 130 bar and 150 bar as evaporation pressures.

Finally, the **Table 8. 10** below presents the recapitulatory costs for boiler and waterwall altogether (without the cladding costs).

		Cost Breakdown	**Convective Section**	**Waterwall**	**Sum of Costs**
Evaporation pressure [bar]	**70 (ref plant)**	**Est. Cost [USD]**	56'522'870	77'273'269	**133'796'139**
	110	**Est. Cost [USD]**	96'207'380	88'571'240	**184'778'620**
	130	**Est. Cost [USD]**	76'926'640	73'949'489	**150'876'129**
	150	**Est. Cost [USD]**	81'764'500	65'344'710	**147'109'210**
	170	**Est. Cost [USD]**	133'938'200	73'125'924	**207'064'124**

Table 8. 10 – Convective section and waterwall costs breakdown

As confirmed from the previous analysis about waterwall and convective section, plants with 130 bar and 150 bar as evaporation pressures show almost the same costs for convective section and waterwall, which are also the lower.

Convective section costs related to plant with 170 bar as evaporation pressure appear too high for only following the two antagonistic trends explained above. Unfortunately, all the dimensioning data acquired in Chapter 7 seem to vary correctly in respect to other data in innovative plants with lower evaporation pressures, so they do not appear to explain the anomalous costs. The industrial secrecy do not allow to recognise the real origin of this exaggerated cost increase from 150 bar to 170 bar as evaporation pressures, since the method of calculations result not visible. Therefore, the analysis of correct costs for this latter plant can be postponed for further investigations, maybe using the help of a software more specialised in costs estimates.

8.2.3. Additional heat exchangers

The additional heat exchangers comprehend the two air pre-heaters as well as the condensate pre-heater. The first two heat exchangers' costs do not vary at evaporation pressure increase, because air mass flow, temperature and pressure is fixed by the fuel's, which is fixed by data hypothesis and does not depend on evaporation pressure. As a result, vapour mass flow spilled from low-pressure turbine, its pressure and temperature required to heat the air up remains the same. On the other hand, the condensate pre-heater resents on the overall water mass flow as well as the quantity of flue gases destined to the discharge through the stack, which in turn depends on the quantity of gas destined to gas quenching.

	Cost Breakdown	**Sum of Costs**	**Second air pre-heater**	**First air pre-heater**	**Condensate pre-heater**

Evaporation Pressure [bar]	**70 (cfr plant)**	**Ref. Cost [USD]**	**880'693**	452'175	294'184	134'334
		Est. Cost [USD]	**924'727**			
	110	**Ref. Cost [USD]**	**893'324**	439'452	319'304	134'569
		Est. Cost [USD]	**937'991**			
	130	**Ref. Cost [USD]**	**893'391**	439'452	319'304	134'636
		Est. Cost [USD]	**938'061**			
	150	**Ref. Cost [USD]**	**893'309**	439'452	319'304	134'554
		Est. Cost [USD]	**937'975**			
	170	**Ref. Cost [USD]**	**896'053**	439'452	319'304	137'298
		Est. Cost [USD]	**940'856**			

Table 8. 11 - Additional heat exchangers' breakdown costs

8.3. AUXILIARIES

Pumps and fans constitutes the plant auxiliaries. Their costs always increase with evaporation pressure because with vapour pressure increases also the quantity of flue gases within the boiler (because flue gases recirculated for gas quench purposes increase) and the pressure to which water has to be raised to.

8.3.1. Pumps

Three pumps are present within the plant: the condensate pump, which raises the pressure of saturated water exiting the condenser to the deaerators' plus the eventual pressure drop within the two low-pressure regenerators (so it covers a few bar pressure gap). The feed pump raises the pressure of saturated water exiting the deaerator in order to increase it to the required evaporation pressure (it covers a hundred bars pressure gap) and finally the wet-dry tower pump, which guarantees the cooling fluid recirculation within the wet-dry cooling evaporative tower as well as the recirculation of condensate within the condenser.

Obviously, the feed pump is the one that operates the major pressure gap so it is the more expensive one.

		Cost Breakdown	**Pumps**	**Condensate pump**	**Feed pump**	**Wet-dry tower pump**
Evaporation Pressure [bar]	**70 (cfr plant)**	**Ref. Cost [USD]**	**841'982**	71'370	309'908	460'703
		Est. Cost [USD]	**897'016**			
	110	**Ref. Cost [USD]**	**805'248**	63'634	311'307	430'307
		Est. Cost [USD]	**857'265**			
	130	**Ref. Cost [USD]**	**830'805**	62'464	342'155	426'186
		Est. Cost [USD]	**884'251**			
	150	**Ref. Cost [USD]**	**1'010'323**	62'411	522'624	425'288
		Est. Cost [USD]	**1'075'445**			
	170	**Ref. Cost [USD]**	**1'066'415**	62'187	580'932	423'297
		Est. Cost [USD]	**841'982**	71'370	309'908	460'703

Table 8. 12 – Pumps' cost breakdown

Costs for pumps decreases as evaporation pressure increases because of the progressively lower quantity of water mass flow within the plant.

8.3.2. Fans

The three fans are all located after the splitter that imposes the flue gases flux values for gas quench, recirculation within the boiler and discharge through the stack.

Reference plant obviously lacks the gas quench fan, but the fan dedicated to FGR needs to be dimensioned for a higher flow rate, so it costs more.

In innovative configuration plants, gas quench fans cost more than FGR fan even if the flue gases flow rate is higher in the second than in the first one. The reason lays in the higher pressure gap covered by the gas quench fan. The stack fan, on the other hand, operates a small pressure gap but absorbs roughly half of flue gases flow rate exiting the convective section.

The **Table 8. 13** below shows the fans' cost breakdown.

		Cost Breakdown	Fans	Gas quench fan	FGR fan	Stack fan
Evaporation Pressure [bar]	**70 (cfr plant)**	**Ref. Cost [USD]**	**666'980**		150'411	516'569
		Est. Cost [USD]	**683'468**			
	110	**Ref. Cost [USD]**	**779'026**	155'992	93'998	529'036
		Est. Cost [USD]	**800'897**			
	130	**Ref. Cost [USD]**	**797'858**	155'836	115'347	526'676
		Est. Cost [USD]	**820'798**			
	150	**Ref. Cost [USD]**	**820'814**	156'201	138'755	525'857
		Est. Cost [USD]	**844'859**			
	170	**Ref. Cost [USD]**	**867'761**	166'033	158'469	543'259
		Est. Cost [USD]	**893'328**			

Table 8. 13 - Fans' cost breakdown

8.4. FLUE GAS TREATMENT SYSTEM

Apart from the SCR and CO catalyst, which are part of the convective section costs, fabric filter and electrostatic precipitator's costs are illustrated in the table below.

		Cost Breakdown	Electrostatic Precipitator	Fabric Filter	Sum of Costs
Evaporation pressure [bar]	**70 (ref plant)**	**Est. Cost [USD]**	10'812'010	7'035'832	**17'847'842**
	110	**Est. Cost [USD]**	12'421'140	7'035'737	**19'456'877**
	130	**Est. Cost [USD]**	12'816'100	7'043'676	**19'859'776**
	150	**Est. Cost [USD]**	13'175'760	7'035'547	**20'211'307**
	170	**Est. Cost [USD]**	13'763'430	7'177'491	**20'940'921**

Table 8. 14 - Flue gas treatment system's cost breakdown

As expected, flue gas treatment system components increases their size (and so their costs) with evaporation pressure increase because flue gases mass flow rate increases too.

8.5. GENERAL COSTS

The **Table 8. 14** below shows the overall plant cost breakdown, reassuming the esteemed costs for every plant considered in the present analysis, summing all the contributions.

		Cost Breakdown	Sum of Costs
Evaporation pressure [bar]	**70 (ref plant)**	**Est. Cost [USD]**	**159'442'717**
	110	**Est. Cost [USD]**	**211'328'492**
	130	**Est. Cost [USD]**	**178'876'986**
	150	**Est. Cost [USD]**	**178'707'135**
	170	**Est. Cost [USD]**	**227'525'761**

Table 8. 15 - Overall boiler's cost breakdown

The owner and contractor costs must be added in order to obtain the owner final cost for the plant, as calculated in the following **Table 8. 16**.

	Evaporation Pressure [bar]				
	70 (cfr plant)	**110**	**130**	**150**	**170**
Cost Summary	**Estimated Cost**				
Sum of Costs	188'208'445	238'844'184	217'589'369	214'571'690	276'899'009
Contractor's Soft & Miscellaneous Costs [USD]	27'677'618.19	38'083'335.71	33'101'233.24	33'274'275.32	48'406'976.07
Contractor's Price [USD]	215'886'063	276'927'519	250'690'602	247'845'965	325'305'985
Owner's Soft & Miscellaneous Costs [USD]	16'938'754.70	21'495'980.96	19'583'044.11	19'311'458.42	23'147'630.31
Total - Owner's Cost [USD]	**232'824'818**	**298'423'500**	**270'273'646**	**267'157'424**	**348'453'615**

Table 8. 16 - Overall plant's cost summary

The overall costs situation reflects the one obtained in the waterwall and convective section case (because they represent the more expensive components): plants with 130 bar and 150 bar as evaporation pressures present almost the same overall costs, which are also the lowest (not considering, of course, the reference plant). For reasons explained before, plant with 170 bar as evaporation pressure presents a cost too high to be considered, but evidences about performances and costs not related to the convective section would conduct to the conclusion that its electricity cost breakdown would not be the lowest and so it would not be the better combination anyway.

Finally, the last **Table 8. 17** presents the electricity breakdown cost, obtained with the procedure explained in Chapter 4.

	Evaporation pressure [bar]				
	70 (ref plant)	**110**	**130**	**150**	**170**
Power generated [kW]	58'119	62'868	63'518	63'529	63'286
Extra power generated with RH [MW]	-	4.749	5.399	5.410	5.167
Extra electriity generated in 1y (7800h) [GWh]	-	37.042	42.112	42.198	40.303
Capital Cost [MUSD]	232.825	298.424	270.274	267.157	348.454
Extra Capital Cost [MUSD]	-	65.599	37.449	34.333	115.629
Assumed capital carrying charge [%]	15	15	15	15	15
Assumed investment for O&M [%]	4	4	4	4	4
Extra costs [MUSD]	-	12.464	7.115	6.523	21.969
Break-even value of electricity [USD/MWh]	#DIV/0!	336.474	168.960	154.585	545.113
Power generated [kW]	58'119	62'868	63'518	63'529	63'286
Extra power generated with RH [MW]	-	4.749	5.399	5.410	5.167
Extra electriity generated in 1y (7800h) [GWh]	-	37.042	42.112	42.198	40.303
Capital Cost [MUSD]	232.825	298.424	270.274	267.157	348.454
Extra Capital Cost [MUSD]	-	65.599	37.449	34.333	115.629
Assumed capital carrying charge [%]	15	15	15	15	15
Assumed investment for O&M [%]	4	4	4	4	4
Extra costs [MUSD]	-	12.464	7.115	6.523	21.969
Break-even value of electricity [USD/MWh]	-	336.474	168.960	154.585	545.113

Table 8. 17 - Whole plant financial outcomes

The **Table 8. 17** above shows that the lower breakeven value of electricity is registered for the innovative plant with 150 bar as evaporation pressure, followed by the plant with 130 bar as evaporation pressure, as foresaw in the previous considerations.

Repeating the calculations for reference plant, obviously not using extra costs and extra power produced, but the overall power produced and the costs of the plant, the break-even value of electricity would result 97.58 USD/MWh, which is lower than the one obtained for every innovative plant.

Chapter 9

Conclusions and Future developments

Finally, the last chapter is dedicated to conclusions of analysis made so far and future development foresaw for the waste-to-energy innovative configuration plant.

9.1. CONCLUSIONS

As analysed in the previous chapter, the plant with 150 bar as evaporation pressure is the one with the lowest owner cost, excluding the reference plant, which presents a lower costs than all innovative configuration plants, but compensated with a much lower performances respect to the other ones.

The waste used as fuel presents great opportunities, but also great risks and challenges, most of all related to the hot corrosion problem. Every residential area produces municipal waste naturally and freely and its disposal is one of the great problems of our century. As a result, coupling the waste disposal problem with a method that not only eliminates waste from residential areas without damaging the environment (not more than a traditional power plant does), but also creates a positive outcome as the production of electric energy (and useful heat, in the most common cases) would seem the perfect solution.

The problem lays in the unfavourable characteristics of the waste fuel, which is rich in ashes and damaging substances as well as moisture, which decreases the maximum heat extractable from the combustion activity, since a part of it is used to vaporise the water. Moreover, waste also contains a relatively great amount of fixed carbon, which does not take part in the combustion process and increases the quantity of ashes in the furnace grate. Finally, its composition is always variable and the plants cannot be fully optimised. For all this reasons, a waste-to-energy plant requires higher quantity of fuel to produce the same electricity obtainable from a traditional gas-fired power plant. The great amount of waste mass flow requires big plants, which presents not-negligible costs.

As a result, waste-to-energy plants results to cost more than it can earn in terms of electricity and heat selling, even if the waste collection and delivery to the plant, instead of being a cost item (as it is in all the traditionally-fired power plants), is a service that the community pay to the plant’s owner. Their construction is currently possible only thanks to incentives, which promote the renewable power plants and the green energy politics.

Therefore, for grate combustors plants fed only with waste, as in the case of the present thesis ones, steam reheat seems an interest option, since it allows a more than 3 percentage point increase in net electric efficiency in the best case, in exchange to a more plant complexion. In fact, the introduction of reheat calls for a comprehensive redesign of the entire integrated boiler, since the gas quench requires the communication of opposite parts of the structure. Flue gas quench allows limiting the size and thus the cost of the radiative section (as is inferable from the **Table 8. 7** in Chapter 8), while keeping safe temperatures in the convective section, as extensively explained in Chapter 5.

Thermoflex estimates show that the plant with 150 bar as evaporation pressure obtains the highest performances and it is also the one that achieves the lowest owner's cost. As a result, the programme attests that the innovative configuration plant with 150 bar as evaporation pressure obtains the lowest brake-even electricity cost, even if the final value is higher than the one obtained with the reference plant. The logic conclusion for these outcomes is that waste-to-energy plants with reheat concede a great increase in performances, which, unfortunately, is overcome by the increase in plant complexion, which leads to a high cost increase. That creates a superior cost in extra energy produced thanks to the innovation, not sustainable given the current cost for electricity in traditional waste-to-energy plants.

9.2. FUTURE DEVELOPMENTS

The innovative configuration plant simulated in the present thesis adopts parameters of temperature and pressure at the limit of technological resistance in terms of materials and technology. If this choice guarantees the best possible outcome from the simulated plants, on the other hand it could put the overall plant correct functioning at risk in case of partial load.

For grid balance reasons, in fact, the plant could not always work with full load. This means that there is the possibility that the total water mass flow within the plant is not the one with which it has been designed, but registers a lower value.

Consequently, heat exchangers within the boiler appear to be over-dimensioned respect to the actual vapour/water flow, and the presence of less fluid within the tubes put the materials resistance at risk. The heat provided by flue gases, in fact, remains the same even in case of off-design functioning, but the less vapour quantity causes the major increase in tubes' material temperature. The parameters chosen for the plant in the present thesis are yet near the materials limit resistance, so an uncontrolled temperature raise in heat exchangers tubes could create dangerous consequences in terms of materials integrity and corrosion issues. A possible future study about the very same plant configuration could include the off-design possibility into account. The correct design of a power plant, in fact, should consider this precaution, adopting more conservative parameters.

The plant analysed in this thesis is only concentrated on the electric energy production, but the most common waste-to-energy plants foresee the cogeneration possibility. In other words, a part of the water mass flow is spilled from the turbine at a certain pressure and is used as hot fluid in a heat exchanger in order to warm up water for domestic uses. This procedure obviously reduces the quantity of electric energy produced, but permits to differentiate the power production, decreasing dimensions (and costs) of the steam cycle. Moreover, the flexibility obtained by the choice of water mass flow to be destined to thermal power production results useful for grid management. Therefore, another possibility to further investigating the same theme is to create a plant, which permits even the cogeneration procedure.

Since the reheat introduction shows its importance and convenience in terms of performances in the present thesis data analysis, another possible future development should concentrate on a different method to decouple the hot corrosion problem in first boiler’s heat exchangers to the efficiency increase in waste-to-energy plants. That could be perpetrated choosing an additional innovative configuration, which, hopefully, requires less costs and results to be more convenient from an economical point of view.

Finally, if a new configuration with reheat would result too onerous in terms of cost effectiveness, other possibilities among the ones listed in Chapter 3 should be taken into account in order to increase the overall plant efficiency.

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Appendix 1

The present appendix aims to provide the complete range of thermodynamic data provided by Thermoflex for the reference plant as well as the four configuration of innovative plant with reheat.

Reference plant – p_{EVA} = 70 bar

Stream	Fluid	P	T	M	H
		bar	C	kg/s	kJ/kg
1 - Outlet of Mixer [37] -> Feedwater inlet of Deaerator [1]	Water	5,956	149,04	69,96	628,06
2 - Feedwater outlet of Feedwater Heater (PCE) [2] -> Inlet 2 of Mixer [37]	Water	5,956	145	48,31	610,67
3 - Drain outlet of Feedwater Heater (PCE) [2] -> Water/steam addition to shell of Feedwater Heater (PCE) [45]	Water	4,51	95	4,8	398,24
4 - Outlet of Fuel Source [4] -> Fuel inlet of Furnace w/ Grate [3]	Fuel	1,0132	25	19,34	
5 - Saturated steam outlet of Furnace w/ Grate [3] -> Inlet 2 of Mixer [19]	Water	69,99	285,78	62,63	2773,29
6 - Feedwater outlet of Feedwater Heater (PCE) [45] -> Feedwater inlet of Feedwater Heater (PCE) [2]	Water	6,26	90	48,31	377,35
7 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [24] -> Inlet of Splitter [20]	Water	69,99	264,28	66,58	1155,81
8 - Outlet 1 of Splitter [8] -> Inlet of STAssembly[1]: ST Group [12]	Water	4,76	154,2	63,63	2754,87
9 - Outlet of Electrostatic Precipitator [39] -> Inlet of Splitter [28]	Gas/Air	1,0019	185,29	152	
10 - Outlet of STAssembly[1]: ST Group [12] -> Inlet of Splitter [9]	Water	4,51	149,58	63,69	2746,42
11 - Outlet 3 of Splitter [9] -> Heating steam inlet of Feedwater Heater (PCE) [2]	Water	4,51	149,58	4,8	2746,42
12 - Outlet of Mixer [19] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [30]	Water	69,99	285,78	65,96	2773,29
13 - Outlet of STAssembly[1]: ST Group [11] -> Inlet of Splitter [8]	Water	4,76	154,2	63,77	2754,87
14 - Outlet 1 of Splitter [9] -> Inlet of STAssembly[1]: ST Group [13]	Water	4,51	149,58	58,89	2746,42
15 - Outlet of STAssembly[1]: ST Group [13] -> Inlet of Splitter [49]	Water	0,79	93,17	58,89	2494,73
16 - Feedwater outlet of Deaerator [1] -> Suction of Pump (PCE) [18]	Water	5,956	149,98	70,09	632,11
17 - Air outlet of Coil (PCE) [23] -> Inlet of Splitter [25]	Gas/Air	1,075	165	113,4	
18 - Outlet 2 of Splitter [20] -> Feedwater of Furnace w/ Grate [3]	Water	69,99	264,28	63,25	1155,81
19 - Outlet of Mixer [29] -> Inlet of Makeup / Blowdown [31]	Water	0,1681	42,41	63,75	177,53
20 - CW outlet of Water-cooled Condenser (PCE) [15] -> Warm CW inlet of Wet-Dry Cooling Tower (PCE) [17]	Water	2,081	37,36	1578,2	156,58
21 - Discharge of Pump (PCE) [16] -> CW inlet of Water-cooled Condenser (PCE) [15]	Water	2,83	20,95	1578,2	88,08
22 - Air outlet of Coil (PCE) [14] -> Air inlet of Coil (PCE) [23]	Gas/Air	1,076	63,17	113,4	
23 - Outlet 3 of Splitter [32] -> Coil inlet of Coil (PCE) [23]	Water	14	257,36	5,552	2944,68
24 - Condensate outlet of Water-cooled Condenser (PCE) [15] -> Inlet 2 of Mixer [29]	Water	0,1681	39,02	52,57	163,37
25 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [30] -> Inlet 2 of Mixer [42]	Water	66,98	411,42	65,96	3196,53

26 - Outlet 3 of Splitter [8] -> Heating stream inlet of Deaerator [1]	Water	4,76	154,2	0,133	2754,87
27 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [21] -> Inlet of STAssembly[1]: ST Group [10]	Water	64,4	450	69,46	3296,96
28 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [30] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [24]	Gas/Air	1,0113	383,68	152	
29 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [21] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [30]	Gas/Air	1,0123	536,05	152	
30 - Flue gas of Furnace w/ Grate [3] -> Gas inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Gas/Air	1,0126	650	152	
31 - Outlet 3 of Splitter [25] -> Under grate air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	68,01	
32 - Outlet of Mixer [42] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [21]	Water	66,98	363,85	69,46	3067,74
33 - Outlet 2 of Splitter [28] -> Inlet of Fan (PCE) [22]	Gas/Air	1,0019	185,29	22,4	
34 - Outlet 1 of Splitter [32] -> Inlet of STAssembly[1]: ST Group [11]	Water	14	257,36	63,34	2944,68
35 - Outlet of Fan (PCE) [22] -> Inlet 2 of Gas/Air Mixer [33]	Gas/Air	1,066	198,43	22,4	
36 - Outlet 1 of Splitter [20] -> Water inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Water	69,99	264,28	3,33	1155,81
37 - Outlet 1 of Splitter [25] -> Over fire air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	45,34	
38 - Outlet of STAssembly[1]: ST Group [53] -> Inlet of Splitter [32]	Water	14	257,36	68,89	2944,68
39 - Outlet of Gas/Air Source [27] -> Air inlet of Coil (PCE) [14]	Gas/Air	1,076	15	113,4	
40 - Outlet 2 of Splitter [49] -> Coil inlet of Coil (PCE) [14]	Water	0,79	93,17	2,626	2494,73
42 - Coil outlet of Coil (PCE) [14] -> Inlet 3 of Mixer [29]	Water	0,1681	56,38	2,626	390,28
43 - Gas outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [21]	Gas/Air	1,0126	621,85	152	
44 - Steam outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet 3 of Mixer [19]	Water	69,99	285,78	3,33	2773,29
45 - Outlet of Makeup / Blowdown [31] -> Suction of Pump (PCE) [6]	Water	0,1681	42,14	64,4	176,38
46 - Outlet of Water Source [35] -> Inlet 1 of Gas/Air Mixer [33]	Water	1,066	105	0,204	2685,32
47 - Outlet of Gas/Air Mixer [33] -> Recirculation inlet of Furnace w/ Grate [3]	Gas/Air	1,066	196,92	22,61	
48 - Discharge of Pump (PCE) [18] -> Inlet of Splitter [41]	Water	75,59	151,68	70,09	643,72
49 - Coil outlet of Coil (PCE) [5] -> Inlet 1 of Mixer [37]	Water	5,956	145	16,1	610,67
50 - Outlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [34] -> Air inlet of Coil (PCE) [5]	Gas/Air	0,9733	175	138,8	
51 - Cool CW outlet of Wet-Dry Cooling Tower (PCE) [17] -> Suction of Pump (PCE) [16]	Water	1,0132	20,83	1578,2	87,44
53 - Outlet of Valve [47] -> Inlet 1 of Mixer [29]	Water	0,1681	47,62	8,56	199,29
54 - Drain outlet of Feedwater Heater (PCE) [45] -> Inlet of Valve [47]	Water	0,7849	47,61	8,56	199,29
55 - Outlet of Fabric Filter [36] -> Inlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [34]	Gas/Air	0,9798	175	138,8	
56 - Discharge of Pump (PCE) [6] -> Inlet of Splitter [40]	Water	6,541	42,61	64,4	178,9
57 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [24] -> Inlet of Electrostatic Precipitator [39]	Gas/Air	1,01	185,29	152	
58 - Air outlet of Coil (PCE) [5] -> Inlet of Fan (PCE) [38]	Gas/Air	0,973	130	138,8	
60 - Outlet 3 of Splitter [40] -> Coil inlet of Coil (PCE) [5]	Water	6,218	42,61	16,1	178,9
61 - Outlet of Mixer [51] -> Inlet 3 of Mixer [37]	Water	13,987	195	5,552	829,83
62 - Outlet of Fan (PCE) [38] -> Inlet of HRSGAssembly[1]: Concrete Stack [44]	Gas/Air	1,0122	137,48	138,8	
63 - Outlet 2 of Splitter [40] -> Feedwater inlet of Feedwater Heater (PCE) [45]	Water	6,541	42,61	48,31	178,9

64 - Outlet of Mixer [46] -> Steam inlet of Water-cooled Condenser (PCE) [15]	Water	0,07	39,03	52,57	2220,02
65 - Outlet 3 of Splitter [28] -> Inlet of Temperature Controller [52]	Gas/Air	1,0019	185,29	129,6	
66 - Outlet 3 of Splitter [41] -> Inlet 3 of Mixer [42]	Water	66,98	151,8	3,504	643,72
67 - Outlet 3 of Splitter [49] -> Heating steam inlet of Feedwater Heater (PCE) [45]	Water	0,79	93,17	3,76	2494,73
68 - Outlet 1 of Splitter [49] -> Inlet of STAssembly[1]: ST Group [48]	Water	0,79	93,17	52,5	2494,73
69 - Outlet of STAssembly[1]: ST Group [48] -> Inlet 2 of Mixer [46]	Water	0,07	39,03	52,57	2220,02
71 - Outlet 2 of Splitter [41] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [24]	Water	75,59	151,68	66,58	643,72
72 - Outlet of STAssembly[1]: ST Group [10] -> Inlet of Splitter [54]	Water	61	442,08	69,01	3282,64
74 - Outlet 1 of Splitter [54] -> Inlet of STAssembly[1]: ST Group [53]	Water	61	442,08	69,01	3282,64

Innovative plant – pEVA = 110 bar

Stream	**Fluid**	**P**	**T**	**M**	**H**
		bar	**C**	**kg/s**	**kJ/kg**
1 - Outlet of Mixer [37] -> Feedwater inlet of Deaerator [70]	Water	5,956	149,01	64,62	627,9
2 - Feedwater outlet of Feedwater Heater (PCE) [2] -> Inlet 2 of Mixer [37]	Water	5,956	145	41,36	610,67
3 - Drain outlet of Feedwater Heater (PCE) [2] -> Water/steam addition to shell of Feedwater Heater (PCE) [45]	Water	4,51	95	3,835	398,24
4 - Outlet of Fuel Source [4] -> Fuel inlet of Furnace w/ Grate [3]	Fuel	1,0132	25	19,34	
5 - Saturated steam outlet of Furnace w/ Grate [3] -> Inlet 2 of Mixer [19]	Water	110,65	318,49	54,79	2707,88
6 - Feedwater outlet of Feedwater Heater (PCE) [45] -> Feedwater inlet of Feedwater Heater (PCE) [2]	Water	6,216	90	41,36	377,34
8 - Outlet 1 of Splitter [8] -> Inlet of STAssembly[2]: ST Group [12]	Water	4,76	231,75	56,59	2923,59
9 - Outlet of Electrostatic Precipitator [41] -> Inlet of Splitter [28]	Gas/Air	1,0005	185,04	204,9	
10 - Outlet of STAssembly[2]: ST Group [12] -> Inlet of Splitter [9]	Water	4,51	227,07	56,93	2914,77
11 - Outlet 3 of Splitter [9] -> Heating steam inlet of Feedwater Heater (PCE) [2]	Water	4,51	227,07	3,835	2914,77
12 - Gas outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet of Splitter [30]	Gas/Air	1,0126	635,11	204,9	
13 - Outlet of STAssembly[2]: ST Group [11] -> Inlet of Splitter [59]	Water	14	365,24	61,13	3183,34
14 - Outlet 1 of Splitter [9] -> Inlet of STAssembly[2]: ST Group [13]	Water	4,51	227,07	50,32	2914,77
15 - Outlet of STAssembly[2]: ST Group [13] -> Inlet of Splitter [49]	Water	0,79	93,17	50,32	2611,87
16 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet of Electrostatic Precipitator [41]	Gas/Air	1,0086	185,04	204,9	
17 - Air outlet of Coil (PCE) [23] -> Inlet of Splitter [25]	Gas/Air	1,075	165	113,4	
18 - Outlet 2 of Splitter [20] -> Feedwater of Furnace w/ Grate [3]	Water	110,65	308,49	55,34	1391,22
19 - Outlet of Mixer [29] -> Inlet of Makeup / Blowdown [31]	Water	0,1681	43,38	57,03	181,59
20 - CW outlet of Water-cooled Condenser (PCE) [15] -> Warm CW inlet of Wet-Dry Cooling Tower (PCE) [16]	Water	2,031	37,36	1491,7	156,58
21 - Discharge of Pump (PCE) [17] -> CW inlet of Water-cooled Condenser (PCE) [15]	Water	2,78	20,94	1491,7	88,06

22 - Air outlet of Coil (PCE) [14] -> Air inlet of Coil (PCE) [23]	Gas/Air	1,076	74,81	113,4	
23 - Outlet 1 of Splitter [59] -> Inlet of STAssembly[2]: ST Group [58]	Water	14	365,24	56,71	3183,34
24 - Condensate outlet of Water-cooled Condenser (PCE) [15] -> Inlet 2 of Mixer [29]	Water	0,1681	39,02	47,4	163,37
25 - Outlet of Mixer [64] -> Inlet of Splitter [63]	Gas/Air	1,0121	520,06	204,9	
26 - Outlet of Valve [56] -> Inlet 1 of Mixer [72]	Water	66,4	157,01	2,608	666,1
27 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet 3 of Mixer [64]	Gas/Air	1,0122	520,02	85,05	
28 - Outlet of Mixer [66] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Gas/Air	1,0103	305,2	204,9	
29 - Feedwater outlet of Deaerator [70] -> Suction of Pump (PCE) [18]	Water	5,956	149,98	64,74	632,11
30 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 2 of Mixer [67]	Water	25,5	410,9	58,99	3263,86
31 - Outlet 3 of Splitter [25] -> Under grate air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	68,01	
32 - Discharge of Pump (PCE) [18] -> Inlet of Splitter [5]	Water	110,81	152,01	64,74	647,36
33 - Outlet 2 of Splitter [28] -> Inlet of Fan (PCE) [22]	Gas/Air	1,0005	185,04	22,4	
34 - Flue gas of Furnace w/ Grate [3] -> Inlet of Temperature Controller [39]	Gas/Air	1,0126	799,5	152	
35 - Outlet of Fan (PCE) [22] -> Inlet 2 of Gas/Air Mixer [33]	Gas/Air	1,066	198,47	22,4	
36 - Outlet 1 of Splitter [20] -> Water inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Water	110,65	308,49	2,92	1391,22
37 - Outlet 1 of Splitter [25] -> Over fire air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	45,34	
38 - Outlet of STAssembly[1]: ST Group [53] -> Inlet of Splitter [32]	Water	27,05	279,29	58,99	2950,86
39 - Outlet of Gas/Air Source [27] -> Air inlet of Coil (PCE) [14]	Gas/Air	1,076	15	113,4	
40 - Outlet 2 of Splitter [9] -> Coil inlet of Coil (PCE) [14]	Water	4,51	227,07	2,772	2914,77
41 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet of Splitter [20]	Water	110,65	308,49	58,26	1391,22
42 - Coil outlet of Coil (PCE) [14] -> Inlet 3 of Mixer [29]	Water	1,2	104,81	2,772	439,34
43 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 1 of Mixer [36]	Gas/Air	1,0112	369,9	134,2	
44 - Steam outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet 3 of Mixer [19]	Water	110,65	318,49	2,92	2707,88
45 - Outlet of Makeup / Blowdown [31] -> Suction of Pump (PCE) [6]	Water	0,1681	43,11	57,59	180,44
46 - Outlet of Water Source [35] -> Inlet 1 of Gas/Air Mixer [33]	Water	1,066	105	0,204	2685,32
47 - Outlet of Gas/Air Mixer [33] -> Recirculation inlet of Furnace w/ Grate [3]	Gas/Air	1,066	196,96	22,61	
48 - Outlet 3 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Gas/Air	1,0126	635,11	119,9	
49 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 3 of Mixer [36]	Gas/Air	1,0111	369,29	70,7	
50 - Outlet 1 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Gas/Air	1,0126	635,11	85,05	
51 - Outlet of Mixer [19] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Water	110,65	318,49	57,71	2707,88
52 - Outlet 3 of Splitter [8] -> Heating stream inlet of Deaerator [70]	Water	4,76	231,75	0,119	2923,59
53 - Outlet 1 of Splitter [28] -> Inlet of Fan (PCE) [42]	Gas/Air	1,0005	185,04	52,89	
54 - Outlet of Fan (PCE) [42] -> Control stream 1 of Temperature Controller [39]	Gas/Air	1,0126	187,6	52,89	
55 - Outlet of Temperature Controller [39] -> Gas inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Gas/Air	1,0126	650	204,9	
56 - Discharge of Pump (PCE) [6] -> Inlet of Splitter [40]	Water	6,456	43,57	57,59	182,93

57 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet 1 of Mixer [66]	Gas/Air	1,0105	305,47	127,1	
58 - Air outlet of Coil (PCE) [43] -> Inlet of Fan (PCE) [38]	Gas/Air	0,9717	130	138,6	
59 - Outlet of Mixer [52] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Water	110,65	279,29	58,26	1230,73
60 - Outlet of Fabric Filter [50] -> Inlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [71]	Gas/Air	0,9784	175	138,6	
61 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Inlet 3 of Mixer [66]	Gas/Air	1,0103	304,75	77,87	
62 - Outlet of Fan (PCE) [38] -> Inlet of HRSGAssembly[1]: Concrete Stack [44]	Gas/Air	1,0121	137,74	138,6	
63 - Outlet 2 of Splitter [40] -> Feedwater inlet of Feedwater Heater (PCE) [45]	Water	6,456	43,57	41,36	182,93
64 - Outlet of Mixer [46] -> Steam inlet of Water-cooled Condenser (PCE) [15]	Water	0,07	39,03	47,4	2319,76
65 - Drain outlet of Feedwater Heater (PCE) [45] -> Inlet of Valve [47]	Water	0,7849	48,57	6,863	203,33
66 - Outlet of Valve [47] -> Inlet 1 of Mixer [29]	Water	0,1681	48,59	6,863	203,33
67 - Outlet 3 of Splitter [49] -> Heating steam inlet of Feedwater Heater (PCE) [45]	Water	0,79	93,17	3,028	2611,87
68 - Outlet 1 of Splitter [49] -> Inlet of STAssembly[2]: ST Group [48]	Water	0,79	93,17	47,29	2611,87
69 - Outlet of STAssembly[2]: ST Group [48] -> Inlet 2 of Mixer [46]	Water	0,07	39,03	47,4	2319,76
70 - Outlet 2 of Splitter [5] -> Feedwater inlet of Feedwater Heater (PCE) [51]	Water	110,81	152,01	11,07	647,36
71 - Feedwater outlet of Feedwater Heater (PCE) [51] -> Inlet 2 of Mixer [52]	Water	110,65	279,24	11,07	1230,48
72 - Outlet of STAssembly[1]: ST Group [10] -> Inlet of Splitter [54]	Water	66,4	389,89	60,68	3141,15
73 - Outlet 3 of Splitter [54] -> Heating steam inlet of Feedwater Heater (PCE) [51]	Water	66,4	389,89	2,608	3141,15
74 - Outlet 1 of Splitter [54] -> Inlet of STAssembly[1]: ST Group [53]	Water	66,4	389,89	58,07	3141,15
76 - Drain outlet of Feedwater Heater (PCE) [51] -> Inlet of Valve [56]	Water	66,4	157,01	2,608	666,1
77 - Outlet 3 of Splitter [5] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Water	110,65	152,02	47,19	647,36
78 - Outlet of Mixer [36] -> Inlet of Splitter [65]	Gas/Air	1,0111	369,7	204,9	
79 - Outlet 3 of Splitter [40] -> Coil inlet of Coil (PCE) [43]	Water	6,221	43,58	16,23	182,93
80 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet 3 of Mixer [52]	Water	110,65	279,3	47,19	1230,79
81 - Outlet of Mixer [72] -> Inlet 3 of Mixer [37]	Water	13,987	181,32	7,029	769,08
82 - Outlet of STAssembly[2]: ST Group [58] -> Inlet of Splitter [8]	Water	4,76	231,75	56,71	2923,59
83 - Outlet 3 of Splitter [59] -> Coil inlet of Coil (PCE) [23]	Water	14	365,24	4,421	3183,34
84 - Outlet 1 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Gas/Air	1,0111	369,7	77,87	
85 - Outlet of Mixer [67] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Water	25,5	361,77	61,58	3153,84
86 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet of STAssembly[1]: ST Group [10]	Water	101,2	450	61,6	3241,54
87 - Outlet 3 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Gas/Air	1,0111	369,7	127,1	
88 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet of STAssembly[2]: ST Group [11]	Water	25	450	61,58	3351,05
89 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Water	27,05	279,29	58,99	2950,86
90 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet 1 of Mixer [64]	Gas/Air	1,0121	520,09	119,9	

91 - Outlet 1 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Gas/Air	1,0121	520,06	70,7	
92 - Outlet 3 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Gas/Air	1,0121	520,06	134,2	
93 - Cool CW outlet of Wet-Dry Cooling Tower (PCE) [16] -> Suction of Pump (PCE) [17]	Water	1,0132	20,83	1491,7	87,44
94 - Coil outlet of Coil (PCE) [43] -> Inlet 1 of Mixer [37]	Water	5,956	145	16,23	610,67
95 - Coil outlet of Coil (PCE) [23] -> Inlet 3 of Mixer [72]	Water	13,987	195	4,421	829,83
96 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Water	26,27	318,2	58,99	3050,79
97 - Outlet of Temperature Controller [1] -> Inlet of Fabric Filter [50]	Gas/Air	1,0005	175	138,6	
98 - Outlet of Mixer [68] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Water	102,45	361,78	61,6	2963,8
99 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 2 of Mixer [68]	Water	102,45	408,22	57,71	3119,7
100 - Outlet 1 of Splitter [5] -> Inlet of Splitter [69]	Water	102,45	152,14	6,474	647,36
101 - Outlet 1 of Splitter [69] -> Inlet 3 of Mixer [68]	Water	102,45	152,14	3,884	647,36
102 - Outlet 3 of Splitter [69] -> Inlet 1 of Mixer [67]	Water	25,5	153,24	2,589	647,36
103 - Outlet 3 of Splitter [28] -> Inlet of Temperature Controller [1]	Gas/Air	1,0005	185,04	129,6	
104 - Outlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [71] -> Air inlet of Coil (PCE) [43]	Gas/Air	0,9719	175	138,6	
105 - Outlet of Gas/Air Source [73] -> Control stream 2 of Temperature Controller [1]	Gas/Air	1,0005	15	9,004	

Innovative plant – p_{EVA} = 130 bar

Stream	**Fluid**	**P**	**T**	**M**	**H**
		bar	**C**	**kg/s**	**kJ/kg**
1 - Outlet of Mixer [37] -> Feedwater inlet of Deaerator [7]	Water	5,956	148,85	62,81	627,24
2 - Feedwater outlet of Feedwater Heater (PCE) [2] -> Inlet 2 of Mixer [37]	Water	5,956	145	40,85	610,67
3 - Drain outlet of Feedwater Heater (PCE) [2] -> Water/steam addition to shell of Feedwater Heater (PCE) [45]	Water	4,51	95	3,788	398,24
4 - Outlet of Fuel Source [4] -> Fuel inlet of Furnace w/ Grate [3]	Fuel	1,0132	25	19,34	
5 - Saturated steam outlet of Furnace w/ Grate [3] -> Inlet 2 of Mixer [19]	Water	129,6	330,59	53,28	2667,74
6 - Feedwater outlet of Feedwater Heater (PCE) [45] -> Feedwater inlet of Feedwater Heater (PCE) [2]	Water	6,218	90	40,85	377,34
7 - Feedwater outlet of Deaerator [7] -> Suction of Pump (PCE) [18]	Water	5,956	149,98	62,94	632,11
8 - Outlet 1 of Splitter [8] -> Inlet of STAssembly[2]: ST Group [12]	Water	4,76	231,78	56,12	2923,64
9 - Outlet 3 of Splitter [8] -> Heating stream inlet of Deaerator [7]	Water	4,76	231,78	0,133	2923,64
10 - Outlet of STAssembly[2]: ST Group [12] -> Inlet of Splitter [9]	Water	4,51	227,09	56,45	2914,83
11 - Outlet 3 of Splitter [9] -> Heating steam inlet of Feedwater Heater (PCE) [2]	Water	4,51	227,09	3,788	2914,83
12 - Gas outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet of Splitter [30]	Gas/Air	1,0126	637,64	219,2	
13 - Outlet of STAssembly[2]: ST Group [11] -> Inlet of Splitter [59]	Water	14	365,25	60,67	3183,36

14 - Outlet 1 of Splitter [9] -> Inlet of STAssembly[2]: ST Group [13]	Water	4,51	227,09	49,89	2914,83
15 - Outlet of STAssembly[2]: ST Group [13] -> Inlet of Splitter [49]	Water	0,79	93,17	49,89	2611,94
16 - Outlet 3 of Splitter [28] -> Inlet of Temperature Controller [41]	Gas/Air	1,0007	185,23	129,6	
17 - Air outlet of Coil (PCE) [23] -> Inlet of Splitter [25]	Gas/Air	1,075	165	113,4	
18 - Outlet 2 of Splitter [20] -> Feedwater of Furnace w/ Grate [3]	Water	129,6	320,41	53,82	1461,17
19 - Outlet of Mixer [29] -> Inlet of Makeup / Blowdown [31]	Water	0,1681	43,41	56,56	181,69
20 - CW outlet of Water-cooled Condenser (PCE) [15] -> Warm CW inlet of Wet-Dry Cooling Tower (PCE) [16]	Water	2,025	37,36	1479,4	156,58
21 - Discharge of Pump (PCE) [17] -> CW inlet of Water-cooled Condenser (PCE) [15]	Water	2,773	20,94	1479,4	88,06
22 - Air outlet of Coil (PCE) [14] -> Air inlet of Coil (PCE) [23]	Gas/Air	1,076	74,81	113,4	
23 - Outlet 1 of Splitter [59] -> Inlet of STAssembly[2]: ST Group [58]	Water	14	365,25	56,25	3183,36
24 - Condensate outlet of Water-cooled Condenser (PCE) [15] -> Inlet 2 of Mixer [29]	Water	0,1681	39,02	47,01	163,37
25 - Outlet of Mixer [64] -> Inlet of Splitter [63]	Gas/Air	1,0122	530,21	219,2	
26 - Outlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [1] -> Air inlet of Coil (PCE) [43]	Gas/Air	0,9722	175	138,8	
27 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet 3 of Mixer [64]	Gas/Air	1,0122	530,31	81,1	
28 - Outlet of Mixer [66] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Gas/Air	1,0103	285,63	219,2	
30 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 2 of Mixer [67]	Water	25,5	406,71	59,23	3254,58
31 - Outlet 3 of Splitter [25] -> Under grate air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	68,01	
32 - Discharge of Pump (PCE) [18] -> Inlet of Splitter [5]	Water	129,89	152,38	62,94	650,13
33 - Outlet 2 of Splitter [28] -> Inlet of Fan (PCE) [22]	Gas/Air	1,0007	185,23	22,41	
34 - Flue gas of Furnace w/ Grate [3] -> Inlet of Temperature Controller [39]	Gas/Air	1,0126	839	152	
35 - Outlet of Fan (PCE) [22] -> Inlet 2 of Gas/Air Mixer [33]	Gas/Air	1,066	198,61	22,41	
36 - Outlet 1 of Splitter [20] -> Water inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Water	129,6	320,41	2,83	1461,17
37 - Outlet 1 of Splitter [25] -> Over fire air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	45,34	
38 - Outlet of STAssembly[1]: ST Group [53] -> Inlet of Splitter [32]	Water	27,05	260,48	59,23	2899,6
39 - Outlet of Gas/Air Source [27] -> Air inlet of Coil (PCE) [14]	Gas/Air	1,076	15	113,4	
40 - Outlet 2 of Splitter [9] -> Coil inlet of Coil (PCE) [14]	Water	4,51	227,09	2,772	2914,83
41 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet of Splitter [20]	Water	129,6	320,41	56,65	1461,17
42 - Coil outlet of Coil (PCE) [14] -> Inlet 3 of Mixer [29]	Water	1,2	104,81	2,772	439,34
43 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 1 of Mixer [36]	Gas/Air	1,0115	400,59	152,1	
44 - Steam outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet 3 of Mixer [19]	Water	129,6	330,59	2,83	2667,74
45 - Outlet of Makeup / Blowdown [31] -> Suction of Pump (PCE) [6]	Water	0,1681	43,14	57,1	180,57
46 - Outlet of Water Source [35] -> Inlet 1 of Gas/Air Mixer [33]	Water	1,066	105	0,204	2685,32
47 - Outlet of Gas/Air Mixer [33] -> Recirculation inlet of Furnace w/ Grate [3]	Gas/Air	1,066	197,09	22,61	
48 - Outlet 3 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Gas/Air	1,0126	637,64	138,1	
49 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 3 of Mixer [36]	Gas/Air	1,0114	400,15	67,07	

50 - Outlet 1 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Gas/Air	1,0126	637,64	81,1	
51 - Outlet of Mixer [19] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Water	129,6	330,59	56,11	2667,74
52 - Coil outlet of Coil (PCE) [23] -> Inlet 1 of Mixer [71]	Water	13,987	195	4,421	829,83
53 - Outlet 1 of Splitter [28] -> Inlet of Fan (PCE) [42]	Gas/Air	1,0007	185,23	67,15	
54 - Outlet of Fan (PCE) [42] -> Control stream 1 of Temperature Controller [39]	Gas/Air	1,0126	187,74	67,15	
55 - Outlet of Temperature Controller [39] -> Gas inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Gas/Air	1,0126	650	219,2	
56 - Discharge of Pump (PCE) [6] -> Inlet of Splitter [40]	Water	6,414	43,6	57,1	183,04
57 - Outlet of Mixer [36] -> Inlet of Splitter [65]	Gas/Air	1,0114	400,47	219,2	
58 - Air outlet of Coil (PCE) [43] -> Inlet of Fan (PCE) [38]	Gas/Air	0,9719	130	138,8	
59 - Outlet of Mixer [52] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Water	129,6	260,57	56,65	1136,71
60 - Outlet of Fabric Filter [56] -> Inlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [1]	Gas/Air	0,9787	175	138,8	
61 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Inlet 3 of Mixer [66]	Gas/Air	1,0103	285,74	79,78	
62 - Outlet of Fan (PCE) [38] -> Inlet of HRSGAssembly[1]: Concrete Stack [44]	Gas/Air	1,0122	137,69	138,8	
63 - Outlet 2 of Splitter [40] -> Feedwater inlet of Feedwater Heater (PCE) [45]	Water	6,414	43,6	40,85	183,04
64 - Outlet of Mixer [46] -> Steam inlet of Water-cooled Condenser (PCE) [15]	Water	0,07	39,03	47,01	2319,78
65 - Drain outlet of Feedwater Heater (PCE) [45] -> Inlet of Valve [47]	Water	0,7849	48,6	6,777	203,45
66 - Outlet of Valve [47] -> Inlet 1 of Mixer [29]	Water	0,7849	48,6	6,777	203,45
67 - Outlet 3 of Splitter [49] -> Heating steam inlet of Feedwater Heater (PCE) [45]	Water	0,79	93,17	2,989	2611,94
68 - Outlet 1 of Splitter [49] -> Inlet of STAssembly[2]: ST Group [48]	Water	0,79	93,17	46,91	2611,94
69 - Outlet of STAssembly[2]: ST Group [48] -> Inlet 2 of Mixer [46]	Water	0,07	39,03	47,01	2319,78
70 - Outlet 2 of Splitter [5] -> Feedwater inlet of Feedwater Heater (PCE) [51]	Water	129,89	152,38	6,231	650,13
71 - Feedwater outlet of Feedwater Heater (PCE) [51] -> Inlet 2 of Mixer [52]	Water	129,6	260,28	6,231	1135,35
72 - Outlet of STAssembly[1]: ST Group [10] -> Inlet of Splitter [54]	Water	49,5	329,2	59,43	3015,07
73 - Outlet 3 of Splitter [54] -> Heating steam inlet of Feedwater Heater (PCE) [51]	Water	49,5	329,2	1,287	3015,07
74 - Outlet 1 of Splitter [54] -> Inlet of STAssembly[1]: ST Group [53]	Water	49,5	329,2	58,15	3015,07
76 - Drain outlet of Feedwater Heater (PCE) [51] -> Inlet 3 of Mixer [71]	Water	49,5	157,38	1,287	666,68
77 - Outlet 3 of Splitter [5] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Water	129,6	152,39	50,42	650,13
78 - Outlet 3 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Gas/Air	1,0114	400,47	139,4	
79 - Outlet 3 of Splitter [40] -> Coil inlet of Coil (PCE) [43]	Water	6,353	43,6	16,25	183,04
80 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet 3 of Mixer [52]	Water	129,6	260,6	50,42	1136,88
81 - Outlet of Electrostatic Precipitator [72] -> Inlet of Splitter [28]	Gas/Air	1,0007	185,23	219,2	
82 - Outlet of STAssembly[2]: ST Group [58] -> Inlet of Splitter [8]	Water	4,76	231,78	56,25	2923,64
83 - Outlet 3 of Splitter [59] -> Coil inlet of Coil (PCE) [23]	Water	14	365,25	4,421	3183,36
84 - Outlet 1 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Gas/Air	1,0114	400,47	79,78	

85 - Outlet of Mixer [67] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Water	25,5	370,74	61,12	3174,12
86 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet of STAssembly[1]: ST Group [10]	Water	119,6	450	60,52	3211,81
87 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet 1 of Mixer [66]	Gas/Air	1,0103	285,57	139,4	
88 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet of STAssembly[2]: ST Group [11]	Water	25	450	61,12	3351,05
89 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Water	27,05	260,48	59,23	2899,6
90 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet 1 of Mixer [64]	Gas/Air	1,0122	530,14	138,1	
91 - Outlet 1 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Gas/Air	1,0122	530,21	67,07	
92 - Outlet 3 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Gas/Air	1,0122	530,21	152,1	
93 - Cool CW outlet of Wet-Dry Cooling Tower (PCE) [16] -> Suction of Pump (PCE) [17]	Water	1,0132	20,83	1479,4	87,44
94 - Coil outlet of Coil (PCE) [43] -> Inlet 1 of Mixer [37]	Water	5,956	145	16,25	610,67
95 - Outlet of Gas/Air Source [50] -> Control stream 1 of Temperature Controller [41]	Gas/Air	1,0007	15	9,166	
96 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Water	26,27	330,5	59,23	3080,01
97 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet of Electrostatic Precipitator [72]	Gas/Air	1,0088	185,23	219,2	
98 - Outlet of Mixer [68] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Water	120	362,01	60,52	2907,06
99 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 2 of Mixer [68]	Water	120	408,8	56,11	3084,27
100 - Outlet 1 of Splitter [5] -> Inlet of Splitter [69]	Water	120	152,53	6,294	650,13
101 - Outlet 1 of Splitter [69] -> Inlet 3 of Mixer [68]	Water	120	152,53	4,406	650,13
102 - Outlet 3 of Splitter [69] -> Inlet 1 of Mixer [67]	Water	25,5	153,89	1,888	650,13
103 - Outlet of Mixer [71] -> Inlet 3 of Mixer [37]	Water	13,987	186,74	5,709	793,04
104 - Outlet of Temperature Controller [41] -> Inlet of Fabric Filter [56]	Gas/Air	1,0007	175	138,8	

Innovative plant – p_{EVA} = 150 bar

Stream	**Fluid**	**P**	**T**	**M**	**H**
		bar	**C**	**kg/s**	**kJ/kg**
1 - Feedwater outlet of Deaerator [1] -> Suction of Pump (PCE) [18]	Water	5,956	149,98	61,57	632,11
2 - Feedwater outlet of Feedwater Heater (PCE) [2] -> Inlet 2 of Mixer [37]	Water	5,956	145	40,77	610,67
3 - Drain outlet of Feedwater Heater (PCE) [2] -> Water/steam addition to shell of Feedwater Heater (PCE) [45]	Water	4,51	95	3,78	398,24
4 - Outlet of Fuel Source [4] -> Fuel inlet of Furnace w/ Grate [3]	Fuel	1,0132	25	19,34	
5 - Saturated steam outlet of Furnace w/ Grate [3] -> Inlet 2 of Mixer [19]	Water	149,54	341,89	52,12	2616,17
6 - Feedwater outlet of Feedwater Heater (PCE) [45] -> Feedwater inlet of Feedwater Heater (PCE) [2]	Water	6,219	90	40,77	377,34
7 - Outlet of Electrostatic Precipitator [74] -> Inlet of Splitter [28]	Gas/Air	1,0007	185,03	233,3	

8 - Outlet 1 of Splitter [8] -> Inlet of STAssembly[2]: ST Group [12]	Water	4,76	231,78	56,02	2923,64
9 - Outlet of Mixer [37] -> Feedwater inlet of Deaerator [1]	Water	5,956	148,67	61,42	626,45
10 - Outlet of STAssembly[2]: ST Group [12] -> Inlet of Splitter [9]	Water	4,51	227,1	56,36	2914,84
11 - Outlet 3 of Splitter [9] -> Heating steam inlet of Feedwater Heater (PCE) [2]	Water	4,51	227,1	3,78	2914,84
12 - Gas outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet of Splitter [30]	Gas/Air	1,0126	639,79	233,3	
13 - Outlet of STAssembly[2]: ST Group [11] -> Inlet of Splitter [59]	Water	14	365,25	60,6	3183,36
14 - Outlet 1 of Splitter [9] -> Inlet of STAssembly[2]: ST Group [13]	Water	4,51	227,1	49,81	2914,84
15 - Outlet of STAssembly[2]: ST Group [13] -> Inlet of Splitter [49]	Water	0,79	93,17	49,81	2611,95
16 - Outlet 3 of Splitter [28] -> Inlet of Temperature Controller [41]	Gas/Air	1,0007	185,03	129,6	
17 - Air outlet of Coil (PCE) [23] -> Inlet of Splitter [25]	Gas/Air	1,075	165	113,4	
18 - Outlet 2 of Splitter [20] -> Feedwater of Furnace w/ Grate [3]	Water	149,54	331,89	52,64	1532,53
19 - Outlet of Mixer [29] -> Inlet of Makeup / Blowdown [31]	Water	0,1681	43,41	56,47	181,72
20 - CW outlet of Water-cooled Condenser (PCE) [15] -> Warm CW inlet of Wet-Dry Cooling Tower (PCE) [16]	Water	2,023	37,36	1477	156,58
21 - Discharge of Pump (PCE) [17] -> CW inlet of Water-cooled Condenser (PCE) [15]	Water	2,771	20,94	1477	88,06
22 - Air outlet of Coil (PCE) [14] -> Air inlet of Coil (PCE) [23]	Gas/Air	1,076	74,81	113,4	
23 - Outlet 1 of Splitter [59] -> Inlet of STAssembly[2]: ST Group [58]	Water	14	365,25	56,18	3183,36
24 - Condensate outlet of Water-cooled Condenser (PCE) [15] -> Inlet 2 of Mixer [29]	Water	0,1681	39,02	46,93	163,37
25 - Outlet of Mixer [64] -> Inlet of Splitter [63]	Gas/Air	1,0122	538,25	233,3	
26 - Outlet of Temperature Controller [41] -> Inlet of Fabric Filter [56]	Gas/Air	1,0007	175	138,6	
27 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet 3 of Mixer [64]	Gas/Air	1,0122	538,28	85,16	
28 - Outlet 3 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Gas/Air	1,0116	419,3	153,5	
29 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet of Electrostatic Precipitator [74]	Gas/Air	1,0088	185,03	233,3	
30 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 2 of Mixer [67]	Water	25,5	406,22	59,2	3253,48
31 - Outlet 3 of Splitter [25] -> Under grate air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	68,01	
32 - Discharge of Pump (PCE) [18] -> Inlet of Splitter [5]	Water	149,54	153,48	61,57	656,05
33 - Outlet 2 of Splitter [28] -> Inlet of Fan (PCE) [22]	Gas/Air	1,0007	185,03	22,41	
34 - Flue gas of Furnace w/ Grate [3] -> Inlet of Temperature Controller [39]	Gas/Air	1,0126	878	152	
35 - Outlet of Fan (PCE) [22] -> Inlet 2 of Gas/Air Mixer [33]	Gas/Air	1,066	198,42	22,41	
36 - Outlet 1 of Splitter [20] -> Water inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Water	149,54	331,89	2,77	1532,53
37 - Outlet 1 of Splitter [25] -> Over fire air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	45,34	
38 - Outlet of STAssembly[1]: ST Group [53] -> Inlet of Splitter [32]	Water	27,05	245,17	59,2	2855,03
39 - Outlet of Gas/Air Source [27] -> Air inlet of Coil (PCE) [14]	Gas/Air	1,076	15	113,4	
40 - Outlet 2 of Splitter [9] -> Coil inlet of Coil (PCE) [14]	Water	4,51	227,1	2,772	2914,84
41 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet of Splitter [20]	Water	149,54	331,89	55,41	1532,53
42 - Coil outlet of Coil (PCE) [14] -> Inlet 3 of Mixer [29]	Water	1,2	104,81	2,772	439,34

43 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 1 of Mixer [36]	Gas/Air	1,0116	419,1	163,3	
44 - Steam outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet 3 of Mixer [19]	Water	149,54	341,89	2,77	2616,17
45 - Outlet of Makeup / Blowdown [31] -> Suction of Pump (PCE) [6]	Water	0,1681	43,15	57	180,62
46 - Outlet of Water Source [35] -> Inlet 1 of Gas/Air Mixer [33]	Water	1,066	105	0,204	2685,32
47 - Outlet of Gas/Air Mixer [33] -> Recirculation inlet of Furnace w/ Grate [3]	Gas/Air	1,066	196,92	22,61	
48 - Outlet 3 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Gas/Air	1,0126	639,79	148,2	
49 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 3 of Mixer [36]	Gas/Air	1,0116	419,76	70	
50 - Outlet 1 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Gas/Air	1,0126	639,79	85,16	
51 - Outlet of Mixer [19] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Water	149,54	341,89	54,89	2616,17
52 - Outlet 3 of Splitter [8] -> Heating stream inlet of Deaerator [1]	Water	4,76	231,78	0,152	2923,64
53 - Outlet 1 of Splitter [28] -> Inlet of Fan (PCE) [42]	Gas/Air	1,0007	185,03	81,28	
54 - Outlet of Fan (PCE) [42] -> Control stream 1 of Temperature Controller [39]	Gas/Air	1,0126	187,55	81,28	
55 - Outlet of Temperature Controller [39] -> Gas inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Gas/Air	1,0126	650	233,3	
56 - Discharge of Pump (PCE) [6] -> Inlet of Splitter [40]	Water	6,416	43,61	57	183,09
57 - Outlet 1 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Gas/Air	1,0116	419,3	79,8	
58 - Air outlet of Coil (PCE) [52] -> Inlet of Fan (PCE) [38]	Gas/Air	0,972	130	138,6	
59 - Outlet 2 of Splitter [66] -> Inlet 2 of Mixer [71]	Gas/Air	1,01	271,94	233,3	
60 - Outlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [43] -> Air inlet of Coil (PCE) [52]	Gas/Air	0,9722	175	138,6	
61 - Outlet of Mixer [36] -> Inlet of Splitter [65]	Gas/Air	1,0116	419,3	233,3	
62 - Outlet of Fan (PCE) [38] -> Inlet of HRSGAssembly[1]: Concrete Stack [44]	Gas/Air	1,0122	137,68	138,6	
63 - Outlet 2 of Splitter [40] -> Feedwater inlet of Feedwater Heater (PCE) [45]	Water	6,416	43,61	40,77	183,09
64 - Outlet of Mixer [46] -> Steam inlet of Water-cooled Condenser (PCE) [15]	Water	0,07	39,03	46,93	2319,78
65 - Drain outlet of Feedwater Heater (PCE) [45] -> Inlet of Valve [47]	Water	0,7849	48,61	6,762	203,49
66 - Outlet of Valve [47] -> Inlet 1 of Mixer [29]	Water	0,7849	48,61	6,762	203,49
67 - Outlet 3 of Splitter [49] -> Heating steam inlet of Feedwater Heater (PCE) [45]	Water	0,79	93,17	2,982	2611,95
68 - Outlet 1 of Splitter [49] -> Inlet of STAssembly[2]: ST Group [48]	Water	0,79	93,17	46,83	2611,95
69 - Outlet of STAssembly[2]: ST Group [48] -> Inlet 2 of Mixer [46]	Water	0,07	39,03	46,93	2319,78
71 - Outlet of Fabric Filter [56] -> Inlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [43]	Gas/Air	0,9786	175	138,6	
72 - Outlet of STAssembly[1]: ST Group [10] -> Inlet of Splitter [54]	Water	39,3	284,18	57,96	2917,68
74 - Outlet 1 of Splitter [54] -> Inlet of STAssembly[1]: ST Group [53]	Water	39,3	284,18	57,96	2917,68
76 - Outlet 3 of Splitter [5] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Water	149,54	153,48	55,41	656,05
78 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Inlet 3 of Mixer [70]	Gas/Air	1,01	272,09	79,8	
79 - Outlet 3 of Splitter [40] -> Coil inlet of Coil (PCE) [52]	Water	6,416	43,61	16,23	183,09
80 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Water	149,54	245,2	55,41	1063,47

81 - Coil outlet of Coil (PCE) [23] -> Inlet 3 of Mixer [37]	Water	13,987	195	4,421	829,83
82 - Outlet of STAssembly[2]: ST Group [58] -> Inlet of Splitter [8]	Water	4,76	231,78	56,18	2923,64
83 - Outlet 3 of Splitter [59] -> Coil inlet of Coil (PCE) [23]	Water	14	365,25	4,421	3183,36
84 - Outlet of Mixer [70] -> Inlet of Splitter [66]	Gas/Air	1,01	271,94	233,3	
85 - Outlet of Mixer [67] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Water	25,5	371,08	61,05	3174,9
86 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet of STAssembly[1]: ST Group [10]	Water	138	450	59,2	3180,65
87 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet 1 of Mixer [70]	Gas/Air	1,0102	271,85	153,5	
88 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet of STAssembly[2]: ST Group [11]	Water	25	450	61,05	3351,05
90 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet 1 of Mixer [64]	Gas/Air	1,0122	538,23	148,2	
91 - Outlet 1 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Gas/Air	1,0122	538,25	70	
92 - Outlet 3 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Gas/Air	1,0122	538,25	163,3	
93 - Cool CW outlet of Wet-Dry Cooling Tower (PCE) [16] -> Suction of Pump (PCE) [17]	Water	1,0132	20,83	1477	87,44
94 - Coil outlet of Coil (PCE) [52] -> Inlet 1 of Mixer [37]	Water	6,02	145	16,23	610,67
95 - Outlet of Gas/Air Source [50] -> Control stream 1 of Temperature Controller [41]	Gas/Air	1,0007	14,4	9,009	
96 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Water	26,27	333,4	59,2	3086,82
97 - Outlet of Mixer [71] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Gas/Air	1,01	271,95	233,3	
98 - Outlet of Mixer [68] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Water	138,46	367,27	59,2	2864,48
99 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 2 of Mixer [68]	Water	138,46	407,59	54,89	3037,88
100 - Outlet 1 of Splitter [5] -> Inlet of Splitter [69]	Water	138,46	153,64	6,157	656,05
101 - Outlet 1 of Splitter [69] -> Inlet 3 of Mixer [68]	Water	138,46	153,64	4,31	656,05
102 - Outlet 3 of Splitter [69] -> Inlet 1 of Mixer [67]	Water	25,5	155,26	1,847	656,05
108 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Water	27,05	245,17	59,2	2855,03
105 - Outlet 3 of Splitter [66] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [72] - ECO1.5	Gas/Air	1,0105	298,64	139,3	
106 - Outlet 1 of Splitter [66] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [73] - RH0.5	Gas/Air	1,0105	298,64	92,84	
108 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [73] - RH0.5	Water	27,32	246,26	58,85	2856,91

Innovative plant – p_{EVA} = 170 bar

Stream	**Fluid**	**P**	**T**	**M**	**H**
		bar	**C**	**kg/s**	**kJ/kg**
1 - Outlet of Mixer [37] -> Feedwater inlet of Deaerator [1]	Water	5,356	144,06	60,94	606,61
2 - Feedwater outlet of Feedwater Heater (PCE) [2] -> Inlet 2 of Mixer [37]	Water	5,197	139,99	39,49	589,13
3 - Drain outlet of Feedwater Heater (PCE) [2] -> Water/steam addition to shell of Feedwater Heater (PCE) [45]	Water	3,93	95	3,361	398,2

4 - Outlet of Fuel Source [4] -> Fuel inlet of Furnace w/ Grate [3]	Fuel	1,0135	25	19,34	
5 - Saturated steam outlet of Furnace w/ Grate [3] -> Inlet 2 of Mixer [19]	Water	169,37	351,96	51,05	2553,68
6 - Feedwater outlet of Feedwater Heater (PCE) [45] -> Feedwater inlet of Feedwater Heater (PCE) [2]	Water	5,451	90	39,49	377,29
7 - Feedwater outlet of Deaerator [1] -> Suction of Pump (PCE) [18]	Water	5,356	145,03	61,05	610,74
8 - Outlet 1 of Splitter [8] -> Inlet of STAssembly[2]: ST Group [12]	Water	4,16	217,16	55,56	2895,51
9 - Air outlet of Coil (PCE) [42] -> Inlet of Fan (PCE) [38]	Gas/Air	0,9715	130	138,2	
10 - Outlet of STAssembly[2]: ST Group [12] -> Inlet of Splitter [9]	Water	3,93	212,48	55,91	2886,77
11 - Outlet 3 of Splitter [9] -> Heating steam inlet of Feedwater Heater (PCE) [2]	Water	3,93	212,48	3,361	2886,77
12 - Gas outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet of Splitter [30]	Gas/Air	1,0126	641,65	247,1	
13 - Outlet of STAssembly[2]: ST Group [11] -> Inlet of Splitter [59]	Water	14	365,26	60,09	3183,38
14 - Outlet 1 of Splitter [9] -> Inlet of STAssembly[2]: ST Group [13]	Water	3,93	212,48	49,75	2886,77
15 - Outlet of STAssembly[2]: ST Group [13] -> Inlet of Splitter [49]	Water	0,79	93,17	49,75	2612,31
16 - Outlet 3 of Splitter [28] -> Inlet of Temperature Controller [41]	Gas/Air	1,0003	184,6	129,6	
17 - Air outlet of Coil (PCE) [23] -> Inlet of Splitter [25]	Gas/Air	1,075	165	113,4	
18 - Outlet 2 of Splitter [20] -> Feedwater of Furnace w/ Grate [3]	Water	169,37	341,74	51,56	1597,31
19 - Outlet of Mixer [29] -> Inlet of Makeup / Blowdown [31]	Water	0,1681	43,4	56	181,65
20 - CW outlet of Water-cooled Condenser (PCE) [15] -> Warm CW inlet of Wet-Dry Cooling Tower (PCE) [16]	Water	2,022	37,36	1476,6	156,58
21 - Discharge of Pump (PCE) [17] -> CW inlet of Water-cooled Condenser (PCE) [15]	Water	2,771	20,94	1476,6	88,06
22 - Air outlet of Coil (PCE) [14] -> Air inlet of Coil (PCE) [23]	Gas/Air	1,076	74,81	113,4	
23 - Outlet 1 of Splitter [59] -> Inlet of STAssembly[2]: ST Group [58]	Water	14	365,26	55,67	3183,38
24 - Condensate outlet of Water-cooled Condenser (PCE) [15] -> Inlet 2 of Mixer [29]	Water	0,1681	39,02	46,92	163,37
25 - Outlet of Mixer [64] -> Inlet of Splitter [63]	Gas/Air	1,0122	543,37	247,1	
26 - Coil outlet of Coil (PCE) [23] -> Inlet 3 of Mixer [37]	Water	13,987	195	4,421	829,83
27 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet 3 of Mixer [64]	Gas/Air	1,0122	543,27	87,73	
28 - Outlet 3 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Gas/Air	1,0117	429,8	162,4	
29 - Outlet 3 of Splitter [8] -> Heating stream inlet of Deaerator [1]	Water	4,16	217,16	0,11	2895,51
30 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 2 of Mixer [67]	Water	25,5	409,91	58,51	3261,68
31 - Outlet 3 of Splitter [25] -> Under grate air inlet of Furnace w/ Grate [3]	Gas/Air	1,0257	165	68,01	
32 - Discharge of Pump (PCE) [18] -> Inlet of Splitter [5]	Water	169,37	148,94	61,05	637,95
33 - Outlet 2 of Splitter [28] -> Inlet of Fan (PCE) [22]	Gas/Air	1,0003	184,6	22,41	
34 - Flue gas of Furnace w/ Grate [3] -> Inlet of Temperature Controller [39]	Gas/Air	1,0126	916	152	
35 - Outlet of Fan (PCE) [22] -> Inlet 2 of Gas/Air Mixer [33]	Gas/Air	1,066	198,05	22,41	
36 - Outlet 1 of Splitter [20] -> Water inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Water	169,37	341,74	2,72	1597,31
37 - Outlet 1 of Splitter [25] -> Over fire air inlet of Furnace w/ Grate [3]	Gas/Air	1,075	165	45,34	

38 - Outlet of STAssembly[1]: ST Group [53] -> Inlet of Splitter [32]	Water	27,05	232,04	58,51	2814,13
39 - Outlet of Gas/Air Source [27] -> Air inlet of Coil (PCE) [14]	Gas/Air	1,076	15	113,4	
40 - Outlet 2 of Splitter [9] -> Coil inlet of Coil (PCE) [14]	Water	3,93	212,48	2,804	2886,77
41 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet of Splitter [20]	Water	169,37	341,74	54,28	1597,31
42 - Coil outlet of Coil (PCE) [14] -> Inlet 3 of Mixer [29]	Water	1,2	104,81	2,804	439,34
43 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 1 of Mixer [36]	Gas/Air	1,0117	429,26	179,2	
44 - Steam outlet of HRSGAssembly[1]: Evaporator (PCE) [26] -> Inlet 3 of Mixer [19]	Water	169,37	351,96	2,72	2553,68
45 - Outlet of Makeup / Blowdown [31] -> Suction of Pump (PCE) [6]	Water	0,1681	43,14	56,52	180,56
46 - Outlet of Water Source [35] -> Inlet 1 of Gas/Air Mixer [33]	Water	1,066	105	0,204	2685,32
47 - Outlet of Gas/Air Mixer [33] -> Recirculation inlet of Furnace w/ Grate [3]	Gas/Air	1,066	196,54	22,61	
48 - Outlet 3 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Gas/Air	1,0126	641,65	159,4	
49 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2 -> Inlet 3 of Mixer [36]	Gas/Air	1,0117	431,18	67,96	
50 - Outlet 1 of Splitter [30] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Gas/Air	1,0126	641,65	87,73	
51 - Outlet of Mixer [19] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Water	169,37	351,96	53,77	2553,68
52 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Water	27,05	232,04	58,51	2814,13
53 - Outlet 1 of Splitter [28] -> Inlet of Fan (PCE) [52]	Gas/Air	1,0003	184,6	95,08	
54 - Outlet of Fan (PCE) [52] -> Control stream 1 of Temperature Controller [39]	Gas/Air	1,0126	187,19	95,08	
55 - Outlet of Temperature Controller [39] -> Gas inlet of HRSGAssembly[1]: Evaporator (PCE) [26]	Gas/Air	1,0126	650	247,1	
56 - Discharge of Pump (PCE) [6] -> Inlet of Splitter [40]	Water	5,647	43,55	56,52	182,79
57 - Outlet 1 of Splitter [65] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1	Gas/Air	1,0117	429,8	84,76	
58 - Outlet 3 of Splitter [40] -> Coil inlet of Coil (PCE) [42]	Water	5,647	43,56	17,03	182,79
59 - Water outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2	Water	169,37	233,2	54,28	1008,2
60 - Outlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [43] -> Air inlet of Coil (PCE) [42]	Gas/Air	0,9718	175	138,2	
61 - Outlet of Mixer [36] -> Inlet of Splitter [65]	Gas/Air	1,0117	429,8	247,1	
62 - Outlet of Fan (PCE) [38] -> Inlet of HRSGAssembly[1]: Concrete Stack [44]	Gas/Air	1,0122	137,77	138,2	
63 - Outlet 2 of Splitter [40] -> Feedwater inlet of Feedwater Heater (PCE) [45]	Water	5,647	43,55	39,49	182,79
64 - Outlet of Mixer [46] -> Steam inlet of Water-cooled Condenser (PCE) [15]	Water	0,07	39,03	46,92	2319,92
65 - Drain outlet of Feedwater Heater (PCE) [45] -> Inlet of Valve [47]	Water	0,7849	48,55	6,277	203,25
66 - Outlet of Valve [47] -> Inlet 1 of Mixer [29]	Water	0,7849	48,55	6,277	203,25
67 - Outlet 3 of Splitter [49] -> Heating steam inlet of Feedwater Heater (PCE) [45]	Water	0,79	93,17	2,916	2612,31
68 - Outlet 1 of Splitter [49] -> Inlet of STAssembly[2]: ST Group [48]	Water	0,79	93,17	46,83	2612,31
69 - Outlet of STAssembly[2]: ST Group [48] -> Inlet 2 of Mixer [46]	Water	0,07	39,03	46,92	2319,92
70 - Outlet of Electrostatic Precipitator [74] -> Inlet of Splitter [28]	Gas/Air	1,0003	184,6	247,1	
71 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1 -> Inlet of Electrostatic Precipitator [74]	Gas/Air	1,0084	184,6	247,1	

72 - Outlet of STAssembly[1]: ST Group [10] -> Inlet of Splitter [54]	Water	65	328,98	57,1	2966,65
73 - Outlet of Fabric Filter [56] -> Inlet of HRSGAssembly[1]: SCR/CO Catalyst (HRSG) - Horizontal HRSG [43]	Gas/Air	0,9782	175	138,2	
74 - Outlet 1 of Splitter [54] -> Inlet of STAssembly[1]: ST Group [53]	Water	65	328,98	57,1	2966,65
76 - Outlet of Temperature Controller [41] -> Inlet of Fabric Filter [56]	Gas/Air	1,0003	175	138,2	
77 - Outlet 3 of Splitter [5] -> Water inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Water	169,37	148,94	54,28	637,95
78 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Inlet 3 of Mixer [70]	Gas/Air	1,0098	257,04	84,76	
82 - Outlet of STAssembly[2]: ST Group [58] -> Inlet of Splitter [8]	Water	4,16	217,16	55,67	2895,51
83 - Outlet 3 of Splitter [59] -> Coil inlet of Coil (PCE) [23]	Water	14	365,26	4,421	3183,38
84 - Outlet of Mixer [70] -> Inlet of Splitter [66]	Gas/Air	1,0095	257,8	247,1	
85 - Outlet of Mixer [67] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3	Water	25,5	370,5	60,54	3173,59
86 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet of STAssembly[1]: ST Group [10]	Water	156,4	450	58,51	3147,99
87 - Gas outlet of HRSGAssembly[1]: Economiser (PCE) [24] - ECO2 -> Inlet 1 of Mixer [70]	Gas/Air	1,0095	258,2	162,4	
88 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [62] - RH3 -> Inlet of STAssembly[2]: ST Group [11]	Water	25	450	60,54	3351,05
90 - Gas outlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2 -> Inlet 1 of Mixer [64]	Gas/Air	1,0123	543,42	159,4	
91 - Outlet 1 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Gas/Air	1,0122	543,37	67,96	
92 - Outlet 3 of Splitter [63] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1	Gas/Air	1,0122	543,37	179,2	
93 - Cool CW outlet of Wet-Dry Cooling Tower (PCE) [16] -> Suction of Pump (PCE) [17]	Water	1,0132	20,83	1476,6	87,44
94 - Coil outlet of Coil (PCE) [42] -> Inlet 1 of Mixer [37]	Water	5,356	140	17,03	589,18
95 - Outlet of Gas/Air Source [50] -> Control stream 1 of Temperature Controller [41]	Gas/Air	1,0003	14,4	8,609	
96 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [60] - RH1 -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [34] - RH2	Water	26,27	341,76	58,51	3106,32
97 - Outlet of Mixer [71] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [57] - ECO1	Gas/Air	1,0095	257,8	247,1	
98 - Outlet of Mixer [68] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) - Parallel Flow [61] - SH2	Water	156,82	371,71	58,51	2814,86
99 - Steam outlet of HRSGAssembly[1]: Superheater (PCE) [21] - SH1 -> Inlet 2 of Mixer [68]	Water	156,82	411,17	53,77	3006,88
100 - Outlet 1 of Splitter [5] -> Inlet of Splitter [69]	Water	156,82	149,13	6,775	637,95
101 - Outlet 1 of Splitter [69] -> Inlet 3 of Mixer [68]	Water	156,82	149,13	4,743	637,95
102 - Outlet 3 of Splitter [69] -> Inlet 1 of Mixer [67]	Water	25,5	151,06	2,033	637,95
103 - Outlet 2 of Splitter [66] -> Inlet 2 of Mixer [71]	Gas/Air	1,0095	257,8	247,1	
105 - Outlet 3 of Splitter [66] -> Gas inlet of HRSGAssembly[1]: Economiser (PCE) [72] - ECO1.5	Gas/Air	1,0101	287,02	143,3	
106 - Outlet 1 of Splitter [66] -> Gas inlet of HRSGAssembly[1]: Superheater (PCE) [73] - RH0.5	Gas/Air	1,0101	287,02	101,6	
108 - Outlet 2 of Splitter [32] -> Steam inlet of HRSGAssembly[1]: Superheater (PCE) [73] - RH0.5	Water	27,32	233,07	58,64	2815,86