Pontifícia Universidade Católica
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Maria Luisa Nerys de Moraes Carneiro

Energy, exergy, economic, environmental (4E) analysis of hybrid systems for electricity generation from municipal solid waste and natural gas

Tese de Doutorado

Thesis presented to the Programa de Pós–graduação em Engenharia Mecânica of PUC-Rio in partial fulfillment of the requirements for the degree of Doutor em Engenharia Mecânica.

Advisor: Prof. Marcos Sebastião de Paula Gomes

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To my father, who has inspired me to become an engineer.
To my grandparents, for their unconditional love and support.
To my mother, for giving me life and raising me as a fighter.
To my husband, who inspires and supports me every day.

Acknowledgments

There is a Brazilian popular saying that says “God helps those who wake up early”, meaning those who work hard succeed. Having this work accomplished after several nights of unable sleep, worry, dedication and overcoming is also a good translation to this saying. In Engineering there is no space for hushing, skipping steps or overlooking. In fact, this is also the receipe for perfection. Doctor of Science, the title actually justifies it quite well. After so much thought one becomes a master in the art of thinking. Scrutinizing every single detail every step of the way may seem exhausting. It has been a very long journey indeed. A long and lonely journey. Loneliness may come along with three dangerous feelings: dismotivation, depression and madness. To fight each one of those a couple of measures can be taken. For dismotivation, surround yourself with interesting capable people, novel environments and tools. For depression, surround yourself with family, friends and leasure events. For madness, develop good self-organization skills.

In the beginning, many have brought up questions, but I have never had a doubt. In the middle, many have raised up questions, and I have had several doubts. In the end, some have issued a couple of questions, but I have answered them all. In the future, hopefully, many will build up questions and I will probably seek to develop answers, because knowledge never ends. We scientists only stop studying once we stop thinking, and that is when we stop existing.

I thank my supervisor, Prof. Marcos Sebastião Gomes, for believing in me, letting me express my creativity and pushing my work to beyond my expectations. I also thank Prof. Stefano Consonni, from Politecnico di Milano, for his constructive criticism and attention; LEAP and MatER team for welcoming and supporting me in a wonderful experience full of learning and challenges. Finally, I thank CNPq - Conselho Nacional de Desenvolvimento Científico e Tecnológico, of the Ministry of Science, Technology, Innovation, Communication of Brazil, for finacing all this work.

Abstract

Carneiro, Maria Luisa Nerys de Moraes; Gomes, Marcos Sebastião de Paula (Advisor). **Energy, exergy, economic, environmental (4E) analysis of hybrid systems for electricity generation from municipal solid waste and natural gas**. Rio de Janeiro, 2019. 194p. Tese de Doutorado – Departamento de Engenharia Mecânica, Pontifícia Universidade Católica do Rio de Janeiro.

This thesis aims to foster the development of hybrid waste-energy technology and its application in countries with access to natural gas reserves, such as Brazil. The method consists of evaluating it through an integrated analysis of energy, exergy, economic and environmental indicators. The investigated system consists of a topping and a bottoming cycle integrated through a heat recovery boiler. Raw non-recyclable urban waste feeds the waste boiler while natural gas feeds a gas turbine. The goal is to propose a cycle with high efficiency able to generate electricity/treat waste within affordable costs, which is achieved with an optimized design. The cycle's performance is proportional to the share of natural gas thermal input, that is, the greater the amount of waste the lower the efficiency (waste has lower calorific value). Therefore, the challenge is to seek greater efficiency with higher percentage of waste, including also the cost aspect. In general, there are two scenarios in the world: developed and developing countries. The former has several non-hybrid plants, few natural gas reserves and face reduced waste production, causing underutilization of the existing systems. This thesis provides a complete analysis of potential alternatives for "re-potentiation" of underutilized plants, demonstrating how worse their performance is compared to the optimized cycle. On the other hand, developing countries present increasing waste generation, under exploited reserves of natural gas and zero waste-to-energy plants. Therefore, there is a great potential for building hybrid waste-to-energy systems, which this work demonstrates are a much more reasonable alternative than non-hybrid conventional waste-to-energy plants.

Keywords

Waste-to-energy; Incineration; Hybrid combined cycles; Exergy; Natural gas; Municipal solid waste.

Resumo

Carneiro, Maria Luisa Nerys de Moraes; Gomes, Marcos Sebastião de Paula. **Análise energética, exergética, econômica e ambiental (4E) de sistemas híbridos para geração de eletricidade a partir de resíduos sólidos urbanos e gás natural**. Rio de Janeiro, 2019. 194p. Tese de Doutorado – Departamento de Engenharia Mecânica, Pontifícia Universidade Católica do Rio de Janeiro.

Esta tese tem como objetivo encorajar o desenvolvimento da tecnologia *híbrida de lixo-energia* e sua aplicação em países com acesso a reservas de gás natural, tal como o Brasil. O método consiste em avaliá-la por meio de uma análise integrada de indicadores de desempenho energético, exergético, econômico e ambiental. A tecnologia inclui dois ciclos integrados: um ciclo superior e um inferior que interagem por meio de uma caldeira de recuperação. O lixo urbano não reciclável, in natura, alimenta o ciclo inferior na caldeira de lixo enquanto o gás natural alimenta o ciclo superior na turbina a gás. O objetivo da modelagem destes sistemas consiste em propor um layout de máxima eficiência aliado a um custo acessível de produção de eletricidade/ tratamento do lixo, o que é atingido com um design otimizado. O desempenho termodinâmico mais eficiente é proporcional ao percentual de gás natural, isto é, quanto maior a quantidade de lixo em relação ao gás, menos eficiente é a planta, uma vez que o lixo tem menor poder calorífico. Portanto, o desafio é buscar uma maior eficiência com maior percentual de lixo, aliando também o aspecto custo. De uma forma geral, dois cenários coexistem no mundo atual, o dos países desenvolvidos e o dos países em desenvolvimento. Os primeiros possuem inúmeras aplicações de usinas não híbridas, poucas reservas próprias de gás natural e enfrentam redução na produção de lixo, causando a sub-utilização dos sistemas. Esta tese traz uma análise completa de potenciais alternativas para "re-potenciamento" de plantas sub-utilizadas, demonstrando o quão pior é o seu desempenho em comparação com o ciclo otimizado. Nos países em desenvolvimento ocorre o inverso: há crescente geração de resíduos, reservas pouco exploradas de gás natural e nenhuma usina lixo-energia. Portanto, existe um grande potencial para implantação de usinas híbridas, as quais, conforme a tese demonstra, são uma alternativa muito mais razoável do que implantar usinas convencionais não híbridas.

Palavras-chave

Lixo-energia; Incineração; Ciclos combinados híbridos; Exergia; Gás natural; Resíduos sólidos municipais.

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List of Abreviations

3E – energy, economic, environmental assessment
4E – energy, exergy, economic, environmental assessment
APH – Air pre-heater
BOP – Balance of the plant
CAPEX – Capital expenditure
CHP – Combined heat and power
CV – Control volume
CP – Compressor
CPI – Consumer price index
DEA – Deaerator
ECO – Economizer
EE – Energy-ecologic efficiency or ecological efficiency
EPA – The U.S. Environmental Protection Agency
EU – European Union
EXC – Excel®
FGS – Flue gas cleaning system
FWH – Feed water heater
GT – Gas turbine
GTS – Gas turbine system (compressor + combustion chamber + gas turbine)
HRB – heat recovery boiler
HCC – Hybrid combined cycle
HHV – Higher heating value
HRSG – Heat recovery steam generator
HX – Heat exchanger
LHV – Lower heating value
LCOE – Levelized cost of electricity production
LCOW – Levelized cost of waste treatment
MSW – Municipal solid waste
NG – Natural gas
ST – Steam turbine
TFX – Thermoflex®
WTE – Waste-to-energy
WTE-GT – Waste-to-energy plant integrated to a gas turbine

1
Introduction

The proper management of municipal solid waste (MSW) is a worldwide issue and draws particular attention in developing countries where the majority of municipalities dispose their urban waste through the most inexpensive way. There are various technologies for recovering energy from waste. The most used are thermal (incineration, fast and slow pyrolysis, gasification, production of refuse derived fuel), biochemical (composting, anaerobic digestion) and chemical conversions (esterification and other processes to convert waste to biodiesel) [1]. Recycling and energy recovery have been used in developed countries through the so-called Waste-to-Energy (WTE) technology [2]. In developing countries, incineration is considered as the most reliable and economical when used for electricity generation through mass burning without pre-treatment of MSW [3]. Pyrolysis is still in research phase and is not feasible for commercial purpose at large scale [3]. The U.S. Environmental Protection Agency (EPA) named WTE technology as one of the cleanest sources of energy due to the steadily diminishing levels of dioxin, furan, mercury, and other volatile metal emissions over the last 25 years [4].

In WTE thermal treatments the calorific power of urban or agricultural waste is recovered and heat and/or electricity are produced in a power cycle. The generated energy can be distributed to industries and households. However, a critical disadvantage is that MSW has a low energy content and high moisture, i.e., relatively small lower heating value (LHV), especially in developing countries. In addition, MSW burn generates acid gases which demands careful combustion control in order to contain boiler corrosion and diminish maintenance costs. Other reasons for relatively low performances of WTE plants are mainly due to the combined effects of economic and technical constraints such as small size plants, simple cycle configuration and high stack loss with indirect or no air pre-heating [5]. As a consequence, WTE plants' average thermal efficiency is usually very low (22%) [6]. Compared to fossil fuel power plants, whose efficiencies are, for instance: oil 40%, natural gas 54% and coal 40% [7]. A solution would be to combine MSW with other fuel of higher LHV in such a way that the average thermal efficiency could be increased up to 30–40%. For large-scale plants, even small efficiency gains translate into measurable carbon dioxide (CO_2) reductions, thus making hybrid plants an attractive alternative [8].

Udomsri et al. [9] and other studies cited along this thesis show that hybrid WTE plants using combined power cycle (called hybrid combined cycle - HCC) fueled by natural gas (NG) at topping cycle and MSW at bottoming cycle are more efficient than a composite of individual single-fuel power plants performing at the same energy input ratio and same conditions. Higher efficiencies can be achieved if superheating is realized in external fossil-fired boilers [10]. They do not have the corrosion limitations of waste-fired boilers, as implemented in the hybrid plants of Bilbao/Spain, Mainz/German [11] and Vantaa/Finland. The Zabalgarbi facility, located in the Spanish city of Bilbao, has been designed with the idea of an integrated power plant, being the biggest in terms of the electricity generation using waste-to-energy/gas turbine (WTE-GT) technology [12].

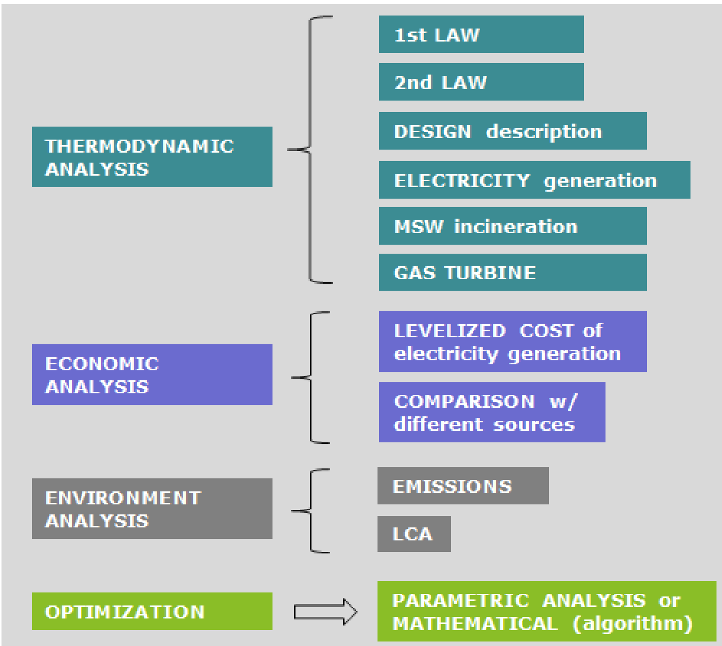


Figure 1.1: Scheme of the literature review developed to assure novelty.

Modeling of waste thermal treatment is a current theme that has been studied intensively around the world. A literature review has been made in order to guarantee that the main topics approached by this thesis have never been presented elsewhere. The main topics and subtopics included in the literature review are shown in fig. 1.1. As a result, no study has been found presenting all those elements simultaneously. In particular, only two works presented an analysis including energy/exergy/economic/environmental (4E) aspects and an optimization of some hybrid configuration involving biomass and natural gas, but none of them presented an optimization of the cycle itself nor the arrangement consisted of a combined hybrid cycle with a gas turbine in the topping cycle and a Rankine WTE in the bottoming cycle.

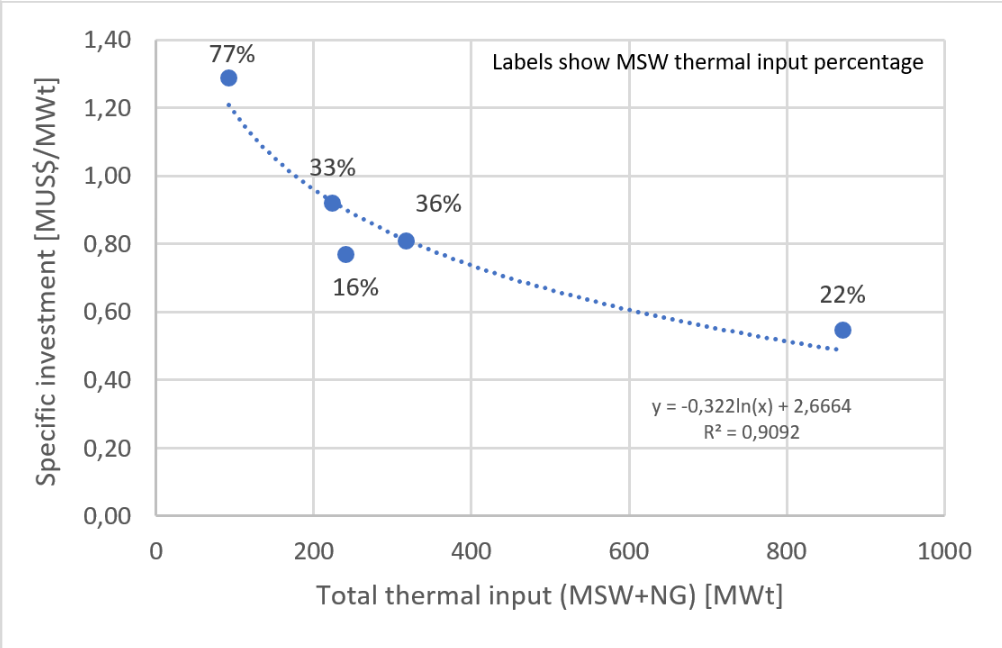


Figure 1.2: Specific costs of hybrid WTE-GT plants as a function of the total thermal input.

Key parameters of 22 existing and theoretical waste fired plants have been collected from the literature for those presenting cost estimate, including WTE and WTE-GT plants worldwide. In order to have a function with which one would be able to predict the costs of such plants the following graphs were built. Figure 1.2 shows the specific investment of WTE-GT plants as a function of the total fuel (MSW+NG) thermal input (in thermal megawatt - MW_t). Figure 1.3 shows the specific investment of WTE-GT plants as a function of the MSW thermal input percentage (MSW/total fuel). Figure 1.4 shows the main information regarding the hybrid plants shown in 1.2 and 1.3, where only Zabalgarbi is a real plant, all others are fictitious. Figure 1.5 shows the specific costs for single-fueled WTE plants as a function of the plant's MSW thermal input, where plants 2-12 are real and 14-22 are fictitious. Number 22 is the foreseen 1st WTE plant of Brazil, to be built in Barueri, São Paulo. The plant of Isseane (nbr. 8), in France, calls the attention for its extreme cost due to its extravagant layout and architecture. As observed, in general the costs of the single-fueled WTE plants are in the order of 2 million dollars per MW_t, whereas the hybrid WTE plants cost range is 0.5-1.3 MUS$/$MW_t$. For the hybrids, the cost is influenced by the ratio of MSW/NG, where the higher the

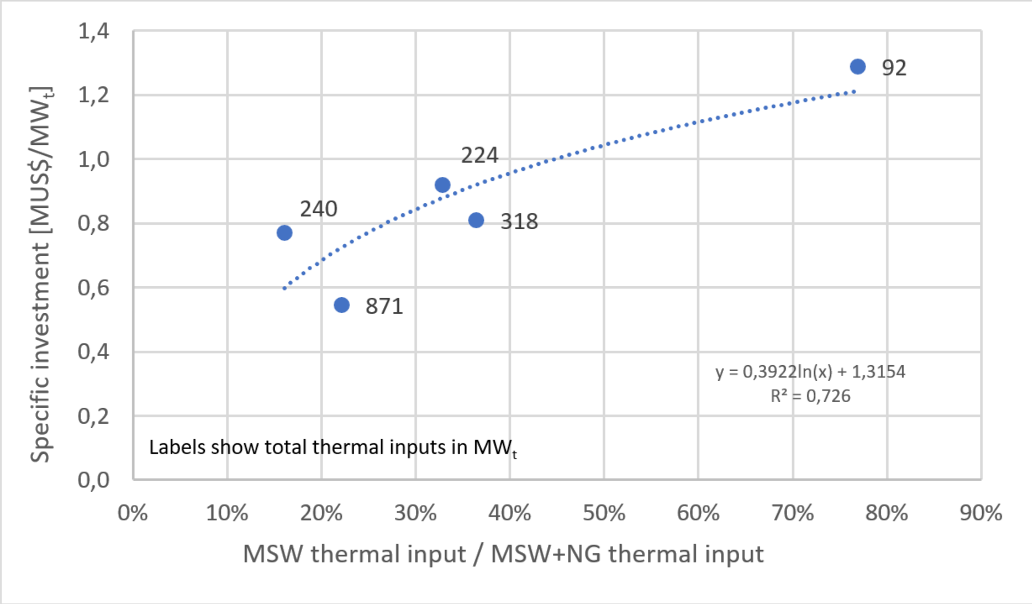


Figure 1.3: Specific costs of hybrid WTE-GT plants as a function of MSW thermal input percentage.

NG percentage the lower the specific costs, as observed in fig. 1.3.

The size of the plant is very important in the economic scale. As observed in figures 1.2 and 1.5, the higher the fuel's thermal input the lower the costs. According to [13], in general, the economy of operation and investment becomes better with larger plants and the cleaning of effluents can be made more efficient. The increment in size has to be compared with the cost for longer transport of waste. The grate-fired WtE plant in Amsterdam (v. fig. 1.5), which is the largest in Europe in terms of capacity and an example of an advanced design, where several improvements were made possible by the large size of the plant [13]. The overwhelming majority of waste conversion is carried out through incineration and the efficiency of combustion systems can be increased by several measures. Countries with warm climates, such as the case of Brazil, only produce electric power which causes the efficiency to be in the order of 25% or less, because corrosion on boiler tubes limits the steam temperature, and moreover, the small sizes of plants make the Rankine cycle inefficient [13]. Gasification using engines will not raise the efficiency considerably and for small sizes, a gasifier with engines is more efficient than a boiler with Rankine cycle, but for larger sizes the boiler system is slightly better [13]. Studies comparing conventional incineration with gasification such as [14, 15] conclude that, in terms of energy performance, they are very similar.

Name	Country	MSW capacity (t/y)	MSW thermal input (MW_t)	NG thermal input (MW_t)	Total thermal input (MWt)	MSW thermal input percentage	Total power output MW_e	Total investment ($)	
Zabalgarbi	Spain	289,780	74	150	224	33%	99.5	1.44E+08	Euro 2004
Consonni/00 WTE/CC small	Italy	100,000	39	202	240	16%	117.8	1.26E+08	US$ 1999
Consonni/00 WTE/CC medium	Italy	300,000	116	202	318	36%	145.8	1.74E+08	US$ 1999
Consonni/00 WTE/CC large	Italy	500,000	193	678	871	22%	445.2	3.22E+08	US$ 1999
Ribeiro & Sioen	Brazil	289,284	71	21	92	77%	32.0	1.03E+08	Euro 2013

Figure 1.4: Description of the WTE-GT plants shown in figs. 1.2 and 1.3, where Euro → Dollar conversion is 1.1 and all costs are shown in US$ of 2017 (converted using CPI index) [6, 16, 17].

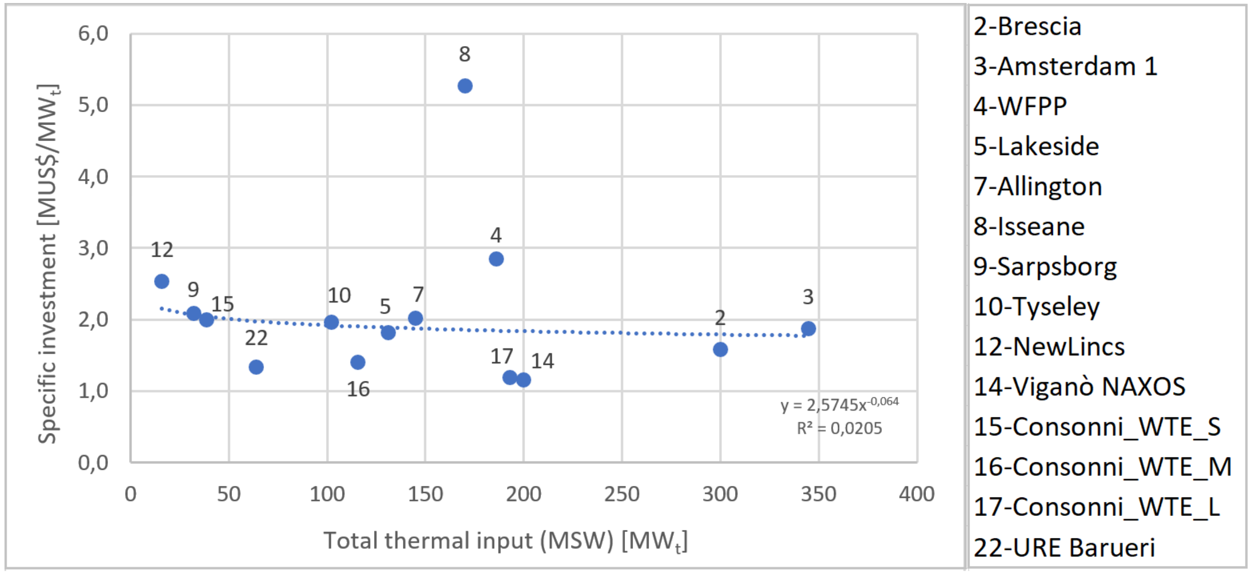


Figure 1.5: Specific costs of single-fueled WTE plants as a function of MSW (total) thermal input.

Another issue regards the choice of the waste combustor type. For large-sized plants, studies point grate-fired furnaces as the most appropriate choice for operating without plant sorting, which are now the dominant conversion device [13]. According to [13] more sophisticated systems, combining pyrolysers or gasifiers and combustors with the primary aim of melting part of the ashes and of reducing dioxin formation, were proposed in Germany and further developed and built in Japan. It appears that the present trend again is for simple (but well developed) grate combustion in combination with suitable ash treatment [13]. Some studies are dedicated to compare grate-fired and fluidized bed combustion, such as in [18, 19, 20, 13, 21]. Despite of the diverse conclusions, Leckner [13] points out the main reasons for applying grate-incinerators, stating that the grate is mechanically more complex than a fluidized bed but it has three advantages:

1. Most ashes leave the system as bottom ashes from the grate, whereas for the fluidized bed more ashes may leave the system as fly ashes.

2. Bottom ashes leave the grate at the end, after combustion, whereas in the case of the well-mixed fluidized bed, the bottom ashes leave together with bed material and some fuel.

3. Energy conversion in fluidized bed is likely to produce larger quantities of both bottom and fly ashes than in a grate-fired furnace.

1.1 Waste management scenario in Brazil

Brazil has a culture of landfills. Since very little environmental control is actually imposed to the existing “proper” landfills (not even leachate treatment is accomplished), such alternative is tremendously cheaper than any other waste management strategy. If adequate environmental practices were asserted to landfills and subsidies were given to WTE plants, the current situation could definitely be overcome. The Brazilian potential for incineration is largely unexplored. The National Bureau of Energy Research estimates that only 0.03% of the collected waste is treated at incineration facilities, the most part being hazardous waste, such as hospital residues. Up to the present time, there are no commercial-scale WTE plants fueled by non-hazardous MSW operating in the country [22]. Currently, there is a single project under development aiming to install a MSW WTE facility in the city of Barueri, in São Paulo State. The most populated cities, such as São Paulo and Rio de Janeiro, face the scarcity of available areas. Indeed, in 2018 São Paulo State government (the richest State of Brazil) is developing its local Waste Management Policy, where the best waste treatment route considering economic, environmental and social aspects has been investigated. The results of such multicriteria analysis point out that the best alternative is source separation followed by thermal treatment. In Brazil, most electricity is produced by hydroelectric plants, which are renewable and have low operating cost but are located far away from the big urban centers. Their main disadvantage is that during droughts their capacity is largely reduced and thermal power plants have to be activated to supply the energy demand. Besides that, the Brazilian production of natural gas has substantially increased recently due to exploitation of "Pre-salt" oil reservoirs. Indeed, estimates from the Energy Ministry predicts that Brazil tends to achieve gas self-sufficiency by 2021. Moreover, the government announces that Brazil might become an exporter soon and the benefits of building new non-flexible natural gas power plants are being investigated.

1.2
Review on waste to energy evaluation and applications

About 200 scientific publications were reviewed from international journal articles, conference proceedings and theses/dissertations from 2000 to 2018 about energy/exergy/economic/environmental analysis and/or optimization of Waste-to-Energy and/or natural gas systems. About 62% of the reviewed studies approached biomass incineration; from those 87% considered MSW, but only 26% accounted for hybrid WTE-GT technology. In particular, only two works included exergy aspects but did not present an exergy balance of the plant. In [23] the authors present exergy-economic equations applied to a steam cycle. In [24] the authors present the method to both single-fueled and hybrid plants. None of those addressed 4E assessments focused on MSW/NG-fired hybrid combined cycles including a comparison of the LCOE between different power sources. Hence, the present work aims at filling such gap. Summarized below are the main studies found in the literature with similar aspects as this work.

Bianchi et al. [25] investigate the integration of a conventional WTE power plant with a gas turbine (GT). The authors affirm that only three WTE–GT plants exist in European Union and Asia: Zabalgarbi (Spain), Moerdijk (the Netherlands), and Takahama (Japan). However, Udomsri [9] mentions the existence of four other plants in Europe: Gärstad CHP plant, built in 1983, in Linköping/Sweden and upgraded to hybrid-cycle in 1995; another one built in 1991 in Karlskoga/Sweden (GT not in operation today); the third one built in 1992 in Horsens/Denmark; and the fourth one built in 2003 in Mainz/Germany. Bianchi et al. [26] investigate the repowering of existing under-utilized WTE power plants with gas turbines. The results show that for one of the cases the steam turbine (ST) power output could be increased by 6.5 MW with efficiencies ranging between 32-36%.

Balcazar et al. [27] present a WTE system based on the integration of GT to a MSW incinerator in São José dos Campos, Brazil. The results show that incineration has fewer CO_2 emissions compared to landfilling, in absolute terms (t/y). In terms of specific emissions (t/MWh), the hybrid system is revealed to be more attractive than the stand-alone incineration plant. The study also calculates the levelized cost of electricity for the WTE-GT system in the Brazilian context but it does not detail the assumptions adopted for such LCOE calculation.

Ribeiro & Sioen [6] propose three configurations of WTE-GT called Optimized Combined Cycle (OCC). The authors affirm that OCC is an advantageous technology for its high efficiency and little consumption of NG compared to Zabalgarbi (Bilbao, Spain). Efficiency results are about 35%. An economic assessment calculates the internal rate of return, considering an electricity sale price of € 60-70/MWh and a capital expenditure of €103 million. The environmental emissions of OCC are estimated by Ribeiro & Kimberlin [28] as 34 ton CO_2/year.

Holanda & Balestieri [23] perform a 3E analysis of single-fueled WTE systems through Thermoeconomic Functional Analysis (TFA) method. Holanda & Balestieri [29] apply the method also to dual-fueled combined cycle which uses some exergy quantification in the economic analysis. Those works evaluate a single fueled plant and a hybrid combined cycle. The goal is to determine which atmospheric emission control system has fewer environmental and economic costs. The study considers the context of Guaratinguetá city, Brazil, and concludes that the incineration plant is only viable if the amount of waste generated in the city were 3-4 times greater and the electricity sale price were as high as US$ 40.00/MWh.

Consonni [16] performs a 3E analysis of WTE-GT compared to conventional WTE plants considering different scales. Consonni et al. [18],[19] perform a 3E analysis of four strategies of energy recovery from MSW in Italy. In [18] mass and energy balances are developed. In [19] estimates of emissions and costs are done. The strategies consist in a "mass burn" incineration with and without mechanical pre-treatment, comparing it to the combustion of pre-treated residues in a fluidized bed combustor. A Rankine steam cycle is used and scale effects are also evaluated. Results show that treating the waste ahead of the WTE plant reduces the energy recovery and does not provide environmental or economic benefits.

Villela & Silveira [30] evaluate the ecological efficiency of two types of power plants, NG and Diesel-fired systems, in the Brazilian context. The results show that NG performs better than Diesel, presenting an ecological efficiency (EE) value of 94% of the former against 91% for the latter, considering a thermal efficiency of 54% for the combined cycle. A similar method is used later by Coronado et al. [31] to quantify EE for biodiesel and/or Diesel combustion. The results show that for a plant fueled 100% by biodiesel (B100), EE is 98% and, whenever fueled 100% by Diesel EE decreases to 92%. Whereas, if a blend of 20% of biodiesel and 80% of Diesel (B20) is used, EE is 93%.

Rocco et al. [32] develop a joint application of an Exergy Analysis and a Life Cycle Assessment (ELCA) to evaluate and improve thermodynamic performances of an existent WTE power plant in the Italian context. The major exergy destructions are observed at the grate furnace, steam turbine and super-heater (96% of internal irreversibilities).

Therefore, no study has been found so far presenting energy, exergy, environmental and economic (4E) analysis of hybrid waste/gas power plants quantifying exergy destruction and environmental performance. Most of them apply energy/economic/environmental (3E) analysis (exclude exergy) and/or do not take into account the cost of emission pollution abatement system nor the levelized cost of electricity production (LCOE) and comparison with other energy conversion technologies. Moreover, very few studies focus on repowering of existing single-fueled waste-to-energy plants and none of them includes the environmental and economic aspects of the repowering options. From the above-mentioned figures, most facilities are small/medium-sized, and to the author's knowledge, there is no greenfield project of an advanced large-sized hybrid (waste/gas) power plant being investigated in the world considering all 4E aspects.

This dissertation contains 7 chapters presenting methodological and application approaches structured as follows:

Chapter 1 "Introduction" contains the motivation for the study, a compilation of relevant topics regarding energy conversion from waste and the technology' state-of-the-art.

Chapter 2 "Exergy assessment" is a methodological chapter describing the development of a tool in Excel® called 3E EXC to perform exergy assessment of complex thermopower systems modeled in Thermoflex® commercial software. The tool also supports simplified environmental and economic analyses.

Chapter 3 "4E analysis and feasibility of hybrid waste-to-energy plants" describes a methodological proposal for evaluating the feasibility of thermopower plants integrating energy-exergy-economic-environmental (4E) aspects in an original perspective. Besides of presenting the method, the chapter also brings an application to demonstrate it through a feasibility analysis of a waste/gas-fired plant → **includes a published article in a journal**.

Chapter 4 "Repowered waste-to-energy plants" brings an application of a 4E evaluation of brownfield projects for transforming single-fueled waste-to-energy plants into hybrid systems fueled by urban waste and natural gas → **includes a published poster/extended abstract in a congress**.

Chapter 5 "Energy-ecological efficiency" describes a methodological proposal and application for evaluating the environmental performance of thermopower plants → **includes a published article in a journal**.

Chapter 6 "Advanced hybrid waste-to-energy plants" brings applications of 4E analysis to high-performance hybrid waste-to-energy systems modeled in Thermoflex® and presents an optimal layout based on the investigation of energy, exergy, economic and environmental indicators.

Chapters 3-6 present both the method description and applications for which they have been structured in the form of article, i.e. Abstract; Introduction; Method; Results and discussion; Conclusive remarks and future work.

The last chapter "Conclusion" brings the major findings of the overall study and the contribution of this research to the scientific community and the general public.

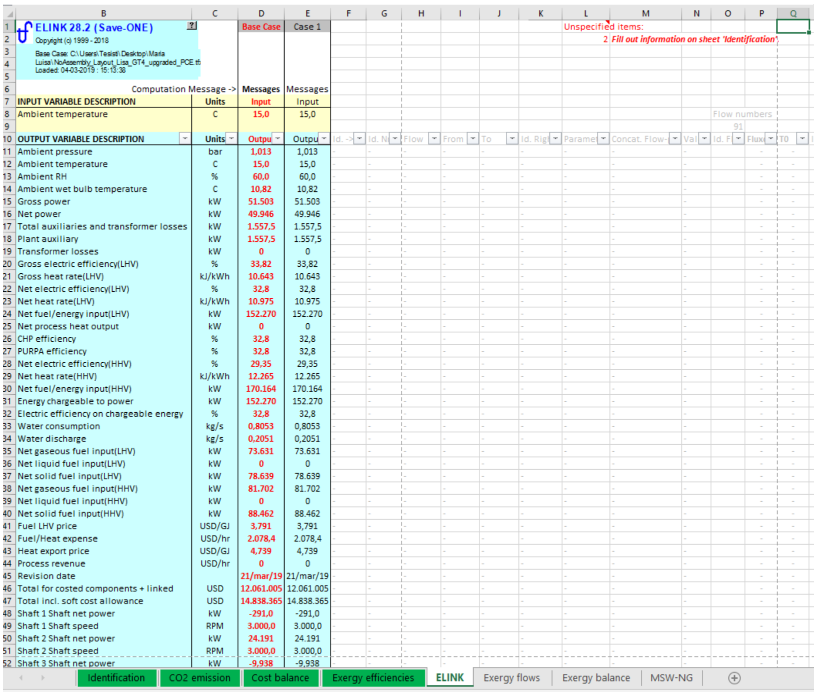
2
Exergy assessment

Exergy analysis is well suited for furthering the goal of more effective energy resource use, for it enables the location, cause, and true magnitude of waste and loss to be determined [33]. Such information can be used in the design of new energy-efficient systems and for increasing the efficiency of existing systems, also provides insights that elude a purely first-law approach [33]. Performing exergy balances of thermal systems might be a simple task when commercial software are available. Software like Thermoflex® sometimes present advanced versions that can, in addition to calculate mass and energy balances, also perform automatic calculation of exergy destruction in complex energy systems. Although Thermoflex® (TFX) has been used for the development of part of this thesis, the version available did not allow such straightforward procedure. In this sense, an additional methodology was required to allow automated exergy assessments of complex hybrid cycles modeled in TFX.

In summary, this is a methodological chapter aiming to describe the procedure proposed to create a code in Excel® (EXC) to read the TFX outputs and perform the exergy calculations of the systems studied in this thesis. The layout of the developed EXC tool is shown in fig. 2.1 and the content of each sheet is shown in fig. 2.2. The tool has been built to operate as automatically as possible after the TFX outputs are pasted on the "E-link sheet" (v. fig. 2.2), but some intervention from the user is still required (v. green sheets in fig. 2.2). It can also be observed from fig. 2.2 that, in addition to the exergy parameters, the tool also calculates very simplified balances of carbon dioxide and cost flows. The CO_2 indicators are based on the own TFX outputs, while the cost balance is based on basic financial incomes and charges informed by the user (it does not include capital amortization).

It is important to highlight that Thermoflex® versions 28-1 and 28-2 were considered to the development of the tool. Thus, it is not guaranteed that it will work with other versions because the TFX outputs' standard might change. In addition, the model built in TFX should accomplish the following restrictions in order to be compatible with the tool:

- The components cannot be labeled or named;
- There should be no assemblies in place;



	B	C	D	E
1	ELINK 28.2 (Save-ONE)		Base Case	Case 1
2	Copyright (c) 1999 - 2018			
3	Base Case: C:\Users\Tesisti\Desktop\Maria			
4	Luisa\NoAssembly_Layout_Lisa_GT4_upgraded_PCE.tfx			
5	Loaded: 04-03-2019 : 15:13:38			
6	Computation Message ->		Messages	Messages
7	INPUT VARIABLE DESCRIPTION	Units	Input	Input
8	Ambient temperature	C	15,0	15,0
9				
10	OUTPUT VARIABLE DESCRIPTION	Units	Outpu	Outpu
11	Ambient pressure	bar	1,013	1,013
12	Ambient temperature	C	15,0	15,0
13	Ambient RH	%	60,0	60,0
14	Ambient wet bulb temperature	C	10,82	10,82
15	Gross power	kW	51.503	51.503
16	Net power	kW	49.946	49.946
17	Total auxiliaries and transformer losses	kW	1.557,5	1.557,5
18	Plant auxiliary	kW	1.557,5	1.557,5
19	Transformer losses	kW	0	0
20	Gross electric efficiency(LHV)	%	33,82	33,82
21	Gross heat rate(LHV)	kJ/kWh	10.643	10.643
22	Net electric efficiency(LHV)	%	32,8	32,8
23	Net heat rate(LHV)	kJ/kWh	10.975	10.975
24	Net fuel/energy input(LHV)	kW	152.270	152.270
25	Net process heat output	kW	0	0
26	CHP efficiency	%	32,8	32,8
27	PURPA efficiency	%	32,8	32,8
28	Net electric efficiency(HHV)	%	29,35	29,35
29	Net heat rate(HHV)	kJ/kWh	12.265	12.265
30	Net fuel/energy input(HHV)	kW	170.164	170.164
31	Energy chargeable to power	kW	152.270	152.270
32	Electric efficiency on chargeable energy	%	32,8	32,8
33	Water consumption	kg/s	0,8053	0,8053
34	Water discharge	kg/s	0,2051	0,2051
35	Net gaseous fuel input(LHV)	kW	73.631	73.631
36	Net liquid fuel input(LHV)	kW	0	0
37	Net solid fuel input(LHV)	kW	78.639	78.639
38	Net gaseous fuel input(HHV)	kW	81.702	81.702
39	Net liquid fuel input(HHV)	kW	0	0
40	Net solid fuel input(HHV)	kW	88.462	88.462
41	Fuel LHV price	USD/GJ	3,791	3,791
42	Fuel/Heat expense	USD/hr	2.078,4	2.078,4
43	Heat export price	USD/GJ	4,739	4,739
44	Process revenue	USD/hr	0	0
45	Revision date		21/mar/19	21/mar/19
46	Total for costed components + linked	USD	12.061.005	12.061.005
47	Total incl. soft cost allowance	USD	14.838.365	14.838.365
48	Shaft 1 Shaft net power	kW	-291,0	-291,0
49	Shaft 1 Shaft speed	RPM	3.000,0	3.000,0
50	Shaft 2 Shaft net power	kW	24.191	24.191
51	Shaft 2 Shaft speed	RPM	3.000,0	3.000,0
52	Shaft 3 Shaft net power	kW	-9,938	-9,938

Figure 2.1: Excel® tool built for the exergy analysis of thermal systems modeled in Thermoflex®.

– There should be no components with unknown or non explicit flow matter.

Two verification procedures can be applied to check if the tool calculations are fine:

A. Satisfactory unbalance is considered to be 0.00 or less. That is, the difference between the overall internal irreversibility and the sum of all equipment internal irreversibilities should approach zero.[1]

B. The calculated total power output (exergy out as electricity) minus the calculated total internal power consumption (exergy in as electricity) should equal the "net power" value indicated in TFX output;

Since the exergy assessment methodology is based on the *energy* outputs from Thermoflex®, it is important to know which hypotheses it assumes for the thermodynamic modeling. The main ones are summarized here below.

1. Combustion is assumed complete. That is, all the carbon present in the fuel is burned to carbon dioxide, all the hydrogen is burned to water, all sulfur is burned to sulfur dioxide, and all other combustible elements are

[1]The accuracy of the code is limited by the accuracy of the TFX internal calculations, i.e., sometimes TFX achieves unbalance values greater than 0.000, for which an unbalance of 0.00 is considered the lowest achievable value based on the tested models.

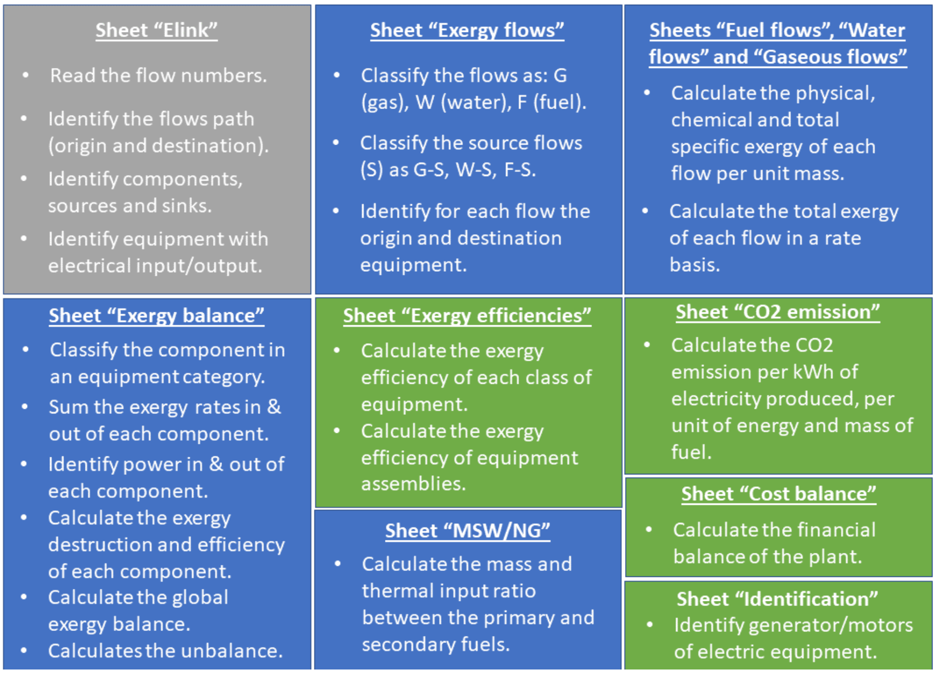


Figure 2.2: Structure of the Excel® tool built to perform exergy analysis: data from TFX is pasted on the gray sheet, blue sheets present calculation and results and green sheets require inputs from the user.

fully oxidized [34]. This means that unburned material and CO formation are not considered.

2. Nitrogen present in the air does not undergo chemical reaction, i.e. it is inert. Hence, TFX does not estimate NO_x formation, even though there is a possibility of adding $DeNO_x$ equipment in the simulated systems.

3. There are only seven possible substances that are considered to compose the gaseous streams of TFX models: N_2, O_2, CO_2, Ar, SO_2, vapor and liquid H_2O.

4. Even though it is not uncommon in real processes that some carbon monoxide and unburned oxygen appear in the products due to incomplete mixing, insufficient time for complete combustion, and other factors are not considered.

Additional hypotheses are:

5. The system is at rest relative to the environment, i.e. potential and kinetic energy/exergy differences are neglected.

6. All gases are ideal.

7. Even though TFX considers some leakages, blow-off and blowdown, such flows are not accounted in the exergy analysis, except when explicitly mentioned. The bottom ash produced from MSW combustion is considered to be kept within the control volume of the system.

8. The reference conditions are 25 °C and 1.013 bar.

9. The slight difference between 1 atm and 1 bar is neglected.

2.1 Exergy analysis method

In the absence of nuclear, magnetic, electrical, and surface tension effects, the *total exergy of a system* E^{tot} can be divided into four components [33]:

$$E^{tot} = E^{ch} + E^{ph} + E^{pot} + E^{kin} \tag{2-1}$$

where on the right side are the chemical, physical, potential and kinetic components of the total exergy, respectively. The sum of the kinetic, potential and physical exergies is also referred to in the literature as the *thermo-mechanical exergy* [33]. From the above-mentioned hypothesis, $E^{pot} = E^{kin} = 0$, eq. 2-1 simplifies to:

$$E^{tot} = E^{ch} + E^{ph} \tag{2-2}$$

E^{ph} is the maximum amount of work that can be obtained from a system as its pressure and temperature (P, T) are changed to the pressure and temperature of the reference-environment (P_0, T_0, also called in the literature as "restricted dead state" [33]). E^{ch} is the maximum amount of work that can be obtained when a stream is brought from the reference-environment state to the dead state (state of null total exergy) [35]. Finally, since the value of the total exergy is defined as the maximum theoretical work obtainable, it is at least zero and therefore cannot be negative [33].

As an extensive property it is convenient to express the exergy in a unit-of-mass (e - often in [kJ/kg]) or molar basis ($\overline{e}$ - often in [kJ/kmol]). That is, in a rate basis $\dot{E}$ (often in [kW]) is obtained by multiplying e by the mass flow rate $\dot{m}$ (often in [kg/s]) or the mole flow rate $\dot{n}$ (often in [kmol/s]):

$$\dot{E} = \dot{m} \cdot e \tag{2-3}$$

$$\dot{E} = \dot{n} \cdot \overline{e} \tag{2-4}$$

Given the output values from TFX regarding specific enthalpy and entropy, it is possible to determine the specific physical exergy (e^{ph}) of a gaseous, liquid of fuel stream i as [34]:

$$e_i^{ph} = h_i - h_{i,0} - T_{i,0} \cdot (s_i - s_{i,0}) \tag{2-5}$$

where h_i [kJ/kg] is the specific enthalpy, s_i [kJ/kg-C] is the specific entropy, the subscript “0” denotes that properties are evaluated at the reference environment conditions (P_0, T_0) and T_0 must be in Kelvin [K]. From such equation, it can be observed that, whenever the stream has the same composition and is at the same P and T conditions as the reference-environment, we have $e^{ph} = 0$ and $e = e^{ch}$. Attention should be paid to refer the stream to the correct reference environment. That is, depending on the characteristics of the stream, whether gaseous/liquid/solid/fuel (pure or mixture), the reference environment should be specified (v. section 2.1.1).

Alternatively, the chemical exergy can also be viewed as the exergy of a substance that is at the reference-environment state [35]. The values for some substances of interest at standard conditions (1.013 bar and 25°C) have been tabulated and are described in tab. 2.1.

Table 2.1: Standard chemical exergy at P=1 atm and T=298 K [34, 35].

Substance	**Standard chemical exergy [kJ/kmol]**
N_2	720
O_2	3970
CO_2	19870
$H_2O_{(g)}$	9500
$H_2O_{(l)}$	900
Ar	11690
SO_2	313400

In energy systems we are frequently dealing with mixture of different gases, such as flue gases from combustion. Assuming each gas mixture is composed of k different ideal gas components, each of which with molar fraction y_k and molar chemical exergy $\overline{e}_k^{ch}$ at P_0, T_0 (obtained from tab. 2.1); the molar chemical exergy of the gas mixture ($\overline{e}_{mix}^{ch}$) can be obtained as [34]:

$$\overline{e}_{mix}^{ch} = \sum_k y_k \, \overline{e}_k^{ch} + \overline{R}\, T_0 \sum_k y_k \ln y_k \tag{2-6}$$

where $\overline{R} = 8.314$ kJ/kmol-K.

Water streams can be found in energy systems as liquid, vapor and mixture of liquid-vapor. Because water is liquid at P_0, T_0 and chemical reactions involving water streams are outside the scope of this analysis, the chemical exergy of water is the same in all points of the cycle, namely 49.96 kJ/kg (or 900 kJ/kmol, v. tab. 2.1).

Calculating the chemical exergy of fuels may be complicated when the fuel is composed of several substances. Bejan et al. [33] affirms that “the use of the higher heating value of a fuel to approximate the fuel chemical exergy is frequently observed in the technical literature". In fact, it has been observed that some authors have correlated the fuel's heating value (HV) and e^{ch} as a ratio. For instance, Dincer et al. [35] describes that $e^{ch}_{fuel,daf} = \Omega \cdot HV_{daf}$, where the subscript "daf" denotes dry and ash free and Ω is a constant. Hence, for the purpose of this thesis it is considered that fuels present their e^{ch} equal to their HHV dry and ash free: [33]:

$$e^{ch}_{fuel} = HHV_{fuel,daf} \tag{2-7}$$

Obviously, for gaseous fuels such as natural gas $HHV_{NG} = HHV_{NG,daf}$, because their ash and moisture content is null. However, for solid fuels such as MSW it is not straightforward to determine $HV_{fuel,daf}$, since most literature/measured data are for the waste as received, i.e. wet MSW with ash. Bejan et al. [33] gives a formula for estimating HHV_{daf} of a solid fuel in MJ/kg, which is assumed here to be valid for MSW [33]:

$$HHV_{MSW,daf} = (152.19\,H + 98.676) \cdot [C/3 + H - (O - S)/8] \tag{2-8}$$

where C, H, O and S are the mass fractions of carbon, hydrogen, oxygen and sulfur, respectively, for MSW dry and ash free, which should be calculated from composition described in tab. 2.2.

Table 2.2: Mass composition of MSW, wet with ash, and LHV at 25 °C (from Thermoflex®).

Substance	**Mass compositon (%)**
LHV wet, w/ ash	10133 kJ/kg
HHV wet w/ash	11675 kJ/kg
Ash	21.00
Moisture	25.20
Carbon	28.10
Hydrogen	3.90
Oxygen	20.60
Nitrogen	0.40
Sulfur	0.30
Chlorine	0.50

Even though Thermoflex® already calculates the properties of the gases at T/P, the exergy assessment requires it to be calculated also at T_0/P_0. The software considers the gas flows as ideal gases, for which $h = h(T)$. It has been noted that Shomate's Equation is used by Thermoflex® to calculate the *molar standard enthalpy* of gases ($\bar{h}^o$) in kJ/kmol, referred to 25 °C [36]:

$$\bar{h}^o - \bar{h}^o_{298.15} = (A\,t + B\,\frac{t^2}{2} + C\,\frac{t^3}{3} + D\,\frac{t^4}{4} - \frac{E}{t} + F - H) \cdot 1000 \tag{2-9}$$

where A, B, C, D, E, F and H are tabulated constants that depend on the gas; $t = T/1000$ where T is the temperature in Kelvin. Thus, this equation is applied to each gas component of the mixture giving $\bar{h}_k^o = h(T)$, then the molar enthalpy of the mixture ($\bar{h}_{mix}^o$) in kJ/kmol is calculated as:

$$\bar{h}_{mix}^o = \sum y_k \cdot \bar{h}_k^o \tag{2-10}$$

where the subscript º denotes that the reference is 25 °C.

There is also the Shomate Equation for calculating the *molar standard entropy* of gases, $\bar{s}^o = \bar{s}(T)$, in kJ/kmol-K [36]:

$$\bar{s}^o = A\,ln(t) + B\,t + C\,\frac{t^2}{2} + D\,\frac{t^3}{3} - \frac{E}{2\,t^2} + G \tag{2-11}$$

where G is also a constant which, similarly to the others, is tabulated depending on the gas. To calculate the molar entropy of each component of a gas mixture it is important to consider the partial pressure of each component k, i.e. $\bar{s}_k = s(T, P_k)$, and the following relation [33]:

$$\bar{s}_k(T, y_k\,P) = \bar{s}_k^o - \bar{R}\,ln\left(\frac{y_k\,P}{P_{ref}}\right) \tag{2-12}$$

where $P_{ref} = 1.013$ bar and P is the total pressure of the gaseous mixture. Then, the molar entropy of the mixture $\bar{s}_{mix}$ is calculated as:

$$\bar{s}_{mix} = \sum y_k \cdot \bar{s}_k \tag{2-13}$$

The gaseous mixtures at T/P present all components in the gaseous phase. However, when they are evaluated at T_0/P_0 some condensation may occur, specially if T is high, such as in the WTE boiler flue gases. In this case, the molar fraction of condensed water has to be determined before applying equations 2-9 to 2-13.

2.1.1 Reference-environment

Since exergy is a measure of the departure of the state of the system from that of the environment, it is therefore an attribute of the system and environment together [33]. As the environment has to do with the actual physical world [33], it is important to choose the correct reference environment for each type of stream as described in tab. 2.3.

The molar composition of the reference air is described in tab. 2.4 as considered by default in Thermoflex® .

Table 2.3: Streams and corresponding reference environment.

Stream at P, T	**Reference environment**
Water	Liquid water at P_0, T_0
Air	Pre-defined air environment at P_0, T_0[1]
Flue gases	Flue gases at P_0, T_0[2]
Gaseous fuel	Gaseous fuel at P_0, T_0
Solid fuel	Solid fuel at T_0
Ash	Pre-defined inorganic environment[3]

[1] Described in tab. 2.4.

[2] Attention should be paid to water condensation that can occur when flue gases are brought to T_0.

[3] Not considered in this methodology due to boundary expansion, v. hypothesis 7.

Table 2.4: Pre-defined air environment @ P=1.013 bar, T=25 C and relative humidity 60% from Thermoflex® .

Substance	**Molar compositon (%)**
N_2	76.62
O_2	20.56
CO_2	0.03
$H_2O_{(g)}$	1.87
$H_2O_{(l)}$	0
Ar	0.92
SO_2	0

2.1.2 Exergy balance in a control volume

One of the major goals of the exergy assessment is to determine the amount of exergy destructed ($\dot{E}_d$) due to entropy generation (2nd Law of Thermodynamics). For this, an exergy rate balance is required in order to quantify the exergy rates entering and leaving the system. That is, $\dot{E}_d$ can be calculated generally as the difference between exergy entering and leaving a control volume (CV):

$$\dot{E}_d = \sum \dot{E}_{entering} - \sum \dot{E}_{leaving} \tag{2-14}$$

Analogous to energy, exergy can be associated to heat ($\dot{E}_q$), work ($\dot{W}_{cv}$) or mass transfer rates. At steady state, the rate of exergy transfer accompanying the power is the power itself [34]. For a general CV operating in *steady state* the exergy rate balance is described by the following equation [34]:

$$\dot{E}_q - \dot{W}_{cv} + \sum_e \dot{E}_e - \sum_s \dot{E}_s - \dot{E}_d = 0 \tag{2-15}$$

where $\dot{E}_q$ is the exergy rate [kW] associated to heat transfer to/from surroundings; $\dot{W}_{cv}$ is the exergy rate [kW] associated to shaft work; $\dot{E}_e$ is the total exergy rate [kW] associated to mass flow at the CV inlet; $\dot{E}_s$ is the total exergy rate [kW] associated to mass flow at the CV outlet; $\dot{E}_d$ is the exergy destruction rate [kW] within the CV. $\dot{E}_d$ is proportional to the entropy generation within the system, often called *irreversibility, availability destruction or lost work* [33], which always assumes positive values or null for ideal systems. To the other terms, the sign convention applies as well, i.e., it is considered as default a negative sign for work and other exiting flows, and a positive sign for heat and other entering flows.

The term $\dot{W}_{cv}$ is easily obtained from TFX accountings of electrical generation and consumption, however, obtaining $\dot{E}_q$ is not as straightforward. It can be obtained precisely through 2-16 [34] if the time rate of heat transfer $\dot{Q}_j$ and the instantaneous average outer surface temperature T_j at the control surface where the heat transfers take place are known.

$$\dot{E}_q = \sum_j \dot{E}_{qj} = \sum_j \dot{Q}_j \left(1 - \frac{T_0}{T_j}\right) \tag{2-16}$$

Actually, there is a frequent misconception to interpret $\dot{E}_q$ due to heat loss (or gain) as part of the exergy destruction, which is not true, since $\dot{E}_q$ and $\dot{E}_d$ are distinct terms. While $\dot{E}_d$ is intrinsic to the process, $\dot{E}_q$ is associated to a heat loss, for instance, it can be diminished by increasing insulation. Therefore, it should be highlighted that whenever $\dot{E}_q$ is different from zero and cannot be determined separately, the result of the exergy balance gives indeed $\dot{E}_d \pm \dot{E}_q$.

Thermal systems operate with major heat source being produced inside combustors through exothermal reactions (combustion of fuel) and then “shared” to the rest of the cycle through heat exchangers. Ideally, combustors and heat exchangers are supposed to “contain” the heat, that is, the heat transfer occurs only internally in such a way that the heat losses to surroundings are supposed to be minimal. Similarly, the other equipment of thermal systems such as turbine, pump, compressor, valve, deaerator, pipe, desuperheater, etc., are expected to have minimal heat transfer from/to surroundings. There is no doubt that, in practice, $\dot{E}_q$ exists, especially due to heat loss from combustors, however to calculate it one should be able to measure T_j and $\dot{Q}_j$. Since such parameters are not always available and their estimation would require an excessive effort, it is rather more practical to consider $\dot{E}_q$ and $\dot{E}_d$ as an ensemble. This means that eq. 2-15 [34] is applied to obtain the system's irreversibility as a sum of the terms $\dot{E}_q$ and $\dot{E}_d$, i.e., it is considered that each component has a *useless amount of exergy* ($\dot{I}$) that is obtained as:

$$\dot{I} = \dot{E}_d - \dot{E}_q = \sum_e \dot{E}_e - \sum_s \dot{E}_s - \dot{W}_{cv} \tag{2-17}$$

which is another representation of eq. 2-14.

Figure 2.3 represents the overall CV of a hybrid WTE-GT plant (only producing electricity), showing all input and output exergy streams. It can be

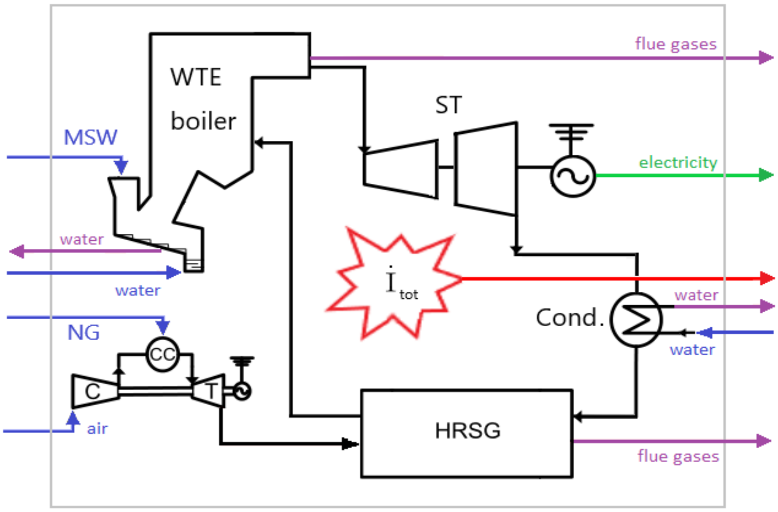


Figure 2.3: Scheme showing the exergy flows crossing the border of a hypothetical WTE-GT system.

observed from such figure that the exergy flows leaving the CV are depicted as red ($\dot{I}_{tot}$), green ($\dot{W}_{net}$) and purple arrows ($\dot{E}_{out}$), whereas exergy flows entering the CV are depicted as blue arrows ($\dot{E}_{in}$). Applying eq. 2-17 to such overall CV results in the following expression:

$$\dot{E}_{in} = \sum \dot{I} + \dot{E}_{out} + \dot{W}_{net} \qquad (2\text{-}18)$$

which can be translated simply as: all exergy entering the system can only exit in the form of electrical power ($\dot{W}_{net}$), total internal irreversibility ($\dot{I}_{tot} = \sum \dot{I}$) or exergy accompanying the overall outer mass flow ($\dot{E}_{out}$). The term $\dot{E}_{out}$ represents the total useless exergy of flue gases and cooling fluids leaving the overall system, which is called *external irreversibility*. Residues from combustion such as ashes or unburnt material, represent potential flows to be accounted in the $\dot{E}_{out}$ term, as well as blow-off from GT compressor, but for the sake of simplicity, those are considered to remain within the boundaries of the overall CV (v. hypothesis 7), and thus are accounted within the $\dot{I}$ term.

In summary, according to Bejan et al. [33], the environment/surroundings of the overall CV is regarded as free of irreversibilities and all significant irreversibilities are located within the system and its immediate surroundings. Internal irreversibilities are those located within the system and external irreversibilities reside in the immediate surroundings [33].

An important parameter in the exergy assessment is the exergetic efficiency or exergy efficiency (ϵ). It is useful to distinguish means for utilizing energy resources that are thermodynamically effective from those that are less so [34]. It can also be used to investigate the effectiveness of engineering measures taken to improve the performance of a thermal system, which is done by comparing the efficiency values determined before and after modifications in order to show how much improvement has been achieved [34]. Moreover, it can be used to gauge the potential for improvement in the performance of a given thermal system by comparing the efficiency of the system to the efficiency of like systems [34]. A significant difference between these values would suggest that improved performance is possible [34]. In a general way, the relation between exergetic efficiency of a thermal system and its thermal efficiency depends on the temperature values of the reference environment and that of the heat transfer surfaces ($\mathrm{T_u}$, $\mathrm{T_{sc}}$) [34]:

$$\begin{aligned}\epsilon &= \frac{(1 - T_0/T_u)\dot{Q}_u}{(1 - T_0/T_{sc})\dot{Q}_{sc}} \\ &= \eta\left(\frac{1 - T_0/T_u}{1 - T_0/T_{sc}}\right)\end{aligned} \tag{2-19}$$

where T_0 is the environment temperature, $\dot{Q}_u$ is the heat transfer rate at use temperature T_u and $\dot{Q}_{sc}$ is the heat transfer rate at source temperature T_{sc}. From eq. 2-19 it can be observed that whenever T_u is similar to T_0, ϵ is very low and whenever T_u approximates to T_{sc}, ϵ tends to unity. In a more specific way, ϵ is a measure of the efficacy in converting an input into a useful product. It can be calculated as ratio between exergy leaving as useful product ($\dot{E}_u$) and entering as source ($\dot{E}_{sc}$) [34]:

$$\epsilon = \frac{\dot{E}_u}{\dot{E}_{sc}} \tag{2-20}$$

The hybrid WTE-GT system is divided into several control volumes in order quantify the internal irreversibilities and exergy efficiencies. In the following subsections eqs. 2-17 and 2-20 are applied to each CV to obtain the values of $\dot{I}$ and ϵ, respectively.

2.1.3
Gas turbine system

The gas turbine system (GTS) is considered to be composed of compressor (CP), combustion chamber (CC) and gas turbine+generator (GT), as shown in fig. 2.4. This is because TFX models the GTS as a single component, where the desired package is chosen from a list of existing commercial

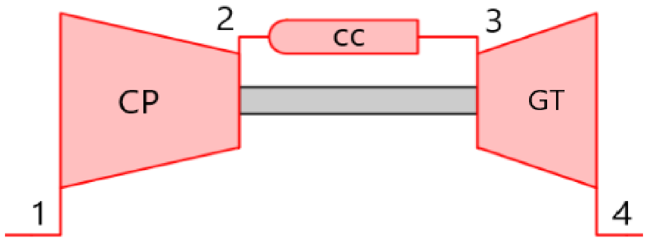


Figure 2.4: Schema of the gas turbine system.

equipment. In this sense, it is convenient to assume the GTS control surface as an imaginary line around fig. 2.4. Applying eq. 2-17 to such CV the following exergy balance is obtained:

$$\dot{I}_{GTS} = \dot{E}_1 + \dot{E}_{NG} - \dot{E}_4 - \dot{W}_{GTS} \tag{2-21}$$

where $\dot{I}_{GTS}$ is the internal irreversibility of the GTS; $\dot{W}_{GTS} = \dot{W}_{GT} - \dot{W}_{CP} - \dot{W}_{GT,aux}$; $\dot{W}_{GT}$ is the GT shaft power output, $\dot{W}_{CP}$ is the compressor power input; $\dot{W}_{GT,aux}$ is the power consumed by GTS internal auxiliary devices; $\dot{E}_{NG}$ is the natural gas total exergy rate; $\dot{E}_4$ is the total exergy rate of the GT exhaust gases and $\dot{E}_1$ is the total exergy rate at compressor inlet.

Since the heat content of GT exhaust gases is a useful product for the bottoming steam cycle of the hybrid WTE plant, based on eq. 2-20 the GTS exergetic efficiency (ϵ_{GTS}) can be calculated as:

$$\epsilon_{GTS} = \frac{\dot{W}_{GTS} + \dot{E}_4}{\dot{E}_{NG}} \tag{2-22}$$

Attention should be paid to account that the fuel inlet stream usually have both chemical and physical exergy components $\dot{E}_{NG} = \dot{E}^{ch}_{NG} + \dot{E}^{ph}_{NG}$. It cannot be neglected that $\dot{E}^{ph}_{NG}$ might be non null, since NG is often added to the combustion chamber in $P, T > P_0, T_0$.

2.1.4 Compressors & pumps

In Thermoflex® the simulation of a compressor or pump sometimes considers it linked to a motor. Applying eq. 2-17 to such ensemble, where flow inlet is at 1 and outlet is at 2, gives its internal irreversibility ($\dot{I}_{cp}$) as the following:

$$\dot{I}_{cp} = \dot{E}_1 - \dot{E}_2 - \dot{W}_{cp} \tag{2-23}$$

where $\dot{W}_{cp}$ is the electrical power consumption (negative) of the ensemble compressor (or pump) + motor (if existing); and $\dot{E}_1$ and $\dot{E}_2$ are the total exergy rate of the fluid at the compressor (or pump) inlet and outlet, respectively.

The exergetic efficiency (ϵ_{cp}) of the compressor (or pump) + motor, based on equation 2-20, is given as:

$$\epsilon_{cp} = \frac{\dot{E}_2 - \dot{E}_1}{-\dot{W}_{cp}} \tag{2-24}$$

2.1.5 Steam turbine

In Thermoflex® the simulation of a turbine always considers it linked to a generator. Applying eq. 2-17 to such ensemble, where steam inlet is at 1 and the outlet is at 2, gives its internal irreversibility ($\dot{I}_{ST}$) as the following:

$$\dot{I}_{ST} = \dot{E}_1 - \dot{E}_2 - \dot{W}_{ST} \tag{2-25}$$

TFX usually accounts for internal power consumptions from auxiliary devices directly connected to the turbines. Thus, it should be considered that $\dot{W}_{ST}$ is the ST *net power output* (positive), i.e., discounted of the auxiliary consumptions: $\dot{W}_{ST} = \dot{W}_{ST,shaft} - \dot{W}_{ST,aux}$, where $\dot{W}_{ST,shaft}$ is the ST shaft power output and $\dot{W}_{ST,aux}$ is the power consumed by auxiliary devices directly connected to the steam turbine.

The steam turbine exergetic efficiency (ϵ_{ST}), based on equation 2-20, can be calculated as:

$$\epsilon_{ST} = \frac{\dot{W}_{ST}}{\dot{E}_1 - \dot{E}_2} \tag{2-26}$$

2.1.6 Heat exchangers without mixing

Even though heat exchangers seem, from the 1st Law point of view, to operate without loss when heat transfer to surroundings is ignored, they are a site of thermodynamical ineffectiveness from the perspective of the 2nd Law [34]. Exergy is destroyed during processes of fluid friction (pressure drop) and heat transfer from stream to stream. In fact, temperature difference between the streams is an indicator of heat transfer irreversibility [34].

Applying eq. 2-17 to a heat exchanger (HX), where both streams[2] hot and cold have inlets at 1 and outlets at 2, gives its internal irreversibility ($\dot{I}_{HX}$) as the following:

$$\dot{I}_{HX} = \dot{E}_{1,h} + \dot{E}_{1,c} - \dot{E}_{2,h} - \dot{E}_{2,c} \tag{2-27}$$

where the subscript "h" denotes hot stream,"c" denotes cold stream.

The exergetic efficiency gauges the exergy increase of the cold stream (useful product) using the exergy decrease from the hot stream (source), thus, based on eq. 2-20, the HX exergy efficiency (ϵ_{HX}) is:

[2] Hot and cold streams should have temperatures above T_0 for the validity of eq. 2-19.

$$\epsilon_{HX} = \frac{\dot{E}_{2,c} - \dot{E}_{1,c}}{\dot{E}_{1,h} - \dot{E}_{2,h}} \tag{2-28}$$

2.1.7
Direct contact heat exchanger

Heat exchangers with mixing of cold and hot streams, such as deaerator (DEA) and water tank (WT), are often present in real energy systems. They have two entering streams (1, 2) but only one exiting stream (3). Analogously to the HX without mixing, the internal irreversibility of the HX *with mixing* ($\dot{I}_{DEA}$) can be obtained from eq. 2-17 [34] as:

$$\begin{aligned} \dot{I}_{DEA} &= \dot{E}_1 + \dot{E}_2 - \dot{E}_3 \\ &= \dot{m}_1 e_1 + \dot{m}_2 e_2 - \dot{m}_3 e_3 \end{aligned} \tag{2-29}$$

From the mass rate balance $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$ and based on eq. 2-20 it is possible to obtain its exergy efficiency (ϵ_{DEA}) as:

$$\epsilon_{DEA} = \frac{\dot{m}_2(e_3 - e_2)}{\dot{m}_1(e_1 - e_3)} \tag{2-30}$$

2.1.8
WTE boiler

A steam cycle furnace or boiler is where fuel is burned for producing heat used to evaporate water. In WTE plants MSW furnace is also called incinerator or waste combustor. In the case when MSW burns over a moving grate (often leaning), with air blown both over and under the grate, the incinerator is of a “grate” type. A schema of the WTE furnace boiler is shown in fig. 2.5, where are also depicted the two major zones of heat transfer between the combustions gases and water, namely convective (CONV) and radiative (RAD). The radiative zone is closer to the grate (first passages of the combustion gases), thus, with higher temperatures, where *radiative evaporators* are placed. The convective zone is the following passage of the combustion gases, where smaller temperatures are found with respect to the RAD zone. In CONV zone are placed heat exchangers such as *convective evaporators*, *superheaters* and *economizers*.

An imaginary rectangle can be pictured around fig. 2.5 to represent the control surface, across which the different flows enter and leave the CV. The observed mass flows entering the CV are: water to be evaporated, waste fuel, both air flows (primary and secondary) and, in this case, liquid water for grate cooling (optional, because cooling can also be done with air). The mass flows leaving the CV are flue gases, steam and liquid water from grate cooling, whereas the bottom ash is considered to remain withing the CV (v. hypothesis 7 in the beginning of this chapter).

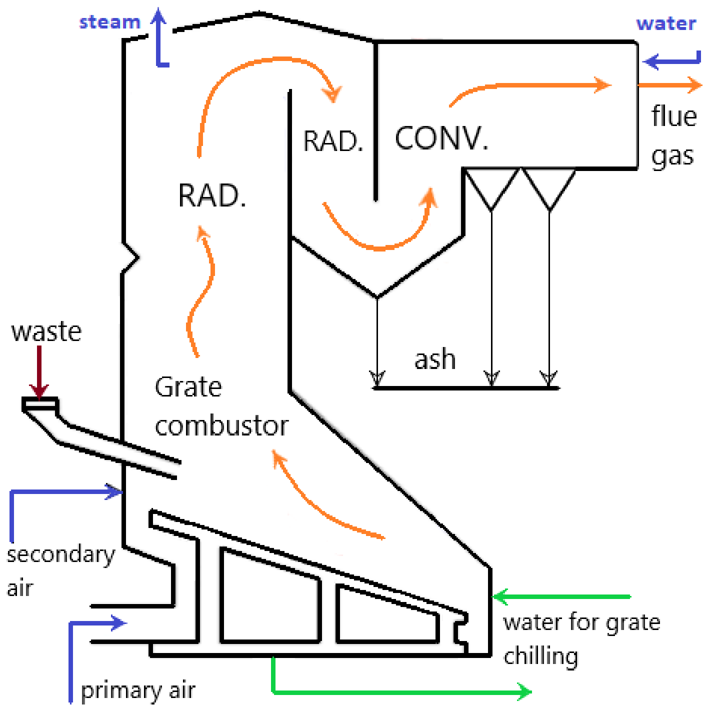


Figure 2.5: Schema of the WTE boiler.

In this case, the WTE boiler internal irreversibility ($\dot{I}_{inc}$) can be calculated from eq. 2-17 as:

$$\dot{I}_{inc} = \dot{E}_{a1,in} + \dot{E}_{a2,in} + \dot{E}_{v,in} + \dot{E}_{w,in} + \dot{E}_{MSW} - \dot{E}_{v,out} - \dot{E}_{w,out} - \dot{E}_{flue} \quad (2\text{-}31)$$

where $\dot{E}_{a1,in}$ and $\dot{E}_{a2,in}$ are the air total exergy rates entering the CV under and over the grate, respectively; $\dot{E}_{v,out}$ and $\dot{E}_{v,in}$ are the total exergies of steam at boiler outlet and water at boiler inlet, respectively; $\dot{E}_{w,in}$ and $\dot{E}_{w,out}$ are the total exergy of liquid water entering and leaving the CV for grate cooling, respectively; $\dot{E}_{flue}$ is the flue gases total exergy rate; and $\dot{E}_{MSW}$ is the MSW chemical exergy rate (MSW enters the CV hypothetically at T_0). The bottom ash exergy is neglected due to hypothesis 7, as a result its contribution is included in the value of $\dot{I}_{inc}$.

The incinerator exergetic efficiency is given as:

$$\epsilon_{inc} = (\dot{E}_{v,out} - \dot{E}_{v,in})/\dot{E}_{MSW} \quad (2\text{-}32)$$

2.1.9
Valves and pipes

Although energy is conserved in pipes and throttling valves, exergy is destroyed due to friction and uncontrolled expansion [34]. The irreversibility in a valve or pipe ($\dot{I}_{vl}$) with one inlet (1) and one outlet (2) can be obtained by applying eq. 2-17 simply as:

$$\dot{I}_{vl} = \dot{E}_1 - \dot{E}_2 \tag{2-33}$$

Similar logics can be applied to components with more than one inlet/outlet, such as mixers and splitters.

2.1.10
Flue gas cleaning system

The pollution control is very important to the sustainability of WTE plants, without which it is not possible to accomplish with environmental laws for waste burn. In general, emissions from the gas turbine are not treated, being the pollution abatement components destined to treat only MSW flue gases. It is assumed that the flue gas cleaning system (FGS) is composed of an equipment described in TFX library as "Dry FGD - Lime Spray Drying Flue Gas Desulfurization". Basically it is a semi-dry scrubber followed by a fabric filter, as represented in fig. 2.6. Also used in the Spanish plant of Zabalgarbi/Bilbao, this abatement route is able to control the main pollutants from MSW combustion, such as acid gases, dust, dioxins & furans. In particular, nitrogen oxides (NO_x) are often abated directly inside the furnace through controlled-combustion procedures and addition of chemicals. Since TFX does not perform NO_x emission calculations, such procedures are not simulated.

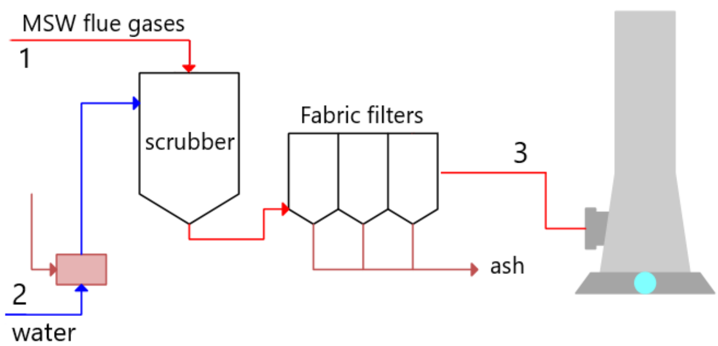


Figure 2.6: Schema of the MSW flue gas cleaning system.

Imagining the FGS control surface as a rectangle around fig. 2.6, excluding the chimney, the only outer flow is the cleaned flue gas (point 3) because water and ash remain within the CV (v. hypothesis 7 in the beginning of the chapter). Moreover, since TFX does not make explicit the details about the chemical products addition (represented by the purple rectangle with an arrow in fig. 2.6), it is not possible to calculate the exergy of such chemical flow, for which it is not included in the balance. Thus, the FGS internal irreversibility ($\dot{I}_{FGS}$) can be calculated from eq. 2-17 as:

$$\dot{I}_{FGS} = \dot{E}_1 + \dot{E}_2 - \dot{E}_3 \tag{2-34}$$

where $\dot{E}_1$ is the dirty flue gas total exergy rate entering the CV; $\dot{E}_2$ is the liquid water total exergy rate; and $\dot{E}_3$ is the total exergy rate of the cleaned flue gas entering the chimney.

2.2 Conclusive remarks

The method consisted in elaborating an Excel code to calculate exergy balances of cycles fully modeled in Thermoflex® and run via its Excel® component E-link. The developed tool is called “3E EXC” because, besides of providing an exergy assessment, it also performs simplified CO_2 (environmental) and cost balances (economic). This tool will be useful in several analysis of the many cycles simulated in TFX, as it will be shown in the following chapters. The greatest challenge so far has been to obtain a satisfactory error of the exergy balance, that is, the overall internal irreversibility should be equal to the sum of all equipment internal irreversibilities. In this sense, the code has shown to be satisfactory, presenting an unbalance of 0.00 in all tests. It is important to highlight that this accuracy is limited by Thermoflex, that is, since not all times TFX internal calculations achieve unbalances of 0.000, such error source is also reflected in the 3E EXC tool.

3
4E analysis and feasibility of hybrid waste-to-energy plants

This chapter presents a method called energy-exergy-environmental-economic (4E) analysis applied to a hypothetical hybrid waste-to-energy (WTE-GT) plant. The thermodynamic cycle is built using the software Engineering Equation Solver®. Many studies have evaluated the performance of new configurations of integrated waste-to-energy/gas turbine cycles, but a question still remains: "Instead of building new natural gas power plants or installing conventional waste-to-energy facilities, should one be investing in flexible efficient hybrid plants?" This chapter aims at presenting a novel comprehensive approach to help answering this question. The strategy intends to demonstrate how to integrate four known key procedures (energy, exergy, economic and environmental analysis) in an original manner in order to evaluate the feasibility of such systems.

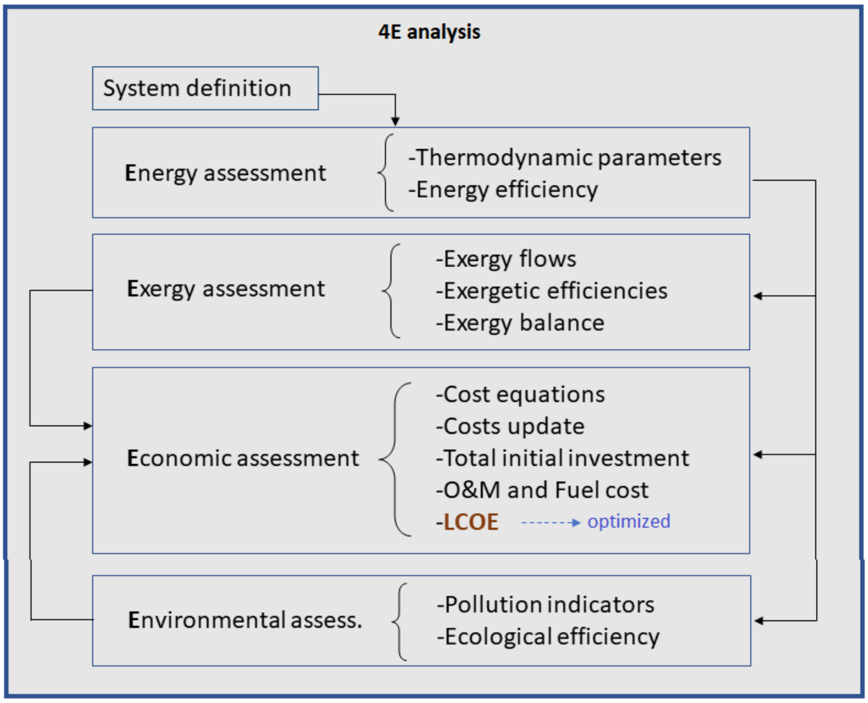


Figure 3.1: The 4E analysis structure.

A scheme of the technique is shown in fig. 3.1. It consists of a conventional Energy-Exergy analysis followed by a new Environmental-Economic approach.

The novelties are in the economic and environmental parts, namely: i) the use of multiple cost equations for estimating the plant's initial investment range and ii) the indirect estimate of the pollution abatement system cost through an energy-ecological efficiency indicator. A case study configuration (similar to the existing plant of Bilbao, v. fig. 3.2) is used as exercise to demonstrate the method and its comparability potential, as well as to allow discussion using tangible results. The case-studied plant has a power output capacity of 107 MW_e, thermal efficiency of 36%, ~57% of MSW share, ecological efficiency of 89%, thermal waste input capacity of 155 MW_t and levelized cost of electricity production of ~US$ 100 per MWh. It is compared to several existing single-fueled waste-to-energy facilities and other energy sources, including renewable and non-renewable. As unique findings of this research, it is shown that the proposed economic method allows to predict costs quite accurately and that the specific investment costs of such technology are very attractive compared to existing single-fueled waste-to-energy facilities in Europe and other electricity sources in the Brazilian context. The proposed method proved to be a comprehensive procedure to analytically evaluate the feasibility of hybrid waste-to-energy power plants, which present good potential in the Energy conversion field.

3.1 Introduction

No study has been found showing an energy, exergy, environmental and economic (4E) analysis method to evaluate hybrid WTE-GT power plants quantifying exergy destruction. Most of them apply energy/economic/environmental (3E) analysis (exclude exergy) and/or do not consider the cost of emission pollution abatement system nor levelized cost of electricity production (LCOE)/comparison with other sources. In addition, when numerical approaches are used, most of them apply commercial software, which are often expensive and inaccessible to young students or researchers in less privileged Institutes. Besides that, those tools do not provide open access to all assumptions or equations involved in the economic and environmental analyses. Therefore, the main objective of the present chapter is to present a new comprehensive method to analytically assess the 4E performance of hybrid WTE-GT plants, allowing the user to compare it to other power systems of different sources. Additionally, it aims at presenting unique quantified information about the advantages of such plants in hopes of fostering their research and application. The originality of the proposed procedure includes the use of a wide array of cost equations to provide a larger capital expenditure (CAPEX) range estimate with the environmental cost being estimated through a specific indicator. In addition, as findings of the application of such method to a chosen cycle, different sensitivity analysis are performed and unique comparisons are made between the obtained costs of several sources. The method allows for the possibility of being applied to other systems in countries worldwide providing a useful tool for researchers and decision makers. A case study is chosen to explain the method because it allows to explore its comparability potential and present discussion based on tangible results. Hence, the goals of the study are to contribute both in terms of methodological knowledge (by proposing a method) and enhancing understanding of WTE-GT technology.

Regarding the case-studied plant, it consists of an original design of a WTE-GT system based on the existing plant of Bilbao (Spain), whose operating conditions were chosen within the context of a large-sized city (Rio de Janeiro) of a developing country. As all gas-fueled power plants, it may be of particular interest to countries where natural gas is available as a reasonable source. Additionally, it is the first time that such specific configuration is investigated in the Brazilian context. In order to situate the present study and its contribution relative to other recent WTE researches, either in regional or global applications, it is possible to mention:

– Branchini [12] investigates the thermodynamic performance of several configurations of WTE-GT systems, but it does not consider exergy, economic nor environmental aspects.

– Bianchi et al. [25] evaluate the integration of a conventional WTE power plant with a gas turbine. Bianchi et al. [37] focus on the repowering of existing under-utilized WTE power plants with gas turbines. However, those studies do not consider environmental or detailed economic aspects.

– Consonni [16] performs a 3E analysis of WTE-GT compared to conventional WTE plants considering different scales. Consonni & Silva [38] evaluate off-design performance of integrated waste-to-energy, combined cycle plants. Udomsri et al. [39] investigate thermo-economic aspects of WTE-GT systems. Udomsri et al. [9] focus on the environmental aspects of such systems. However, those studies do not present exergy analysis, information on cost updates, details on emissions of pollutants other than CO_2, nor a comparison with other sources.

– Qiu & Hayden [40] explore the energy viability of two WTE-GT configurations, including a fluidized bed MSW boiler, but they do not take into account the pollution abatement system, nor economic/exergy aspects.

- Poma et al. [41] investigate thermo-economic aspects of WTE-GT systems, including exergoeconomic concepts, for electricity and combined heat and power (CHP) production in the context of an Italian province, but such work does not include environmental aspects.
- Tan et al. [42] evaluate energy and carbon reduction potential in Malaysia for various WTE strategies and calculates the potential for carbon emission avoidance resulting from the displacement of fossil fuels. Tan et al. [43] evaluate energetic, economic and environmental impacts of four waste treatment alternatives. Nizami et al. [44] proposes a strategy for waste management in Saudi Arabia, including a number of WTE technologies. All those studies assess energy, economic and environmental aspects of different WTE technologies, with applications in Asia and the Middle-East, but those do not include details on the configuration of power cycles nor an exergy analysis.
- Rocco & Colombo [45] presents an exergy life cycle assessment of a waste-to-energy plant. Rocco et al. [32] calculate the exergy destructions and investigate environmental impacts in terms of primary non-renewable resources displacement. Both studies perform a 4E analysis of WTE plants but do not include a WTE-GT configuration nor the calculation of LCOE or comparison with other sources.
- Rigamonti et al. [46] proposes an indicator for the assessment of the environmental and economic performance of WTE systems, considering thermodynamic (energy and exergy) aspects. However, the approach is focused on integrated MSW management systems in general and does not include details on the configuration of a hybrid power cycle.
- Ferreira & Balestieri [15] present, a comparative analysis of waste-to-energy alternatives for a low-capacity power plant in Brazil. Leme et al. [47] present a techno-economic analysis and environmental impact assessment of energy recovery from waste in Brazil. Balcazar et al. [27] perform an analysis of hybrid waste-to-energy for medium-sized cities. Ribeiro & Sioen [6] design of a high efficiency waste to energy plant in Brazil. Ribeiro & Kimberlin [28] investigate high efficiency waste to energy power plants combining waste and natural gas or ethanol. Holanda & Balestieri [23] optimize gas cleaning routes in waste incineration steam cycle. Holanda & Balestieri [29] optimize gas cleaning routes in waste incineration combined gas/steam cycle. All those studies investigate WTE systems in the Brazilian context, including energy, environmental and economic aspects. However, none of them develops an exergy analysis to

> quantify irreversibilities nor uses ecological efficiency to estimate the air pollution control cost. Among those, only two studies calculate the LCOE of MSW incineration facilities but make a comparison only with other WTE technologies. Ferreira & Balestieri [15] make a comparison between WTE-GT and gasification. Leme et al. [47] confront biogas-fueled systems and MSW incineration. Hence, none of them makes comparisons between WTE systems and other sources of electricity production.

From the consulted literature described above an in section 1.2, 12% of the publications presented a complete 4E analysis of power plants. From those, only four articles regarded MSW incineration and none of them included electricity cost estimation nor comparison between diverse power sources. As mentioned, from the broad bibliographic review developed, it is possible to conclude that this is also the first study applying the ecological efficiency method to a MSW/NG-fired hybrid combined cycle. Hence, this chapter aims at filling such gaps by presenting a new comprehensive method to evaluate four aspects (energy, exergy, environment and economy) of a chosen configuration of WTE-GT, used as an example to explain the technique. An analytical approach is adopted so it does not make use of commercial software. This has the major advantage of granting transparency to all steps of the analysis, besides of not requiring access to expensive tools. The procedure is applied to a new configuration, similar to the Zabalgarbi plant (operating in Bilbao, Spain). Besides of basic mass and energy balances, the strategy includes an exergy assessment which, besides of serving as input to the economic analysis, allows the investigation aiming at future improvement. The economic and environmental parts of the analysis hold the method's originality. The first novelty consists in applying an indicator based on the atmospheric emissions to estimate the cost of the pollution control equipment without the need to specify its route. The second original contribution regards the estimate of the total initial investment cost range (overnight CAPEX) using a wide array of cost equations. Such costs are then compared to eleven important standalone WTE plants in Europe and Brazil, including the envisioned URE Barueri (predicted as the first WTE plant of Brazil). In addition, the study's major local contribution is to compare the LCOE of an WTE-GT system to other energy sources (fossil, biomass, wind, hydroelectric, etc.) in Brazil. Finally, a parametric analysis is developed evaluating the influence of the plant's availability, NG price, gate fee, depreciation and maintenance on the LCOE.

3.2 Method

As mentioned, this chapter proposes a method called "4E analysis" that is supposed to be applicable to all sorts of thermopower plants. It interconnects the four "e's" - **E**nergy, **E**xergy, **E**nvironmental, **E**conomic, through four links: energy↔exergy, energy↔economic, energy↔environmental and exergy↔economic. The only link that is not directly made is exergy↔environmental. The strategy consists of conducting energy and exergy analysis followed by environmental and economic assessments.

Defining the system's configuration and basic thermodynamic parameters is a preliminary step. The energy analysis is the first fundamental assessment in the procedure because it supplies all the following analyses in the 4E methodology. An important goal of such step is to determine the 1st Law efficiency of the system along with other useful parameters. Afterwards, an exergy analysis is performed for determining the exergy losses and efficiencies of the system's components according to section 2.1.2. Subsequently, an environmental assessment is carried out by evaluating the atmospheric pollutants' emissions in order to obtain the ecological efficiency (EE). EE, calculated using approach III, as described in section 5.3.3, is then used for estimating the cost of the environmental pollution control system. Finally, an economic assessment uses cost equations proposed by several authors to estimate the acquisition values of the main equipment and obtain the plant's CAPEX range. This is so because often literature presents estimates of energy systems investments expressed as ranges rather than a single value. After defining the operation and maintenance costs, the levelized cost of electricity can finally be determined, allowing for the feasibility investigation through comparisons with further energy sources.

3.2.1 System characterization

In order to explain the procedure, a particular configuration of a MSW and NG-fired combined cycle is used as a case study contextualized in the city of Rio de Janeiro. This city is chosen because it has abundant sources of MSW and NG. The most attractive types of hybrid dual-fuel combined cycle configurations for improved energy conversion efficiency of MSW are governed by the possibility to employ superheating of the steam in the gas turbine exhaust [48]. According to Branchini [12] there are two basic types of arrangements: "steam/water side integrated HCC" and "windbox repowering". In the first one, thermal energy from the topping cycle exhaust is utilized for feedwater preheating and/or steam superheating and/or additional steam generation parallel to the bottoming cycle [12]. In the windbox repowering, the topping cycle exhaust, with or without pre-cooling, is supplied to the bottoming boiler and used as combustion air for firing the bottoming cycle fuel [12]. The method is applied to a configuration of the first type, as shown in fig. 3.2, where only the most relevant components are depicted.

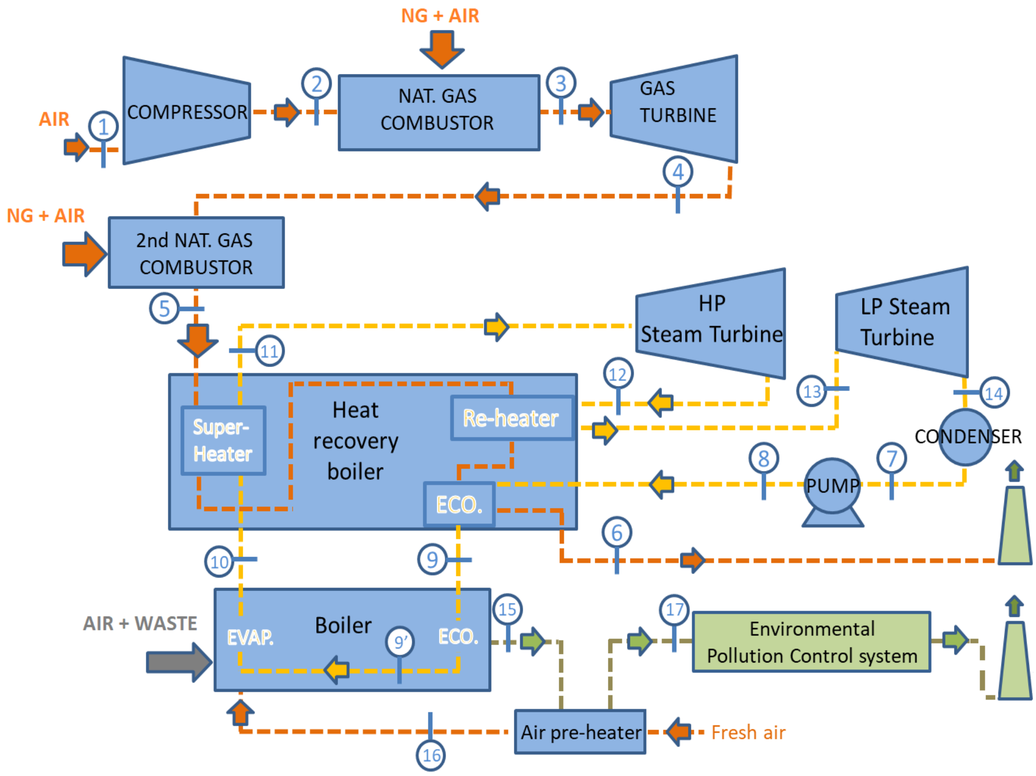


Figure 3.2: Diagram of the investigated WTE-GT system.

In summary, the hybrid combined cycle can be divided in two main parts, topping and bottoming cycle, as described below:

I. Topping cycle - The fluid pathway is represented by the red dashed line numbered from 1 to 6 (v. fig. 3.2). It starts with ambient air being compressed in a compressor (CP) and addressed to a combustion chamber (CC), where NG is combusted (first NG combustor), and then expanded in a GT for electricity generation. The thermal energy of the high temperature gases exiting the GT is used to superheat the steam in a HRSG (here called heat recovery boiler – HRB) with supplementary firing (second NG combustor). The GT flue gases are released through a stack without treatment. As also pointed out by Bianchi et al. [25], such arrangement has the purpose of eliminating the corrosion problems into the superheater of the WTE by moving the steam superheating process to inside the HRSG.

II. Bottoming cycle - MSW is incinerated on a firing grate furnace (also called moving grate furnace or incinerator, in fig. 3.2 indicated as "Boiler"). The air is pre-heated before entering the furnace (point 16). From the furnace exit (point 15, green line – fig. 3.2), the flue gases pass through an air pre-heater and then enter the environmental control system (point 17) for pollution abatement. The water/steam pathway is represented by the yellow dashed line numbered from 7 to 14 in fig. 3.2. It starts with saturated liquid water being pumped from the condenser exit (point 7) to the economizer (ECO) in the HRSG (point 8). The liquid water passes through another economizer and a steam generator (boiler at 100 bar), where it achieves the vapor state (point 10) as exiting the boiler. The vapor is superheated to a sufficiently high temperature at the HRSG (point 11) thanks to a second NG combustor placed at the HRSG inlet. After superheated, the steam is expanded in a two-stage ST for electricity generation before exiting the ST (point 14). Saturated vapor coming from the high-pressure steam turbine (HP-ST) is extracted to a re-heater (point 12) in the HRSG and then addressed to the low-pressure steam turbine (LP-ST) (point 13).

The proposed configuration is based on the one of Zabalgabi/Bilbao plant, with several simplifications in the layout, but adopting as much as possible similar operative parameters. The system's evaporation pressure is 100 bar and the superheating temperature is 506 °C such as in Bilbao, however the MSW capacity is greater than Bilbao's: MSW thermal input capacity is 155 MW_t (MSW LHV of 10 MJ/kg [49] and processing capacity is 56 t/h), which corresponds to a large-sized plant.

3.2.2 Energy analysis

The energy analysis aims at determining the properties (pressure, temperature and physical state) of the fluids at each point and important thermodynamic parameters such as mass flow rates, thermal flows, power production, thermal efficiencies, heat transfer rates and fuel consumption. For this, the following hypotheses are assumed:

i. The system operates in steady state and each component of the system is analyzed as a control volume

ii. Kinetic and potential energy variations are negligible.

iii. All gaseous fluids properties are calculated assuming dry air (composed by N_2, O_2 and Ar).

iv. Duct burner (or 2nd natural gas combustor) has GT flue gases as only oxidant, i.e., no additional fresh air is added. The exhaust gases' temperature at duct burner outlet is calculated assuming complete combustion.

v. Mass flow of MSW exhaust gases is considered as the sum of MSW and air mass flows entering the boiler (i.e. no ash outflow, thus ash is kept within the CV).

vi. Parasitic load of the plant is calculated as $150\,\mathrm{kW\,h\,t_{MSW}^{-1}}$ plus the pump power.

vii. Pressure drop in heat exchangers are considered but those in valves, pipes, nozzles and other local pressure drops are not considered.

viii. Heat exchangers and condenser are adiabatic. Heat losses to surroundings are considered only for combustors.

ix. Air mass flow for MSW combustion is unique, i.e., not distinguished between overfire and undergrate.

x. Steam mass flow rate $\dot{m}_v$ is constant in the vapor cycle and equal to a specified value assumed as input (in the investigated case 55.4 kg/s);

xi. Effects of partial radiant flux to waterwall are not considered, i.e., all heat from MSW combustion is considered to be available for heat transfer inside the furnace, except the portion lost to surroundings.

xii. Temperature and pressure of exhaust gases from MSW combustion at furnace outlet are given as inputs. Adiabatic temperature of MSW combustion is calculated assuming flue gases composition as dry air.

xiii. Mechanical and electrical losses in turbine and electric generator, boiler blow down, compressor blow-off, incomplete combustion and mass losses/leakages are neglected.

The gross and net 1st Law efficiencies of the overall cycle are calculated respectively as:

$$\eta_{tot} = \frac{\dot{W}_{tot}}{\dot{Q}_{MSW} + \dot{Q}_{cc} + \dot{Q}_{SF}} \tag{3-1}$$

$$\eta_{net} = \frac{\dot{W}_{net,ST} + \dot{W}_{net,GT}}{\dot{Q}_{MSW} + \dot{Q}_{cc} + \dot{Q}_{SF}} \tag{3-2}$$

where $\dot{Q}_{MSW}$, $\dot{Q}_{cc}$ and $\dot{Q}_{SF}$ are the thermal input power of fuels at the incinerator, GT combustion chamber and 2nd NG combustor (or duct burner). $\dot{W}_{net,ST}$ is the net power output of the steam turbines and $\dot{W}_{net,GT}$ is the net power output of the gas turbine.

Two alternatives are proposed to structure the thermodynamic calculations. The first one regards describing and solving all equations of mass and energy balance analytically, without the use of any software. The second one makes use of some Engineering Equation Solver® routines, requiring a smaller number of analytical equations than the first. Even though the input variables choice may differ slightly in both methods, they should give the same results. Equations of the first method are listed in Appendix A, as described in [50]. The second method is detailed in subsections 3.2.2.1 to 3.2.2.15, where the energy analysis is structured per control volume (indicated by the working fluid inlet-outlet numeration between parenthesis). EES routines apply to certain CVs (combustors, heat exchangers, turbines and compressors), and are used to solve mass and energy balances quickly. Nevertheless, it should be highlighted that it is not always preferable to use such routines because they tend to tighten the analysis, sometimes resulting in undesired results specially when performing parametric analysis or when operating in off-design conditions.

3.2.2.1
GT Compressor (1-2)

From literature it is found that the isentropic efficiency of the GT compressor ranges between 72-87%[1] [51]. Assuming N_{cp}=0.8 as the CP isentropic efficiency, $\dot{m}_{air1}$=122.84 kg/s as the mass flow rate of air at compressor inlet, CP_{rate}=29.1 as the compressor rate and knowing P_1= 1.01 bar, T_1=293.15 K; the EES routine “Compressor 2” is used to obtain $\dot{W}_{cp}$ (compressor power input) and h_2. From $P_2 = P_1 \cdot CP_{rate}$ and h_2, the value of T_2 is determined.

3.2.2.2
GT combustion chamber (2-3)

The loss of pressure in a combustor is a major problem since it affects both the fuel consumption and power output. According to [52], pressure loss occurs in a combustor because of diffusion, friction/ momentum and it ranges between 2-10% of static pressure (compressor outlet pressure). In [53] the CC pressure drop is assumed as 5%. In this sense, it is here assumed DP_{cc}=3% as the pressure drop normalized by the inlet pressure. Moreover, assuming AF_1=63 as the air-fuel ratio, LHV_{NG}= 47730 kJ/kg and N_{cc}=0.98 as the CC

[1] Part-load operation is outside the scope of this work.

thermal efficiency (based on the LHV) [54]; the EES routine "Combustor 1" is used to obtain T_3, P_3, the outlet mass flow rate of flue gases ($\dot{m}_3$) and the heat rate transfered to the fluid ($\dot{Q}_{23}$).

3.2.2.3 Gas turbine (3-4)

Assuming the gas pressure losses in the heat exchangers of the HRSG as PL_{SH}, PL_{RH} and PL_{ECO} (standing for superheater, re-heater and economizer, respectively), knowing that $P_4 = P_1 + PL_{SH} + PL_{RH} + PL_{ECO}$ and assuming N_{GT}=94% as the GT isentropic efficiency; EES routine "Turbine 2" is used to obtain h_4, from which T_4 is determined, and the power produced by the fluid expansion in the gas turbine ($\dot{W}_{GT}$, which does not discount the CP power).

3.2.2.4 Gas turbine power cycle (GTS)

The net power output of the GTS machine is:

$$\dot{W}_{net,GT} = \dot{W}_{GT} - \dot{W}_{cp} \tag{3-3}$$

The thermal efficiency of the gas turbine power cycle is:

$$\eta_{GT} = \dot{W}_{net,GT}/\dot{Q}_{cc} \tag{3-4}$$

$$\dot{Q}_{cc} = LHV_{ng} \cdot \dot{m}_{ng1} \tag{3-5}$$

where $\dot{m}_{ng1}$ is the natural gas mass flow rate at CC.

3.2.2.5 Duct burner (4-5)

According to Ganapathy [55], a supplementary-fired HRSG has a duct burner (DB) located upstream. In fact, a supplementary firing device or duct burner (referred as "2$^{\text{nd}}$ NG combustor" in fig. 3.2) is placed at the HRSG inlet, where the addition of supplementary combustion air is optional. Typically, a duct burner has a rectangular cross-section and fits into the ductwork carrying the exhaust gases where, in general, no additional air is used because there is 13-15% of oxygen by volume in the GT exhaust flow [55]. In this sense, it is considered that there is no additional air added to the DB.

In order to have a similar air-fuel ratio as in the CC, it is assumed the DB air-fuel ratio is $AF_2 = 3.64\, AF_1$. Neglecting the pressure loss ($D_{SF} = 0$) and assuming DB thermal efficiency as N_{SF}=1; EES routine "Combustor 1" is used to give the heat transfer rate to the fluid ($\dot{Q}_{SF}$), the mass flow rate at point 5 ($\dot{m}_5$), T_5 and P_5 (from which h_5 can be determined). The mass flow rate of NG at DB is determined by a simple mass balance (with $\dot{m}_3 = \dot{m}_4$):

$$\dot{m}_5 = \dot{m}_3 + \dot{m}_{ng2} + \dot{m}_{air2} \tag{3-6}$$

3.2.2.6 Superheater (10-11)

According to [55], the starting point for determining gas and steam temperature profiles in heat exchangers is the assumption of pinch and approach points. In fact, the EES routine for heat exchangers requires as input the approach temperature difference (T_{ap}). In [55] it is suggested T_{ap} range is 10-40 °C. Hence, 18 K is adopted as the superheater approach temperature difference.

Also based on [55], it is assumed that pressure losses of water and gas in the superheater are 8% and 1%, respectively. Other assumptions for the SH inlet are: P_{10}=100 bar and x_{10}=1 (saturated steam). Then, using EES routine "Heat_Exchanger 2" gives h_{11}, P_{11} (from which T_{11} is obtained), the heat transfer rate to steam ($\dot{Q}_{SH}$), and the pressure and enthalpy ($P_{gout,SH}$, $h_{gout,SH}$) of flue gases at SH outlet.

3.2.2.7 High pressure ST (11-12)

According to [12], the steam turbine isentropic efficiency range is 80-90%. Other sources [56], [25] assume 85% and 90%. Consulting Thermoflex® library for an additional reference, it is found that the default ST "dry step efficiency" is 85%, however, it varies with the stage (ranging between 87.5-92.2% in the investigated cases) and being higher for LP than for HP. In this sense, it is assumed N_{STA}=90% as the isentropic efficiency of HP ST. Additionally, based on educated guess it is assumed P_{12}=694 kPa, then using EES routine "Turbine 2" gives h_{12} (from which T_{12} is determined) and the power output of the HP-ST ($\dot{W}_{STA}$).

3.2.2.8 Re-heater (12-13)

The same assumptions for pressure losses and approach temperature difference applied to SH are adopted for the re-heater; then using EES routine "Heat_Exchanger 2" gives h_{13}, P_{13} (from which T_{13} is obtained), the heat transfer rate to steam ($\dot{Q}_{RH}$), and the pressure and enthalpy ($P_{gout,RH}$, $h_{gout,RH}$) of flue gases at RH outlet.

3.2.2.9
Low pressure ST (13-14)

As for the HP-ST, it is assumed N_{STB}=90% as the isentropic efficiency of LP ST. In [25] and [12] the condenser pressure is 0.1 bar. Other assumptions are 0.07-0.1 bar [16] and 0.056 bar [9]. In this sense, it is assumed P_{14}=12.38 kPa (0.12 bar) as the condenser pressure. Then using EES routine "Turbine 2" gives h_{14} (from which T_{14} is determined) and the power output of the LP-ST ($\dot{W}_{STB}$).

3.2.2.10
Condenser (14-7)

Additionally to the assumptions about the condenser pressure mentioned in section 3.2.2.9 and the approach temperature difference mentioned in sections 3.2.2.6 and 3.2.2.8, it is assumed $T_{w,in}$=20 °C as the cooling water temperature at condenser inlet. Many authors neglect pressure drop in the condenser or do not specify it. Since EES routine used to model the condenser assumes the cold side as a reservoir, pressure loss at the cold side is neglected. Thus, it is assumed that the condenser has no pressure loss also in the hot side. Using EES routine "Heat exchanger 1" gives h_7 and P_7 (from which T_7 is determined).

The condenser cooling water mass flow rate can be determined as:

$$\dot{m}_w = \dot{m}_v(h_8 - h_1)/(h_{wout} - h_{win}) \tag{3-7}$$

where $\dot{m}_w$ is the mass flow of cooling water at condenser in [kg/s], h_{wout} and h_{wout} are the enthalpies of cooling water at inlet and outlet of condenser, respectively.

As noticed, two different EES routines were applied to heat exchangers so far: "Heat exchanger 1" and "Heat exchanger 2". It should be highlighted that the first one is suited only for condensers because it assumes the heat transfer occurs between a hot stream and a cold reservoir (constant temperature). This is most applicable in cases of plants located at harbours, where sea/river water can be used as cooling water. Actually, authors in [57] affirm that this is the case of the Amsterdam WTE plant, which allows lower condensing pressures around 0.03 bar.

3.2.2.11
Pump (14-7)

The pump isentropic efficiency ranges between 70-75% [56], [58]. Consulting Thermoflex® library for an additional reference, it is found that the default isentropic efficiency of an integral motor pump is 75%. It is assumed N_{pp}=80% as the pump isentropic efficiency and P_8 is calculated from P_{10} and the pressure losses already assumed for the other components. Then, using EES routine "Compressor 2" gives h_8 (from which T_8 is determined) and the pump power input ($\dot{W}_{pp}$).

3.2.2.12
HRSG Economizer (8-9)

Because flue gases should be cooled to a minimum temperature of 110 °C [12], the approach temperature difference in the HRSG economizer is assumed as 77 K. Moreover, water pressure loss in the economizer is assumed as 25% [16], while gas pressure loss is assumed 0.79 kPa [55]. Using EES routine "Heat exchanger 2" gives h_6, P_6, h_9, P_9 (from which T_6 and T_9 are determined) and the heat rate transfered to water ($\dot{Q}_{ECO}$).

3.2.2.13
Evaporator (9'-10)

It is assumed that the WTE boiler evaporator receives steam as saturated liquid, i.e. $x_{9'} = 0$, and the water pressure drop is assumed as 5% [59], [54].

3.2.2.14
Boiler (9-10)

As observed in fig.3.2, the WTE boiler is an assembly that contains the evaporator and an economizer (9-9'). It is considered that it processes a MSW mass flow rate of $\dot{m}_{MSW}$=15.5 kg/s (annual maximum input of 488.889 t/y). As mentioned, LHV_{MSW} is considered as 10 MJ/kg [49].

The main heat loss in a WTE boiler is due to loss of sensible heat with flue gases exiting the furnace. Other losses, representing 3-4% of thermal energy entering with the waste, would include: thermal losses by radiation and convection (heat loss to surroundings), chemical losses by incomplete combustion and thermal losses in unburned fuel [60]. From those, only the first one (heat loss to surroundings) is considered due to hypotheses viii and xi. Consulting Thermoflex® library for an additional reference, it is found that such "minor heat loss" has a default value of 2.5% for MSW grate-fired boilers. In this sense, it is assumed that the heat loss to surroundings is 2% of the MSW thermal input, which corresponds to a thermal efficiency of the furnace of N_{comb}=98%.

In general, the amount of air supplied in a combustion reaction is either greater or less than the theoretical (or stoichiometric) amount [34]. The amount of air actually supplied is commonly expressed in terms of a percent of theoretical air [34]. For example, 150% of theoretical air means that the air actually supplied is 1.5 times the theoretical amount of air, which is equivalent to increasing the stoichiometric amount in 50%. For waste incineration, the air supply is always greater than the stoichiometric amount, for which the most common expression is *excess air*, in reference to the extra air with respect to the stoichiometric amount. That is, 150% (or 1.5) of theoretical air is equivalent to 50% excess air [34]. A broad literature was consulted before assuming an excess air value (related to the air/fuel ratio), because this is a crucial parameter in the energy analysis. Different ranges of excess air are reported in the literature: 1.2-1.8 in [61], 1.75-1.9 in [5], 1.39-1.4 in [5] and [57], 1.6-2.2 in [13] and 1.8-2.0 in [62]. Where the lowest ranges are possible in well-designed modern plants, particularly with sorting and preparing of the fuel, and higher ranges are required in grate-fired boilers to compensate for irregularities in the fuel and air supply [13]. Consulting Thermoflex® library for an additional reference, it is found that the default excess air for waste combustion in a grate-fired boiler is 60%. Taking into account that the temperature of flue gases of MSW incineration ranges between 800-1450 °C [12], and after tests for an educated guess[2], it is assumed the MSW combustion air/fuel ratio as F_{air}=6.5 (excess air 77%). Moreover, it is assumed $P_{16} = 1$ bar as the air supply pressure at furnace inlet (v. hypothesis ix).

Even though MSW combustion in the WTE boiler is supposed to be modeled as in a grate-fired furnace, significant simplifications are required due to the complexity of the real process. Basically, the proposed simulation consists in: using the above-mentioned assumptions, neglecting pressure loss, considering hypotheses v, xi and xii and applying EES routine “Combustor 2” to calculate mass and energy balances. This accounts for the MSW combustion as if it occurred in a combustion chamber, where fuel and oxidant energies are put together, resulting in products that have dry air composition and a certain temperature. Such temperature can be viewed as similar to the adiabatic flame temperature (not to be confused with T_{15}: temperature of flue gases at boiler outlet), which is useful for determining the boiler temperature profile. As a result, P_{15}, $\dot{m}_{gin}$ (outlet mass flow rate of flue gases + ash), $\dot{m}_{air,inc}$ (inlet air mass flow rate) and the heat rate transfered to combustion products ($\dot{Q}_{gin}$) are determined.

[2]TFX simulation for MSW w/ 10 MJ/kg, e=50% gives F_{air}=5.5.

3.2.2.15
Air pre-heater (15-17)

Consulting Thermoflex® library, it is found that a typical composition of MSW with LHV_{MSW} = 10 MJ/kg produces approximately 20% of ash. In this sense, it is considered that only 80% of MSW combustion products is flue gas that actually enters the air pre-heater (APH), i.e. $\dot{m}_{15} = 0.8\,\dot{m}_{gin}$. Literature data reports that the furnace flue gases outlet temperature (T_{15}) should be cooled to a maximum range of 160-190 °C [12] or even 130-180 °C. Data from Bilbao [17] suggests that flue gases are cooled to 150-200 °C before exiting at stack, for which T_{15} is assumed as 200 °C. Additionally, the average temperature of pre-heated air at furnace inlet (T_{16}) should range between 100-115 °C [12], [38]. For this, the APH approach temperature difference is assumed as 95 K. Then, neglecting the pressure drop, EES routine "Heat Exchanger 2" is used for obtaining P_{16}, h_{16}, P_{17}, h_{17}, from which T_{16} and T_{17} are actually determined, as well as the heat rate transfered to cold fluid ($\dot{Q}_{APH}$).

3.2.3
Exergy analysis

As known, energy and exergy analyses are directly connected (1st and 2nd Laws of Thermodynamics), which means the link between energy↔exergy is obvious. However, the link between exergy↔economic is not always intuitive. In the 4E method, it is proposed to use the exergy efficiencies of the WTE boiler, ST and pump in the economic part of the analysis, as shown in section 3.2.5.1 and in [50]. For this, the procedure described in section 2.1.2 to calculate the exergy destruction/efficiency of the components is applied here, with the only difference that instead of using eqs. 2-7 and 2-8 to obtain the MSW chemical exergy (e^{ch}_{MSW}), it is assumed that $e^{ch}_{MSW} = LHV_{MSW}$. This is explained in the next paragraphs.

One of the most difficult cost estimates regards the MSW boiler. This is because there are not many literature data that can be used as reference, since MSW is a non-conventional fuel. Some researchers point out that the best way to estimate the MSW boiler is to consult manufacturers because existing commercial software may present non-accurate estimates. The other option is to rely on literature data, either by using cost equations or by obtaining references from evaluated plants. The problem with conventional cost equations is that most of them do not regard MSW fuel, thus giving underestimated values. Whereas, the problem with relying on existing data from evaluated plants is that they may present very different characteristics from those one is interested in. An alternative is to make use of commercial software, however, they are not always available and might give wrong estimates as well. Indeed, it is known for a fact that estimates for high-pressure WTE boilers given by Thermoflex® software tend to be inaccurate.

From eq. 2-32, that gives the boiler exergy efficiency expression ($\epsilon_{inc} = \Delta\dot{E}/\dot{E}^{ch}_{MSW}$), it can be noticed that, assuming fix the numerator, the higher the chemical exergy of MSW (e^{ch}_{MSW}) the smaller the boiler exergy efficiency, which results in smaller costs of such equipment (less efficient equipment cost less). An alternative to reduce this problem, i.e., obtain a more conservative cost estimate of the boiler, would be to assume the lowest possible value for e^{ch}_{MSW}, which is e^{ch}_{MSW}=LHV_{MSW}. With this, it would be obtained the highest cost estimate for the boiler without having to modify any of the assumptions made in the other steps of the 4E analysis. This would affect the results in two ways. The first is that the irreversibility estimated for the furnace would be smaller then expected (more efficient equipment have lower exergy loss). However, this effect is reduced partially due to hypothesis v (considers the ash is kept within the CV) because exergy that would otherwise exit with ash outflow is accounted as an internal irreversibility). The second is that the overall exergy efficiency of the plant (ϵ_{tot}) would be higher then expected ($\epsilon_{tot} = \dot{W}_{tot}/(\dot{E}_{MSW} + \dot{E}_{NG})$), in particular ϵ_{tot} would be equivalent to the 1st Law efficiency (η_{tot}). Finally, as mentioned, since the 4E analysis does not establish a methodology to link the exergy↔environmental aspects, no further effect of this measure is expected in the environmental part of the method.

3.2.4 Environmental analysis

The theory of the "energy-ecologic efficiency" (EE) is particularly interesting because combines the results of the energy and environmental analyses in an original manner, linking another two of the four "e"s of the 4E method. In summary, EE is an environmental index that has been significantly applied in the literature to measure the performance of thermal systems in terms of its capacity to control polluting emissions. As described in chapter 5, there are several approaches that can be used to calculate the EE value (or ε). Here, the approach described by Villela in [63] (entitled Approach III in section 5.3.3) and eq. 5-9, reproduced below, are used:

$$\varepsilon = \left(0.204\frac{\eta}{(\eta+\Pi)}\ln\left(135 - \ \Pi\right)\right)^{0.5}$$

In this approach, where ε is called "ecological efficiency", Π is calculated with equations 5-3 and 5-16. In addition, the theory affirms that there is a correlation between ε and the cost of the environmental pollution control system (Z_{EPC}). From literature, it is found that Z_{EPC} of WTE and WTE-GT plants ranges between 6-20% of the plant's cost [64], [16], [65]. However, there is a gap in literature that prevents more precise estimates mainly to WTE-GT plants. This leads to one of the novelties of this work, that is to propose a method to estimate the cost of emission control systems in WTE-GT plants using the EE concept. Hence, one of the objectives of this chapter is to prove the validity of Villela's theory [66] by showing that it is possible to estimate the environmental cost of a WTE-GT system by knowing its EE value and the costs of the other equipment. This is demonstrated in details in section 3.2.5.2. The complete theory of the "energy-ecologic efficiency"is scrutinized in chapter 5. The emission factors used in this chapter are shown [50].

3.2.5 Economic analysis

This section presents the link between energy-exergy and economic analyses, where there is another novelty of the method. Several cost functions based on the thermodynamic parameters are applied to the main equipment to compose the two evaluated financial scenarios. In addition, operation & maintenance/fuel costs and the levelized cost of electricity equations are presented.

3.2.5.1 Cost equations and update

The factor that has the greatest influence on market prices over time is the inflation/deflation [67]. There are several economic indexes that allow the normalization of prices from different times. American Consumer Price Index (CPI) is chosen as the updating index because it is largely used in the literature and its database is easily accessible over a wide time range. The expression to update a general cost from a preceding year Y to 2017 is shown as an example [67]:

$$Cost_{2017} = Cost_{Y} \cdot CPI_{2017}/CPI_{Y} \tag{3-8}$$

where $Cost_{2017}$ and $Cost_{Y}$ are the costs in [US$ of 2017] and [US$ of year Y]; CPI_{2017} and CPI_{Y} are the average CPI values in 2017 and year Y, respectively.

Whenever purchase costs cannot be obtained directly from manufacturers, an alternative is to use *cost functions* or *cost equations*, which are often an expression of the cost as a function of some thermodynamic, energy and/or exergy parameters. Some cost equations available in the consulted literature are applied to the main equipment of the WTE-GT plant (GTS, ST,

Table 3.1: Gas turbine cost equations.

Param.	Equation	Unit	Source
Z_{GTS1}	$450\,\dot{W}_{net,GT}$ [a]	US$ of 2012	[68]
Z_{GTS2}	$300\,\dot{W}^2_{net,GT} + 105900\,\dot{W}_{net,GT} + 6277800$	US$ of 2013	[69]
Z_{cp}	$c_{11}\,\dot{m}_{air1}/(c_{12} - N_{cp})\,P_2/P_1\,\ln(P_2/P_1)$ [b]	US$ of 1994	[70]
Z_{cc}	$c_{21}\,\dot{m}_{air1}/(c_{22} - P_3/P_2) \cdot [1 + \exp(c_{23}\,T_3 - c_{24})]$ [b]	US$ of 1994	[70]
Z_{GT}	$c_{31}\,\dot{m}_{GT}/(c_{32} - N_{GT})\,\ln(P_3/P_4) \cdot [1 + \exp(c_{33}\,T_3 - c_{34})]$ [c]	US$ of 1994	[70]
Z_{GTS3}	$Z_{cp} + Z_{cc} + Z_{GT}$	US$ of 1994	[70]

[a] Aeroderivative 40.000 kW model with 39% efficiency.
[b] c_{11}=39.5 [$/(kg/s)]; c_{12}=0.9 [-]; c_{22}=1 [-] (adapted); c_{21}=25.6 [$/(kg/s)]; c_{23}=0.018 [K^{-1}]; c_{24}=26.4 [-]; N_{cp} is the CP isentropic efficiency.
[c] c_{31}=266.3 [$/(kg/s)]; c_{32}=0.99[-] (adapted); c_{33}=0.036 [K^{-1}]; c_{34}=54.4 [-]; N_{GT} is the GT isentropic efficiency; P in [bar]; $\dot{m}_{GT}$ is the GT exhaust gases mass flow.

HRSG, furnace boiler, pump and condenser) as described in [50], while fans, air pre-heater, deaerator and MSW+ash handling devices are estimated from a commercial software and literature [65].

The following example describes the method for calculating the gas turbine cycle (GTS) cost through functions shown tab. 3.1. It can be observed that three GTS cost estimates (Z_{GTS1}, Z_{GTS2}, Z_{GTS3}) are obtained using the methodologies proposed by Boyce [68], Manesh et al. [69] and Frangopoulos [70]. The first two propose the GTS cost as a function of the system's net power output, while the last one proposes to calculate separately the compressor cost (Z_{cp}), the combustion chamber cost (Z_{cc}) and the gas turbine cost (Z_{GT}). Noticing the years when the cost functions were proposed in tab. 3.1, eq. 3-8 is then used for updating the obtained costs to the desired year. In the results presented in this chapter, all costs refer to 2017. Proceeding analogously with the other components as described in [71] and [50], several purchase costs are determined to every equipment.

From this procedure, it is possible to obtain a *cost range* for each equipment whose lower and upper limits correspond, respectively, to the minimum and maximum values obtained from all cost estimates. That is, in the GTS example suppose the obtained estimates give $Z_{GTS2} < Z_{GTS1} < Z_{GTS3}$, thus, the GTS cost range would be given by the values of Z_{GTS2} in the lower-limit and Z_{GTS3} in the upper-limit: $Z^{min}_{GTS} = Z_{GTS2}$; $Z^{max}_{GTS} = Z_{GTS3}$. Summing all minimum and maximum costs would give the purchase cost range of the plant's thermodynamic equipment (Z_{eq}):

$$\begin{aligned} Z^{min}_{eq} &= Z^{min}_{GTS} + Z^{min}_{ST} + Z^{min}_{pp} + Z^{min}_{inc} + Z^{min}_{UC} + Z^{min}_{HRB} + Z^{min}_{extra} \\ Z^{max}_{eq} &= Z^{max}_{GTS} + Z^{max}_{ST} + Z^{max}_{pp} + Z^{max}_{inc} + Z^{max}_{UC} + Z^{max}_{HRB} + Z^{min}_{extra} \end{aligned} \tag{3-9}$$

where the only unknown subscript is UC, standing for "condensing unit", that includes the condenser and the cooling water auxiliary devices (e.g. tower +

pump); and Z_{extra} include additional equipment/components of the WTE-GT plant such as fans, deaerator, air pre-heater and MSW/ash handling devices.

Cost functions to account for *MSW/ash handling costs* are presented by Consonni [16] and Viganò et al. [65]. They consider the cost of waste feeding and ash handling/material hauling, that is, the devices and infra-structure required for MSW handling and feeding into the furnace (bunker building, crane acquisition, etc.), as well as ash withdraw and haul to an inside deposit. Economic estimates from literature and commercial software often neglected such costs[3]. Attention should be paid to avoid confusing those with the ash final disposal cost (v. section 3.2.5.6) or with emission abatement cost. In fact, equipment/devices for the MSW flue gas cleaning should not be included in Z_{extra} because it is calculated through a particular method, as explained in the following section. In addition, it should be highlighted that each waste processing line should have a maximum capacity of 120 MW_t [5], for which the WTE boiler cost should be multiplied by a factor corresponding to an increase in the number of lines accordingly.

3.2.5.2
Equipment costs including the environmental control

Until now, the preceding sections have shown the link between energy-environmental, energy-exergy and energy-exergy-economic analyses, connecting the first 3 “e's” (energy-exergy-economic) of the 4E method. In this section it is shown the last link, which is between the results of the economic and environmental analyses.

Equation 3-10 determines the total cost of the main equipment of the plant (Z_{equip}) including an additional increment that stands for the EPC cost indirectly obtained through the EE method (to consult how to calculate EE v. chapter 5, section 5.3.3). Since the EE value (ε) ranges between 0 and 1, the theory in [66] states that dividing the cost of the main equipment per ε increases its value proportionally to the cost of the pollution abatement system.

$$Z_{equip} = Z_{eq}/\varepsilon \tag{3-10}$$

where Z_{eq} is obtained through eq. 3-9. Thus, the environmental pollution control cost (Z_{EPC}) can be determined as:

$$Z_{EPC} = Z_{equip} - Z_{eq} \tag{3-11}$$

Combining equations 3-10 and 3-11 it can be derived that the percent corre-

[3]MSW/ash handling device costs are not accounted in [50] due to the lack of reference information at the time.

sponding to the *raw cost of the environmental control system* (without civil work and installation) is:

$$Z_{EPC}/Z_{equip} = 1 - \varepsilon \tag{3-12}$$

3.2.5.3
Total initial investment

The second step of an economic analysis is estimating the total capital to be expended to purchase and install the facility, i.e. the total initial investment or overnight CAPEX (Z_{in}). It includes, besides the equipment costs, direct and indirect expenses such as installation, tubing, instrumentation and control, electrical installation, architecture/engineering, infrastructure/support, supervising, building and unforeseen events [72]. In general, two approaches are often used to estimate the total initial investment: one based on the costs of the main equipment, and other that considers some operational parameters of the project [72]. In this work, the first strategy is used as the following:

- *Total process facilities cost* or total plant cost (Z_{TPC}) is estimated as twice the equipment cost: $Z_{TPC} = 2 \cdot Z_{equip}$
- Contingencies, the balance of the plant (BOP) and engineering (design, testing, insurance and safety) are estimated as 26.5% of Z_{TPC} [16].

Hence, the *total initial investment* (Z_{in}) can be determined as:

$$\begin{aligned} Z_{in} &= 2 \times Z_{equip} \times 1.265 \\ &= Z_{TPC} \cdot 1.265 \\ &= 2.53 \times Z_{equip} \end{aligned} \tag{3-13}$$

where Z_{equip} is obtained from eq. 3-10 corresponding to the raw cost of the equipment. In particular:

$$Z_{in}^{min} = 2.53 \times Z_{equip}^{min} \tag{3-14}$$

$$Z_{in}^{max} = 2.53 \times Z_{equip}^{max} \tag{3-15}$$

From this, it can be concluded that the fraction of the total initial investment corresponding to the *total environmental cost of the plant percent* ($Perc_{EPC}$), including civil work, installation, contingencies & engineering, is:

$$\begin{aligned} Perc_{EPC} &= 2.53 \cdot Z_{EPC}/Z_{in} \\ &= 2.53 \cdot Z_{EPC}/(2.53 \cdot Z_{equip}) \\ &= Z_{EPC}/Z_{equip} \\ &= 1 - \varepsilon \end{aligned} \tag{3-16}$$

For energy conversion from solid fuels, such as WTE plants, it is usual to find in the literature costs normalized per ton of input fuel, which means expressing Z_{in} in US$/ton of MSW. This is useful for comparisons between different WTE facilities. Instead, for electricity power plants investment costs are usually normalized per unit of installed power capacity [22], i.e., Z_{in} expressed in US$/kW$_e$, which aids in further comparisons between different power producing technologies.

3.2.5.4 Operation and maintenance costs

Operation and maintenance (O&M) costs include fixed and variable expenses required to operate and maintain the facility, as well as expenditures with building insurance [22]. In general, O&M costs consist of expenses with personal, material and equipment (including replacement and repairing of permanent equipment) required to normal operation of the plant and transmission grid [22]. The O&M costs that do not change significantly with the energy production are classified as “fix”, whereas those depending on the amount of electricity generated are called “variable” [73]. Fixed costs do not vary with the plant’s availability (operating hours in a year), which may include regular operation charges, general and administrative expenses, preventive scheduled maintenance, among others [73]. Variable O&M costs are directly proportional to the amount of power produced, which is why they are generally expressed in US$/MWh, whereas fixed O&M costs are expressed as a percentage of the initial capital investment (US$). According to Viganò [65], “the fraction of O&M costs proportional to investment cost varies significantly with the energy conversion technology”, ranging 3-5% to waste-to-energy plants and 1.5-2.5% to natural gas-fired combined cycles. In addition, it is mentioned by the author that the fraction of O&M costs proportional to the fuel consumption varies significantly with the type of fuel. The study gives some typical ranges within the European context as: 1-2.5 €/MWh for biomass and 7.5-15 €/MWh for waste.

Since the context of the analysis is the Brazilian reality, the typical values described in [22] are adopted as a reference. Hence, variable O&M cost is considered as $Z_{O\&Mvar,NG} = 6$ US$/MWh for NG, $Z_{O\&Mvar,MSW} = 20$ US$/tonne for MSW and fixed O&M cost is calculated as:

$$Z^{min}_{O\&Mfixed} = 5\%\, Z^{min}_{in} \tag{3-17}$$

$$Z^{max}_{O\&Mfixed} = 5\%\, Z^{max}_{in} \tag{3-18}$$

3.2.5.5 Depreciation

According to Bejan et al. [33] there are many methods for depreciating the value of an asset, where the simplest one is straight-line depreciation. It is used for company purposes and is based on the total depreciable investment at the beginning of the economic life period [33]. Maintenance is usually accounted together with straight-line depreciation as in [74], where maintenance (with depreciation) is assumed as 5% of total investment cost. In [75] an annual depreciation rate of 6% is used for all types of solid waste management. In [76], a 4% straight-line depreciation and a maintenance cost of 1.5-2% of CAPEX during a 25 year lifespan is considered for a WTE plant in Mumbai. Fixed assets are considered to depreciate at a rate of 5% during a 20 year depreciation period, whereas the residual value of fixed assets accounts for 30% of CAPEX in a wind power facility [77]. In [78], a straight-line depreciation rate of 5% (or 12.7% of the total capital investment) is assumed. In this sense, it is considered in this work 4-6% as a possible range for the straight-line depreciation applied to the total intial investment of the WTE-GT plant.

3.2.5.6 Ash final disposal cost

Particularly for MSW incineration plants, one may consider the cost regarding the separation of recyclable materials and residues that should not be combusted (e.g. batteries). In addition, important O&M costs are related to the final disposal of non-combustible materials (e.g. metals, bottom and fly ashes). As the separation of recyclable materials before combustion is outside the scope of this work, it is estimated only the cost of ash disposal. Mass of non-combustible materials corresponds to approximately 25% of the incoming MSW mass. Depending on the hazardousness of the ash, one should consider that it may be disposed on two kinds of landfills. The bottom ash (20% in mass) is considered as non-hazardous material, whereas the flying ash (3.6% in mass) is considered as hazardous, requiring a more expensive final disposal. The non-combusted material (1.7% in mass) is considered as majorly scrap metal donated to recycling units. Based on gate fee values reported for developing countries in [47], [56], [22], [79]; two types of final disposal costs are assumed as reference for Brazilian landfills: US$ 15/ton for non-hazardous and US$ 20/ton for hazardous material (flying ash).

3.2.5.7
Fuel cost

An important cost of thermal power plants fueled by non-renewable combustibles is the fuel cost. For the proposed system only natural gas is considered as a costing fuel because MSW is free. Actually, a negative cost (income) can be assigned for MSW, meaning that the plant gets paid by the Municipality to treat the waste. In the literature it is usually called "gate fee" or "tipping fee". As mentioned in section 3.2.5.6, according to the authors in the consulted literature, a reasonable gate fee of US$ 15 per ton of MSW can be assigned. Even though this is a very low value compared to actual gate fees in the European Union, it is actually representative of the lack of management policies in the present Brazilian scenario and due to the absence of leachate treatment in existing landfills, an expensive procedure. For comparison, according to Wiechers [80], Mexico City presents a gate fee of about US$ 20/ton. Thus, in this work, a value of -US$15/ton (negative because is an income) is considered as the lower limit for MSW cost and zero as the upper limit. The natural gas cost in the Brazilian context of 2015 ranged between 8-12 US$/Mbtu [22] (or 27-41 US$/MWh$_t$), for which it is considered 10 US$/Mbtu (34 US$/MWh) as a reference value [73]. Since in the context of 2019 it is known that NG cost values, in Brazilian Reais (R$), are in the range of R$ 200/MWh (without taxes) or R$ 530/MWh (with taxes)[4], a sensitivity analysis is done in the end of this chapter to take into account the fuel cost variation. However, it should be highlighted that all economic estimates presented in this work do not include taxes.

3.2.5.8
Levelized cost of electricity

It is possible to find different methods for estimating the electricity production cost of energy systems. One that is quite simple and largely used internationally to compare different power generating technologies is the levelized cost of electricity (LCOE) [22]. Tolmasquim [73] describes it as a measure of the net present value of the facility-cost of electricity over the lifetime considered for power generation. Brown et al. [74] defines it as the average ratio of annual capital investments (with depreciation), operations, maintenance and fuel expenditures to the electricity generation over the lifetime of the project. That is, LCOE calculation is based on the concept that the total income of power generation equals to the cost of power generation (both of them in present value terms), i.e. the electricity price for an investor

[4]Source: personal consultation with experts.

Table 3.2: Parameters for calculating LCOE of the WTE-GT plant (reference case).

Total initial investment (US$): Lower range Upper range	 $Z_{in}^{min} = 172$ Mi \$ $Z_{in}^{max} = 267$ Mi \$
Discount rate per year (r)	10%
Lifespan (n)	20 years
Availability	92%
Natural gas cost	10 US$/Mbtu
MSW cost: Lower range Upper range	 -15 US$/ton 0 US$/ton
O&M fixed cost: Lower range Upper range	 5% of Z_{in}^{min} 5% Z_{in}^{max}
Depreciation	Straight line 4%
O&M variable NG O&M variable MSW	6 US$/MWh 20 US$/tonne
O&M (ash final disposal)	1.6 Mi US$/year
Electricity power output (nominal capacity)	107 MW

to precisely "break even" on an investment project [77]. It is a simple metric for comparing different sources using as parameters the costs of investment, fuel, O&M, as well as the useful life, average availability and discount rate [22]. LCOE's expression in US$/MWh is [77]:

$$LCOE = \sum_{t=1}^{n} \frac{\left(I_t + \mathrm{O\&M}_t + F_t + D_t\right)}{(1+r)^t} / \sum_{t=1}^{n} \frac{E_t}{(1+r)^t} \tag{3-19}$$

where r is the annual discount rate; t represents the t year life-cycle of the power plant; E_t is the amount of average electricity production in year t; $(1+r)^t$ indicates the discount factor in year t; I_t is the investment (or depreciated investment if $Dpr > 0$) in year t; $\mathrm{O\&M}_t$ is the operation and maintenance costs in year t; F_t is the fuel cost in year t; D_t stands for decommissioning cost in year t; and n represents the lifespan of the power plant.

The LCOE of the WTE-GT plant is calculated through eq. 3-19 according to parameters in tab. 3.2, considering a three-year construction time with one third of the investment cost applied each year during the construction [17]. Costs for decommissioning are not considered. As mentioned in section 3.2.5.5, a straight-line depreciation is included, i.e., a depreciation rate (Dpr) is considered as a constant percentage of the capital cost throughout the lifespan.

As observed, the upper-limit is a conservative scenario in which the gate fee is null (waste is donated to the plant) whereas the lower-limit considers a positive gate fee (i.e. a negative MSW fuel cost).

3.3 Results and discussion

The main properties of the points depicted in fig.3.2 are shown in tab. 3.3 and the T-s diagram is shown in 3.3. As observed, the re-heating pressure (P_{12}) is chosen as 6.9 bar, as used in the reference [34] (~7 bar) for an ideal reheat cycle. In particular, the Bilbao plant has a re-heat pressure of about 17 bar[5]. The fact that such pressure is higher in Bilbao than in the proposed cycle has the practical advantage of requiring smaller tubes (the lower the steam pressure, the greater the tube volume). However, lower re-heat pressure allows the MSW thermal input share to be greater, as desired, at the expense of a lower energy efficiency. That is, increasing the re-heat pressure would require more gas to be added to the duct burner, increasing the NG share (not desired), which leads to a higher overall energy efficiency (as observed in Bilbao), but at the expense of a higher fuel cost. If the proposed cycle operated with P_{12}=17 bar instead of 6.9 bar, the MSW share would be 42% instead of 57%. An economic analysis to determine an optimum re-heat pressure, balancing the steam turbine cost and fuel cost, is outside the scope of this work. Economic results are discussed in section 3.3.4.

3.3.1 Energy performance

The energy analysis results are shown in tab. 3.4. The *topping cycle*, i.e. gas power cycle, generates 39.5 MW_e of electricity to the grid with a thermal efficiency (η_{GTS}) of about 42%. Such value is close to the international average efficiency of NG thermoelectric sources, which is 45% in LHV basis [5]. NG consumed by the GT + 2nd combustor (or duct burner) in the fired HRSG ($\dot{V}_{NG}$) is about 11700 Nm^3/h.

Regarding the *bottoming cycle*, i.e. steam power cycle, it generates 68 MW_e of net electricity to the grid with thermal efficiency (η_{VC}) of 26% (v. eq. A-26 in appendix A). For comparison, the expected range for single-fueled WTE plants is 22-30%, where efficiencies of 30-60% are achieved only with cogeneration [49]. Bianchi et al. [26] study a small scale single-fueled WTE system (LHV MSW of ~13 MJ/kg) where the appraised efficiency range is 23-25%. The boiler efficiency (η_{inc}), which corresponds to the ratio of energy

[5]Information obtained from on-site visit.

Table 3.3: Parameters of the cycle.

Point	T (C)	P (bar)	h (kJ/kg)	s (kJ/kg.K)	x (-)
1	20.0	1.01	293.4	6.84	
2	584.5	29.5	887.5	6.99	
3	1198.0	28.6	1604	7.62	
4	417.3	1.04	703.5	7.71	
5	603.7	1.04	907.7	7.98	
6	116.5	1.01	390.7	7.13	
7	38.0	0.12	159.1	0.55	
8	39.5	164.00	179.7	0.56	
9	93.3	123.00	400.2	1.22	
9'	314.6	105.00	1429.0	3.39	0.0
10	311.0	100.00	2725.0	5.61	1.0
11	505.8	92.00	3398.0	6.66	
12	183.9	6.90	2808.0	6.81	
13	293.0	6.40	3046.0	7.32	
14	50.1	0.12	2416.0	7.53	0.93
15[a]	200.0	1.01	475.9	7.33	
16[b]	99.3	1.01	373.3	7.09	
17[c]	115.0	1.01	389.3	7.13	

[a] Point 15: flue gas at MSW furnace outlet.
[b] Point 16: air at pre-heater outlet.
[c] Point 17: MSW flue gas at air pre-heater outlet.

acquired by the steam to the thermal energy entering with the waste, is calculate as 83%. This is within the expected range reported in [60] (75-85%) and in [81] (75.4-84.2%), being similar to the average value of the European WTE plants (81%). The parasitic load (or internal consumption) appraised for the overall system is 8% (8.4 MW_e) of the gross power output, which is an intermediate value between those of Zabalgarbi/Bilbao (6%) and Tyseley (10%) plants [82]. Indeed, since such load is calculated based on the MSW input, it is expected that the investigated system presents a greater value than Bilbao's due to the higher MSW capacity of the first. This is an additional advantage of grate-fired furnaces that, besides of being better suited for large scale plants, present smaller parasitic loads than other types of technologies, such as fluidized bed and rotary kiln [82]. WTE plants located in the UK, Allington (fluidized bed) and Newlincs (oscillating kiln), have internal consumptions of 21% and 15%, respectively [82].

Regarding the *overall system*, it produces approximately 98 MW_e of net electricity to the grid ($\dot{W}_{net,ST}$) with 107 MW_e of installed capacity ($\dot{W}_{tot}$); net thermal efficiency (η_{net}) of 35.6% and gross efficiency (η_{tot}) of 39.1%. The total thermal input ($\dot{Q}_{MSW}$+$\dot{Q}_{cc}$+$\dot{Q}_{SF}$) is 274 MW_t, where about 57% of it is derived from MSW (155 MW_t). That is, approximately 43% of total thermal input derives from NG, in agreement with results reported in [83] for hybrid WTE-GT plants, where it was identified that the optimum value for the highest

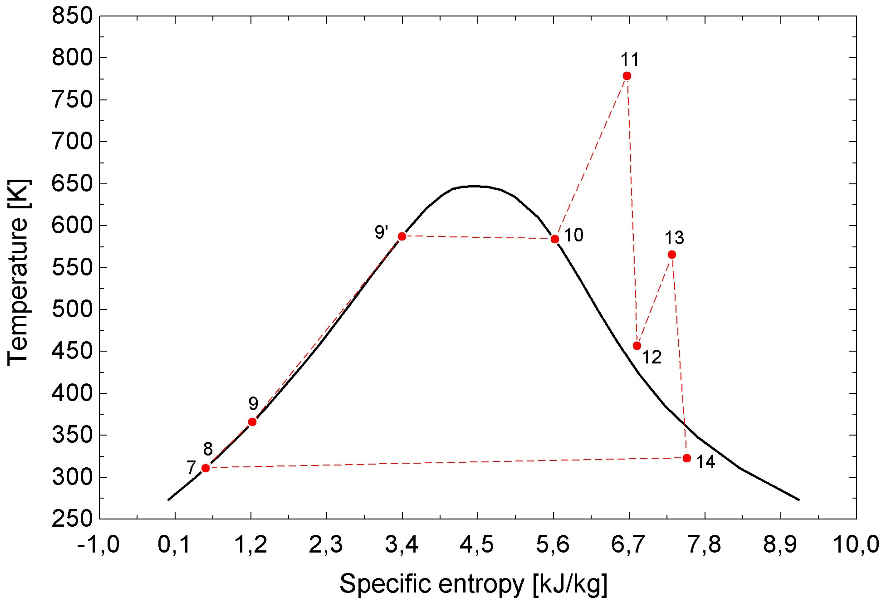


Figure 3.3: T-s diagram of the steam power cycle

energy efficiency is located between 40-50% of NG shared in the total fuel energy input ratio.

Due to the lack of experimental data for dual-fueled WTE facilities with which the obtained results could be validated, subsequent references are presented regarding the energy performance of similar WTE-GT plants. Bianchi et al. [37] find that 1st Law efficiencies range between 35.5% and 42.7% for several WTE-GT layouts, whereas for repowered WTE-GT configurations (v. chapter 4) a range of 32-36% is reported [25]. A more effective validation seems to be possible with Qiu & Hayden's study [40], which presents the energy efficiencies of dual-fuel WTE plants as a function of the NG share. For a system with similar conditions as the investigated design, Qiu finds η_{tot} as approximately 39%, such as the one obtained here. In addition, Ribeiro [6], [28] presents an OCC hybrid design, which is a layout of the type *hot windbox*. It presents η_{tot} of 35% with 77% of thermal input coming from the waste. Even though it has the best performance among all of the above-mentioned cases, according to Branchini [12], the hot windbox configuration has the highest degree of technical complexity of all WTE-GT layouts.

It can be concluded that, besides of the layout type, a key parameter in the energy analysis of dual fuel WTE plants is the MSW/NG ratio; the higher the thermal input from MSW (lower NG input), the lower the thermal efficiency of the plant. That is, Zabalgarbi/Bilbao plant has a power output capacity of 99 MW_e (94 MW_e exported to the grid) with a net thermal

Table 3.4: Results from the energy analysis.

Parameter	Value	Parameter	Value
$\dot{W}_{tot}$	107 MW_e	$\dot{Q}_{MSW}$	155 MW_t
$\dot{W}_{net}$	97.5 MW_e	$\dot{Q}_{cc}$	93 MW_t
$\dot{W}_{net,ST}$	58 MW_e	$\dot{Q}_{SF}$	26 MW_t
$\dot{W}_{net,GT}$	39.5 MW_e	$\dot{V}_{NG}$	11691 Nm^3/h
$\dot{W}_{GT}$	112.4 MW_e	η_{tot}	39.1%
$\dot{W}_{cp}$	73 MW_e	η_{net}	35.6%
$\dot{W}_{STA}$	32.7 MW_e	η_{inc}	83%
$\dot{W}_{STB}$	34.9 MW_e	η_{GTS}	42.4%
$\dot{W}_{pp}$	1 MW_e	η_{VC}	26.4%
Parasitic load	8.4 MW_e	$\dot{Q}_{MSW}/\dot{Q}_{tot}$	56.6%

efficiency of 44% but a MSW thermal input percent of only 33% (71 MW_t capacity). In practical terms, this is an important issue in case the power plant aims to profit from public subsidies for alternative energy sources. According to Murer et al. [57], at the plant of Amsterdam the subsidies are paid only for the produced electricity attributed to the biogenic portion of the waste, and it increases incrementally up to 30% efficiency. This is an example to be followed by hybrid plants in developing countries, whose authorities should implement and review periodically such subsidies. Finally, the advantage of dual fuel plants with respect to single-fueled is proved one more time: the proposed WTE-GT system generates 7 MJ of energy per kg of waste, which is much higher than the values observed for single WTE facilities without additional fuel (2-2.5 MJ/kg).

3.3.1.1
Temperature profiles

Temperature profile is frequently shown as T-Q diagrams such as the ones shown in figs. 3.4 and 3.5. It is observed in the T-Q diagram of the heat recovery boiler (fig. 3.4) that about 58% of the hot gases thermal power is transfered to steam in the superheater, 20% is used for steam reheating and 22% is transferred to liquid water in the economizer. The T-Q diagram of the MSW furnace boiler temperature profile is shown in fig. 3.5. Besides of the assumptions made to simulate the WTE boiler explained in section 3.2.2.14, the major hypothesis that influences directly the results obtained for the gas side is the one that simplifies the composition of the flue gas as standard air (hypothesis iii), which was assumed for simplicity. A more precise method would be to consider the real composition of flue gases. Such simplification

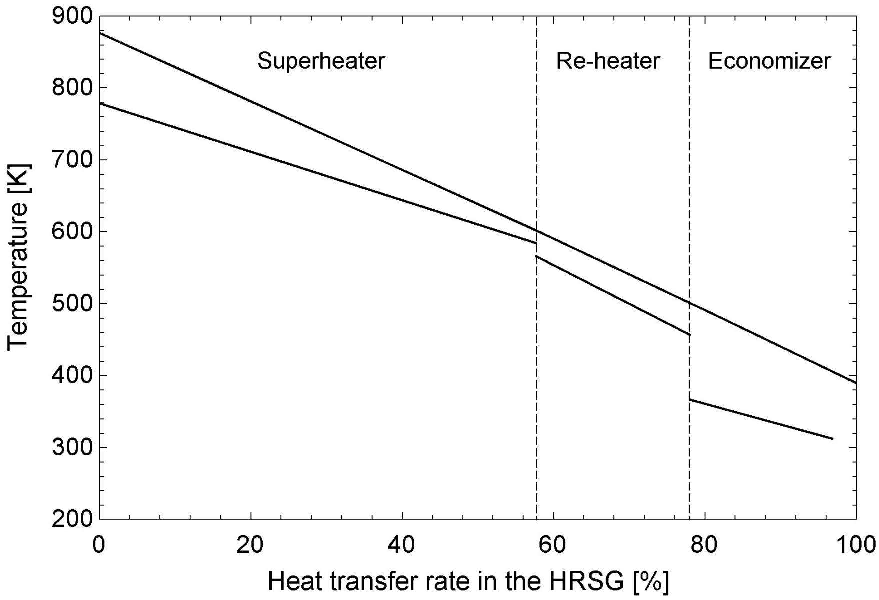


Figure 3.4: Temperatures in the HRSG as a function of heat transfer rate %.

may produce significantly different temperature profiles in the gas side within the WTE boiler, which may influence the operating temperatures of the cold side as well. The effect should be small in the NG combustor, but may be non-negligible in the case of waste combustion. However, for the purpose of demonstrating the method in the present application, this consideration was assumed to have a small impact on the global conclusions discussed in the following sections.

3.3.2 Exergy performance

The calculated irreversibilities are summarized in a Grassman diagram, as shown in fig. 3.6, which consist of a Sankey diagram of exergy flows. As observed, it depicts the useful/useless exergy rates in MW. It can be observed that exergy efficiency of the system, i.e. $\dot{E}_{\text{useful}}/\dot{E}_{\text{in}}$ results in 38.7% instead of 35.6% because the parasitic load is not included.

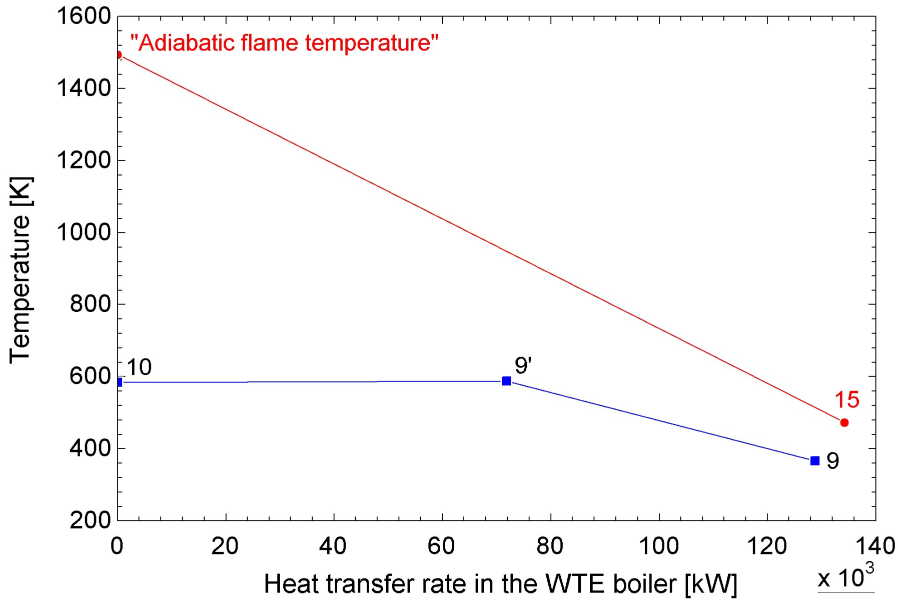


Figure 3.5: Temperatures in the WTE boiler as a function of heat transfer rate.

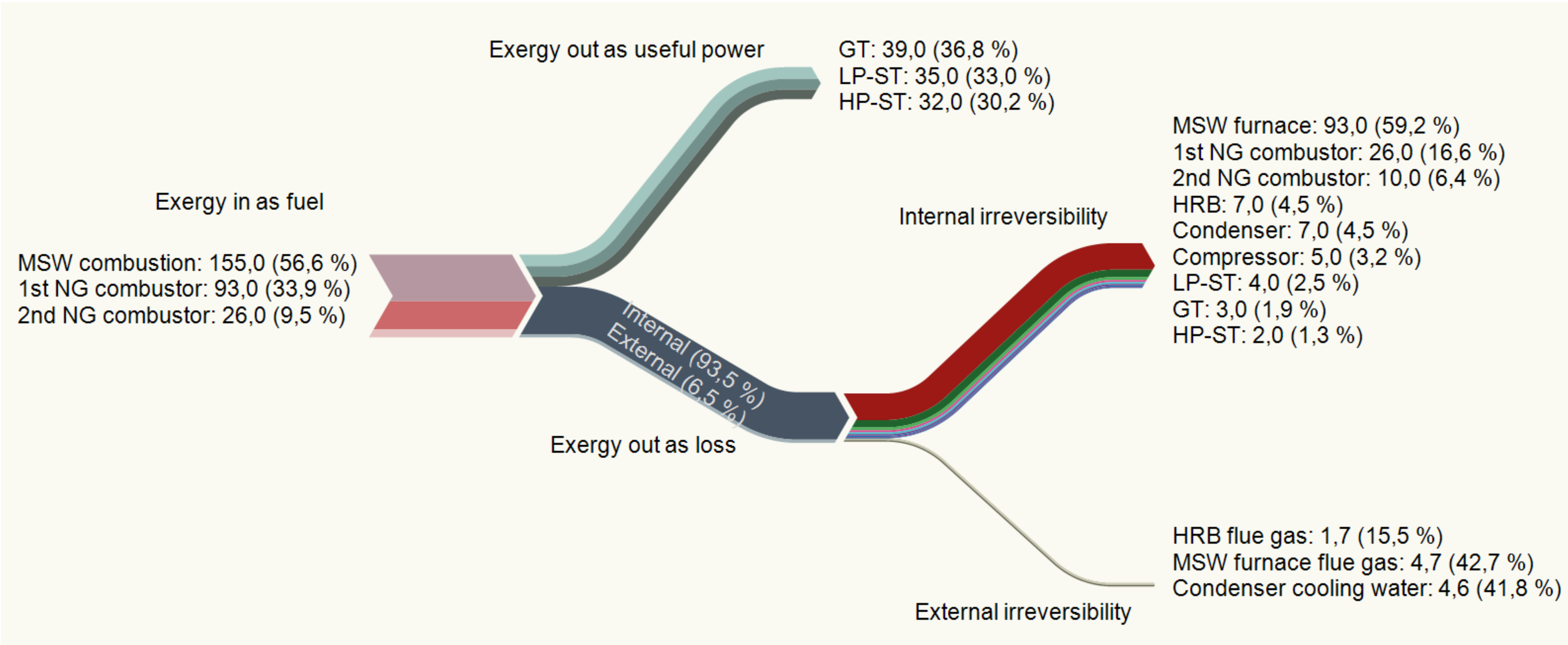


Figure 3.6: Grassmann diagram of the system (exergy rates in MW) showing the exergy losses to the environment (external irreversibility) and within each component (internal irreversibility).

The investigated system has 39% of the exergy transformed in useful electric power (which is equal to the 1st Law efficiency in this case due to assumptions described in section 3.2.3) and 61% is lost through irreversibilities. Most of irreversibilities occur internally (57%), mainly in the combustion equipment, especially the incinerator. This is expected since the majority of losses at thermal systems are due to combustion processes, which are highly irreversible. Rocco et al. [32] find that exergy destructions and losses are higher at grate furnace, steam turbine and super heater. Jadhao et al. [84] affirm that the largest losses occur across the incinerator, which may be attributed to the entropy generation by the highly irreversible combustion process. Brown et al. [74] state that the most important exergy losses are due to chemical reactions/combustion. As a result, exergy efficiencies of the components shown in tab. 3.5 confirm that the lowest exergy performance corresponds indeed to the MSW furnace, 37% (or 17% if using eqs. 2-7 and 2-8). Except for the condenser, the second lowest exergy efficiency is of the HRSG, with 75%. Hence, it can be concluded that improvement efforts could be focused on those.

Table 3.5: Exergy efficiencies of the main components.

ϵ_{STA}	93%	ϵ_{inc}	37[a]%
ϵ_{cc}	85%	ϵ_{STA}	93%
ϵ_{GT}	97%	ϵ_{STB}	91%
ϵ_{HRB}	75%	ϵ_{SF}	78%
ϵ_{pp}	81%	ϵ_{cond}	40%

[a] For $e^{ch}_{MSW} \simeq LHV_{MSW}$, as in [50]. If e^{ch}_{MSW} was obtained w/ eqs. 2-7 & 2-8, $\epsilon_{inc} \simeq$17%.

3.3.3
Environmental performance

As mentioned, the ecological efficiency value (ε) ranges between 0-100%. A system with ε=0% indicates an undesirable situation (maximum pollution) and ε=100% represents an ideal situation (zero pollution). The WTE-GT plant investigated in this chapter has ε=89%, which is similar to that the WTE-GT plant referenced in chapter 5 re-calculated with emission factors from the same sources as this chapter. From this, it is clear that both plants present similar environmental performances, which is explained by the fact that both plants present similar values of the two key parameters that drive the EE result, which are η_{tot} and Π. Namely, η_{tot} $\simeq$39% here and η_{tot}=38% there, and both Π

values here and there are about 0.09 kg/MJ (v. section 5.3, eq. 5-3), calculated with emission factors cited in [50].

As mentioned in details in ch. 5, to the author's knowledge, it is the first time a strategy is proposed to calculate the ecological efficiency of a MSW-fired system, thus, data of WTE plants aiming to validate this result is non-existent. However several studies have used it to evaluate environmental performances of thermal systems. Those adopting Approach III are listed subsequently. Filho et al. [85] calculate EE of sugar cane biomass fired plants. Villela & Silveira [30] evaluate the EE value of NG/Diesel power plants. Coronado [31], [86] obtain EE for biodiesel/Diesel sytems. Santos et al. [87] investigate EE in an NG combined cycle. From the above-metioned references, it is found that EE is estimated as 81% for sugar cane biomass combustion, 91% for Diesel, 92-98% for biodiesel/Diesel and 94-95% for NG. It is plausible that both MSW and sugar cane biomass perform worse than biodiesel/Diesel, Diesel and NG because the last ones are non-solid fuels and do not generate a significant amount of ash as in the case of solid waste/biomass combustion. However, the comparison between MSW and sugar cane biomass is not obvious. From the EE theory, in general, lower LHV and thermal efficiencies lead to higher values of the pollution indicator (Π), which results in a smaller ε (higher pollution). Diesel, biodiesel/Diesel or NG-fired systems have higher efficiencies and greater LHV compared to MSW-NG or biomass-fired plants. The fact that sugar cane biomass has a lower LHV and a lower efficiency than the hybrid WTE plant explains the sugar cane's greater pollution potential (lower ε) compared to hybrid MSW-NG.

Hence, one of the major findings of this study that constitute significant novelty is to provide comparison between the environmental performance of a MSW-NG-fired plant and other thermopower plants based on the EE value. Based on those results and according to the EE method, it can be concluded that a NG or biodiesel-fired systems are considerably more environmentally sound than one fueled by Diesel, sugar cane biomass or MSW-NG. However, it should also be highlighted that the EE concept considers only the atmospheric pollution and does not take into account other environmental burdens such as the fuel's life cycle, contribution to fossil fuel carbon emissions, renewability or land use. Clearly, WTE systems such as MSW and sugar cane biomass present advantage regarding such environmental aspects compared to fossil fuels, which is not reflected by the EE method. Indeed, even though the plastic portion of MSW is actually derived from fossil fuels (non-renewable), in general terms municipal waste can, at least partially, be considered as a renewable fuel as well as agricultural biomass, contrarily to NG or Diesel.

3.3.4 Economic performance

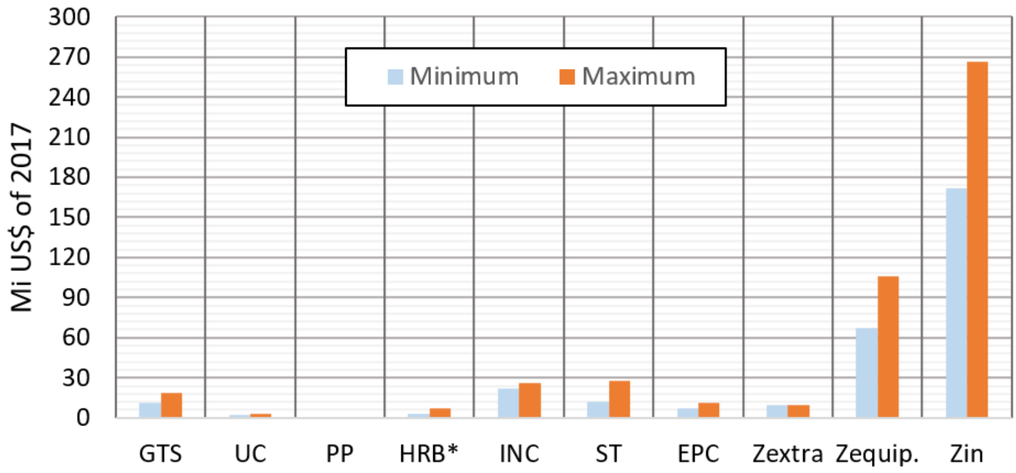


Figure 3.7: Equipment and investment cost range (min. & max.) of the investigated system, where HRB* is the heat recovery steam generator + duct burner.

Figure 3.7 shows the range of equipment costs and the initial investment cost (or overnight CAPEX) of the investigated plant in millions of US$ of 2017. The CAPEX (Z_{in}) ranges between US$ 172-267 Mi (of 2017). In order to validate this result, four procedures were applied to estimate the cost of the plant. The first validation was obtained using the strategy described by Consonni in [16], which appraised Z_{in} around US$ 258 Mi (of 2017). The second validation consisted in using Viganò's method [88] to estimate Z_{in} as US$ 259 Mi (taking the GT cost from Thermoflex® and assuming US$ of 2017). As it can be observed, both values are very similar, indicating that both methods probably use equations from the same source. Thus, a third validation method is used as a tiebreaker, consisting of simulating the exact same system in Thermoflex® and obtaining the cost estimate given by the software. The problem is that the boiler cost appraised by TFX is extremely overestimated, corresponding to 78% of total equipment cost. This is because, as mentioned in section 3.2.3, the WTE boiler is too large, for which it is expected that TFX would give an inaccurate estimate. According to Viganò [65] the cost (acquisition and installation) of a large "grate combustor + boiler (3 lines)" is about 60% of Z_{in}. If we assume the actual WTE boiler cost is about 57% of the value given by TFX to the total equipment cost, the actual Z_{in} appraised using TFX would be US$ 253 Mi (of 2017). Hence, it can be concluded that this is an important finding of this research, proving that the

4E analysis can be used to estimate the total initial investment of the plant quite accurately.

The fourth and last validation of the appraised Z_{in} consisted of using the two plots shown in figs. 3.8 and 3.9. Figure 3.8 shows the investment costs per ton of MSW treated per year as a function of the annual MSW thermal input. Figure 3.9 shows the values of specific investment per electrical output capacity. The specific investments of hybrid plants are represented by red dots while those for single-fueled plants are blue. Comparing the blue trend lines of both figures one can observe that they have similar curvatures. This is because the lower the MSW input or power output, the higher the normalized CAPEX (to the limit if zero input of MSW or zero power output the cost is "infinite"). This is also a consequence of the scale factor, i.e., smaller capacity means greater specific costs. Hence, the more the systems are placed to the bottom-right side of the graphs the better (with a small initial investment it could treat large amounts of waste and produce greater energy).

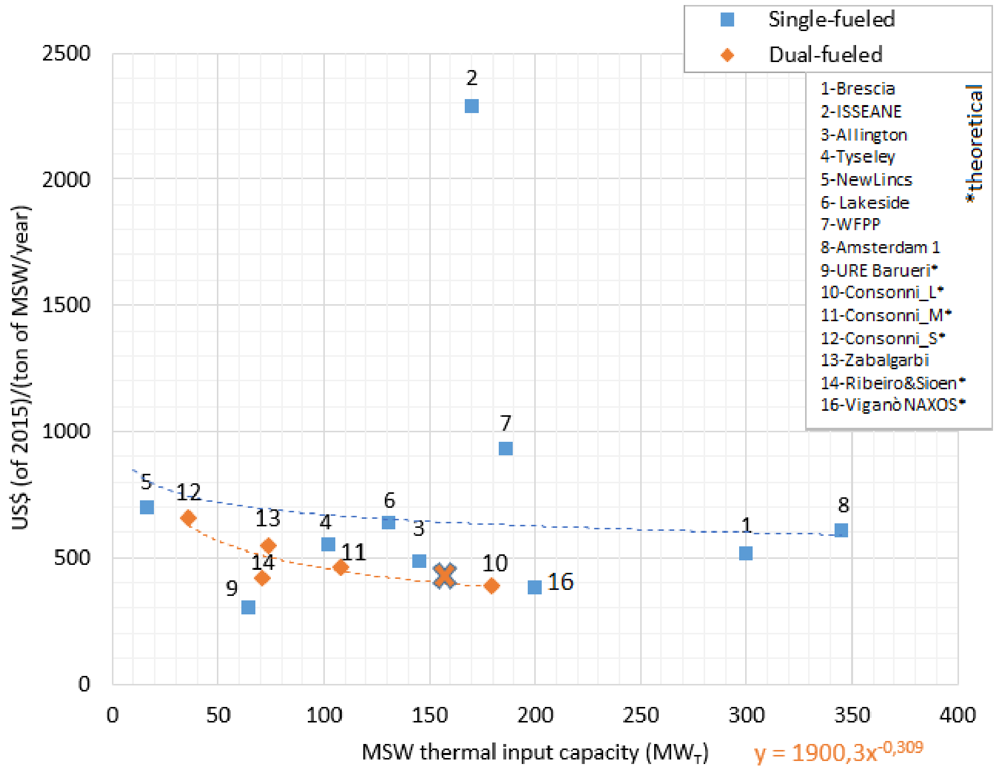


Figure 3.8: WTE plants investment cost per ton of MSW treated per year as a function of annual MSW thermal input in US$ of 2015.

Data obtained from the literature mentioned throughout the text were used to build graphs of figs. 3.8 and 3.9, showing the specific costs of some existing and theoretical European plants and two Brazilian theoretical systems. Technical literature on costs of WTE plants [88] states that graphs of

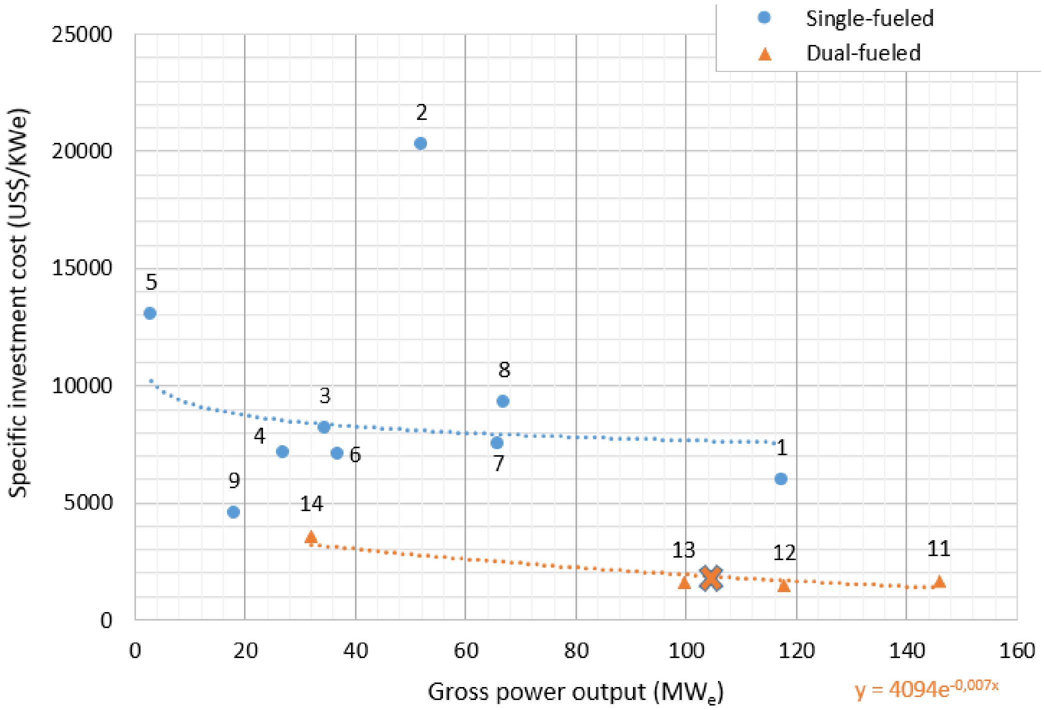


Figure 3.9: Specific investment of WTE plants per installed electrical power output in US$ of 2015.

investment cost per unit electric power ($/MW$_e$) present a much faster decrease with size capacity than investment cost per unit thermal power ($/MW$_t$). This can be observed by comparing the two blue trend lines in figs. 3.8 and 3.9. In this sense, specific costs according to the size of the investigated plant were obtained from the trend equations shown in red at the bottom right of both graphs. In an attempt to follow the technical literature statement, a power function was used to obtain the trend equation in the first graph (fig. 3.8) while an exponential function was used in the second graph (fig. 3.9). From this, it was obtained that the specific costs should be 415 US$ (of 2017) per t/y and 2007 US$ (of 2017) per kW$_e$, while the ranges obtained from the 4E analysis are 352-547 US$/t.y (average 450 US$/t.y) and 1609-2500 US$/kW$_e$ (average 2054 US$/kW$_e$) (averages placed as a cross in both graphs). Moreover, Consonni's [16] estimates for dual fuel plants are 402-480 US$/t.y and 1573-1764 US$/kWe. As noticed, the 4E analysis allows to predict quite accurately the specific investments that should be appraised for a WTE-GT system of such capacity. This is an important outcome because proves that the proposed approach achieves one of its goals that was to provide a new method to estimate the initial investment of hybrid WTE plants. From the distance between the trend lines of single-fueled and dual-fueled WTE plants in the figures, it can be concluded that the choice for a hybrid plant over a single-fueled one might mean, in the case of the investigated system for instance, a specific CAPEX two to five times smaller.

As also mentioned, another goal of the study was to prove the validity of Villela's theory, that proposed a method to estimate the investment on MSW flue gas cleaning system using the EE methodology. From eq. 3-16, it can be observed it corresponds to approximately 11% of the total initial investment. The Confederation of European Waste-to-Energy Plants (CEWEP) [64] reports that existing WTE plants in Norway present this value as 6%. However, evaluating it through Viganò [65] and Consonni's [16] methods gives 10%. Thus, it can be concluded that the proposed method does give in the investigated case a quite good estimate for the cost of EPC with respect to the total cost of the WTE-GT plant.

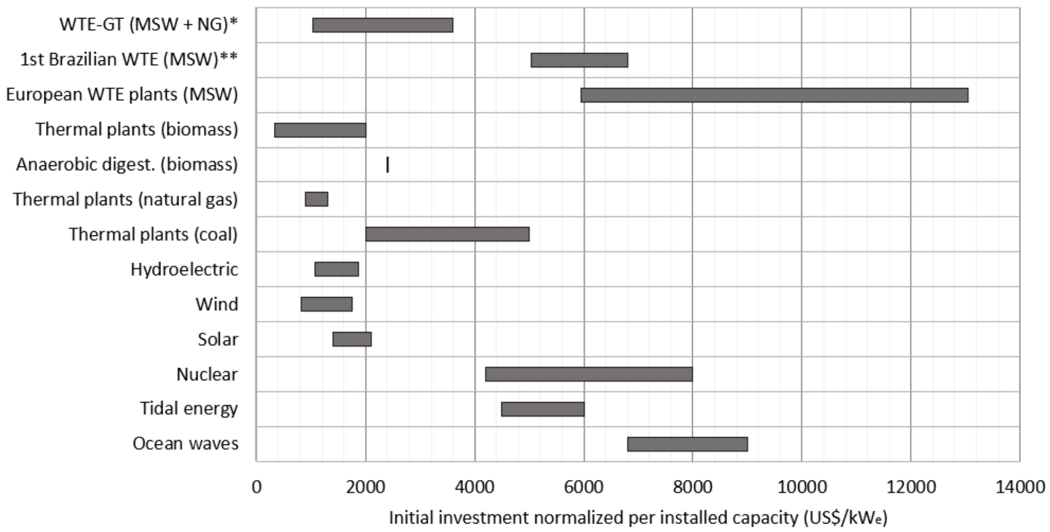


Figure 3.10: Initial investment per installed capacity [US$/kW$_e$] of different sources, where "biomass" stands for wood, sugar cane bagasse or other agriculture waste. *Includes theoretical European/Brazilian plants and the studied plant. **Foreseen 1st Brazilian WTE plant (URE Barueri) to be built in the city of Barueri, Brazil. Wind value considers US$1.00= R$3.20 (Brazilian Reais). Conversion used for the European WTE plants cost update calculations were: €1.00 =US$1.10 and £1.00 =US$1.50. Natural gas thermal plants only consider combined cycle plants.

With additional data obtained from the Brazilian Energy Research Council [73], [22], fig. 3.10 is built showing several initial investment costs normalized per installed electric capacity. It compares different technologies evaluated in the Brazilian context to the aforementioned European WTE facilities. The technology name is followed by the fuel type between parenthesis. The first line shows the theoretical WTE-GT plants and the second line shows the foreseen CAPEX of the WTE plant "URE Barueri" (1st Brazilian WTE plant). Its original CAPEX is given in Brazilian Reais supposedly from 2011 or 2012, and it was updated to US$ of 2015 using the Brazilian index "SELIC tax" both in figs. 3.10 and 3.11. As already observed in fig. 3.9 the hybrid WTE-GT investment is lower than all selected single-fueled WTE plants. Figure 3.10 presents, in addition, two biomass technologies - thermal plants and anaerobic digestion - both are also less expensive than the single-fueled WTE plants. This may be because such biomass fuel (agriculture residues or wood) may have a higher quality than MSW, presenting greater LHV and being more homogeneous. This is also the case of NG-fired plants, whose greater LHV adds to the benefit of the atmospheric pollution control exemption. Indeed, the average investment observed for NG combined cycles is the least expensive of all sources evaluated in fig. 3.10. Additionally, it can be noted that the average investment of hydroelectric, wind and solar only competes with thermal plants of natural gas, lower limit of hybrid WTE-GT plants and upper limit of thermal plants of biomass, being much more attractive than all other sources.

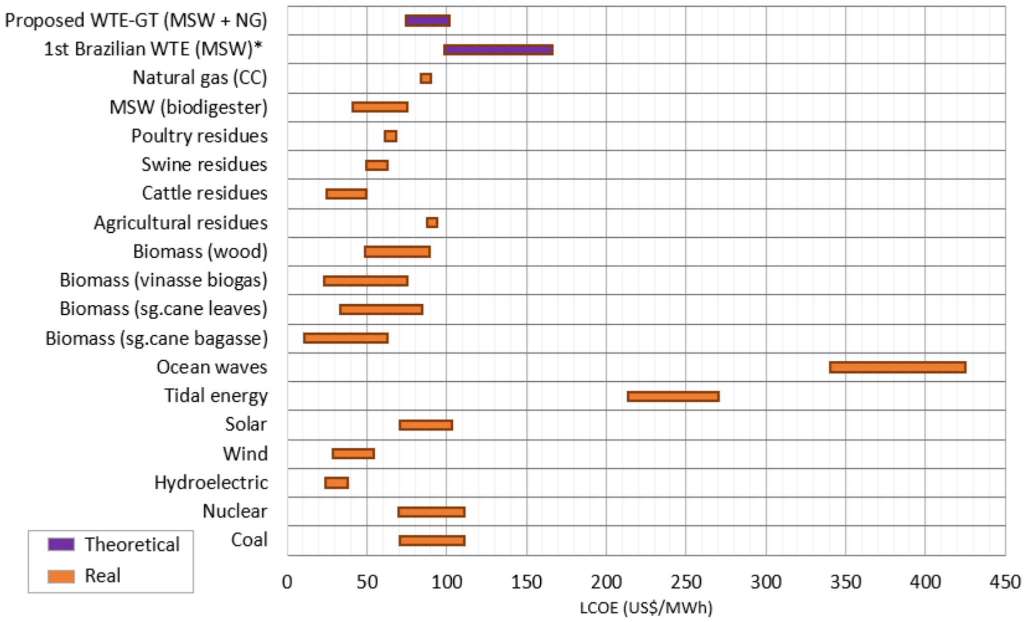


Figure 3.11: Levelized cost of electricity production (LCOE - US$/MWh) of the proposed facility and other power plants in the Brazilian context (own elaboration with data from EPE [73], [22]). The LCOE values calculated for all sources, other than the investigated WTE-GT facility, are expressed in US$ ranging from 2007 to 2015. *Foreseen 1st WTE plant "URE Barueri" to be built in the city of Barueri, Brazil. Decommissioning is only specified for Nuclear as 200-500 US$/kW [73]. Discount rate is considered 10% for biomass and for all other is 8% [73], [22].

Figure 3.11 shows the LCOE of the proposed facility compared to other power plants only in the Brazilian context. Single-fueled European facilities are not included in fig. 3.11 because according to the literature LCOE should be studied in a country-specific basis for more accurate results [77]. This is because LCOE is very dependent on the fuel cost, as also observed in fig. 3.12. In addition, the first Brazilian WTE plant is non-existing, hence its LCOE is only an estimate. It can be observed that the LCOE range of the proposed plant is quite competitive with other sources, such as agricultural residues, wood, solar, nuclear, coal, ocean waves and tidal sources. However, it does not compete with hydroelectric, wind, poultry, swine and cattle residues or MSW processed through biodigestion. Moreover, the average LCOE estimated for the first Brazilian WTE plant is only smaller than the two most expensive sources (tidal and ocean waves), however its lower LCOE range is competitive with solar, nuclear and coal.

In order to have an insight on how the natural gas price and the availability influence the LCOE of the proposed plant different scenarios were evaluated for those variables, as shown in fig. 3.12. It is observed that for the same cost of natural gas, the lower the availability the higher the LCOE range, as expected. In addition, the higher the fuel cost the greater the LCOE, as also expected. Both those behaviors were also observed by Adibhatla & Kaushik [89] in their evaluation of hybrid solar-coal power plants. For an NG cost of 10 or 12 US$/Mbtu, the upper range of LCOE is superior or close to 110 US$/MWh, which makes the proposed facility non-competitive with any bioelectricity or natural gas traditional power plants, but still cheaper than coal, nuclear, solar, tidal and ocean sources. On the other hand, for an NG cost of 8 US$/Mbtu, the upper LCOE is competitive with bioelectricity from agricultural residues or wood only for 95% availability. Figure 3.13 shows how the cost of the waste

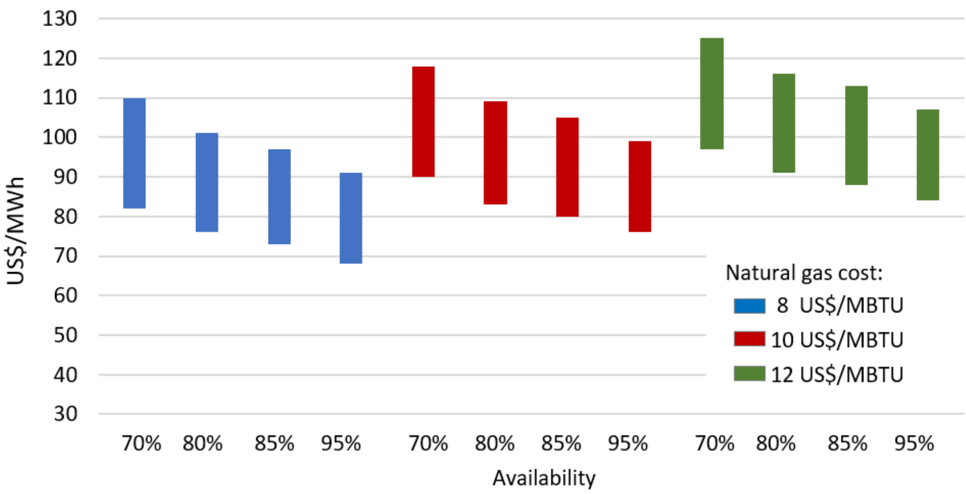


Figure 3.12: LCOE of studied plant for different scenarios of natural gas cost and availability.

influences the LCOE of the proposed plant. As expected, since it represents an income to the facility, the greater the gate fee the lower the $LCOE_{min}$, whereas the $LCOE_{max}$ is not affected by the MSW cost because it is calculated with the assumption of zero gate fee, i.e., as if the waste is received by the facility for free. As observed in fig. 3.14, the LCOE increases similarly with the increase of

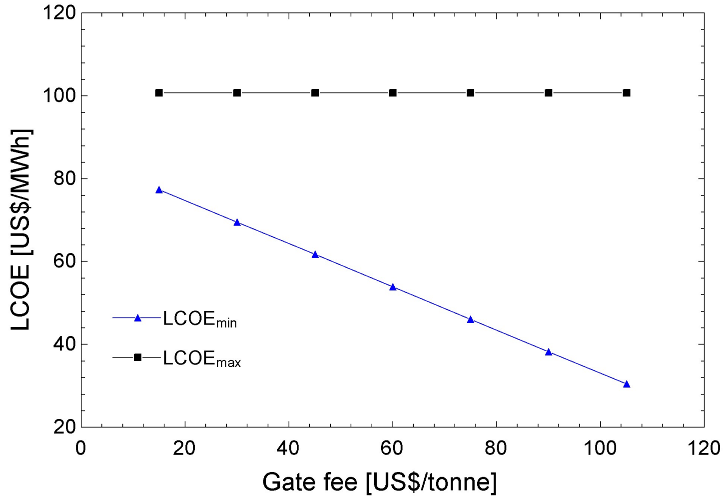


Figure 3.13: LCOE of the studied plant for different scenarios of gate fee.

both depreciation and $O\&M_{fix}$ rate. A depreciation of 8% would increase the LCOE in approximately 28% compared to null depreciation. Whereas, about 13% increase in the LCOE would be expected for a 4% increase in the fix O&M rate.

3.4 Conclusive remarks and future work

This chapter resulted in an article[6] published in *Energy Conversion and Management* journal. It aimed at presenting a new strategy to evaluate the feasibility of WTE-GT systems. To demonstrate the method a proposed design based on the existing plant of Bilbao was evaluated considering theoretical operating conditions. The evaluated system presented a net thermal efficiency of 36%, which is much higher than the average efficiencies achieved by single-fueled WTE facilities in the world. Nevertheless, recent works show that hybrid WTE-GT efficiency range can achieve efficiencies of 35-43% [90], one can thus conclude that there is still potential for improvement. The exergy analysis showed that improvement efforts should be concentrated on the combustion equipment, mainly on the MSW furnace. Therewith, future work should focus on performing variations in the plants' operating conditions and design. The system's ecological efficiency was calculated as 89%. Such value was used to estimate the costs of the pollution abatement devices without

[6]https://doi.org/10.1016/j.enconman.2018.10.007

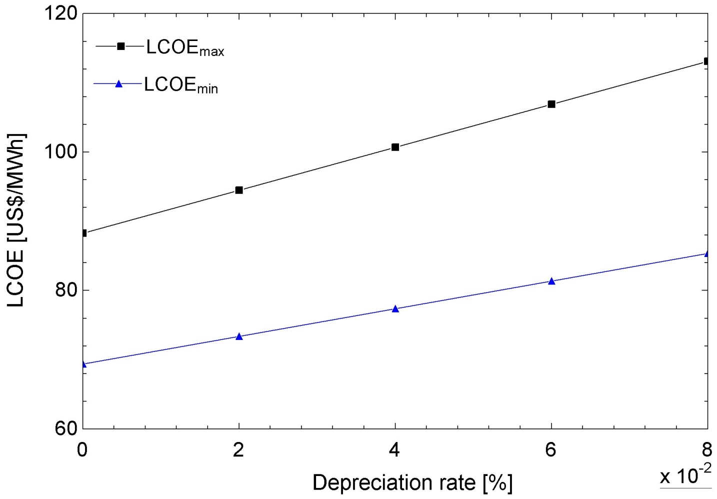


Figure 3.14: LCOE of the proposed facility for different scenarios of depreciation obtained for O&M fix value of 5%.

the need of specifying its characteristics, which characterizes one of the main novelties of the proposed method. It was concluded that such environmental control cost represents around 11% of the plant overnight CAPEX, as also estimated by [65]. However, a more accurate strategy could be used to validate such appraisal, for instance, detailing the EPC devices/chemical inputs and estimating its costs. The method aimed at obtaining an order of magnitude for the CAPEX range through the broadest possible estimate. Such large CAPEX range (172-267 million dollars of 2017) may be explained by the diversity of cost equations employed. Each of which presents its own methodology and assumptions. By using fewer equations (or a single one) for each equipment, the CAPEX range could be refined, if necessary. A narrower CAPEX range could also be obtained by a direct consultation with the equipment manufacturers. The combustion of NG and MSW was simplified assuming standard air. More accurate results could be obtained if the real composition of the flue gas was considered. This may be of relevance in the case of the waste combustion.

A novel comprehensive method was proposed to evaluate the performance of a hybrid WTE-GT plant. It fills the gap observed in the literature where no 4E analysis has been conducted so far for evaluating the feasibility of such technology. As unique findings of this research, it is shown that its specific investment costs are very attractive compared to existing single-fueled waste-to-energy facilities in Europe and other electricity sources in the Brazilian context. To demonstrate the method, a new combined cycle configuration was proposed similarly to the existing plant of Zabalgarbi/Bilbao. It is fueled by NG and MSW, where a GT is combined to a Hirn cycle with a two-pressure

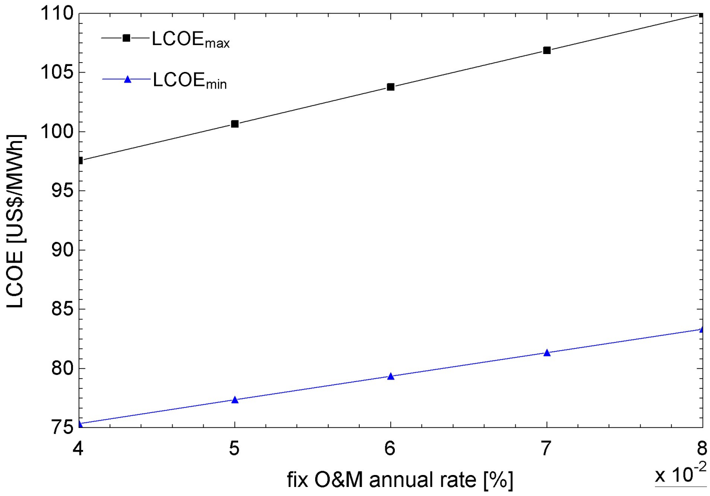


Figure 3.15: LCOE of the proposed facility for different scenarios of fix O&M rates obtained with depreciation factor of 4%.

ST. The evaluated system demonstrated to be more advantageous than single-fueled WTE facilities in the world. In the actual Brazilian scenario, it is also competitive with NG power plants or slightly more expensive than consolidated biomass-to-energy technologies. The method first involved a basic energy assessment to determine the 1st Law efficiencies and additional thermodynamic quantities. Then an exergy analysis was performed for calculating internal and external irreversibilities and exergy efficiencies of the main equipment. Subsequently, a new strategy for evaluating the plant's environmental performance was proposed. It consisted on the calculation of a single-score indicator representing the system's ecological efficiency, also allowing to appraise the cost of the emission abatement equipment without the need to specify its route. In addition, an original economic strategy was developed through the use of several cost equations allowing to obtain the plant's CAPEX range. Finally, a procedure for calculating the levelized cost of electricity (LCOE) was proposed in order to evaluate its competitiveness with other sources. The original local contribution of this work closes the analysis with a comparison between specific investment costs from different energy sources within the actual Brazilian scenario. In conclusion, the proposed method was able to investigate the feasibility of a MSW/NG-fired hybrid power plant in terms of its energy, exergy, environmental and economic performance. Since there are few plants in the world applying such advanced WTE technology, this method can be a useful tool for engineers in the investigation of future designs.

4
Repowered waste-to-energy plants

This chapter intends to apply the 4E analysis (energy-exergy-economic-environmental) methodology presented in ch. 3 to a repowered hybrid waste-to-energy (WTE) plant. To an existing single-fueled WTE system Bianchi et al. [26] proposed a *repowering* option through an integration with a gas turbine in order to increase the plant's efficiency. Because the proposed repowered layout had been designed based on a *real plant* and had only been partially studied (under the view of the 1st Law of Thermodynamics), the objective of this chapter is to expand the original energy analysis to include the exergy, economic and environmental aspects. The system is subjected to a 4E assessment using the *4E analysis* methodology in order to validate the obtained economic costs through comparison with results from other strategies, including a simulation in Thermoflex. Moreover, the *exergy performance* of repowered cycles is evaluated with the method described in chapter 2 aiming to calculate the exergy efficiencies and irreversibilities of a more realistic MSW-fired plant. A sensitivity analysis is performed by investigating different sizes of gas turbines in order to determine the best alternative in terms of energy, exergy, economic and environmental performance. Finally, the economical assessment intends to demonstrate how much could be gained financially if the plant were operating as a dual-fuel WTE plant instead of a single-fueled. This is the first time a 4E feasibility assessment of this kind is applied to a repowered combined cycle integrating a gas turbine to a Rankine cycle fueled by municipal solid waste.

4.1
Introduction

Research on the repowering of underutilized waste-to-energy plants has been increasing in Europe recently due to the rising number of facilities operating with reduced waste input capacity. In opposition to the reality in developing countries, where the waste management is often non compatible with the increasing MSW production, Northern Europe faces a decrease in the amounts of refuse destined to incineration. Figure 4.1 shows the decrease in the processed capacity of a WTE plant in Italy, object of this study, from the start-

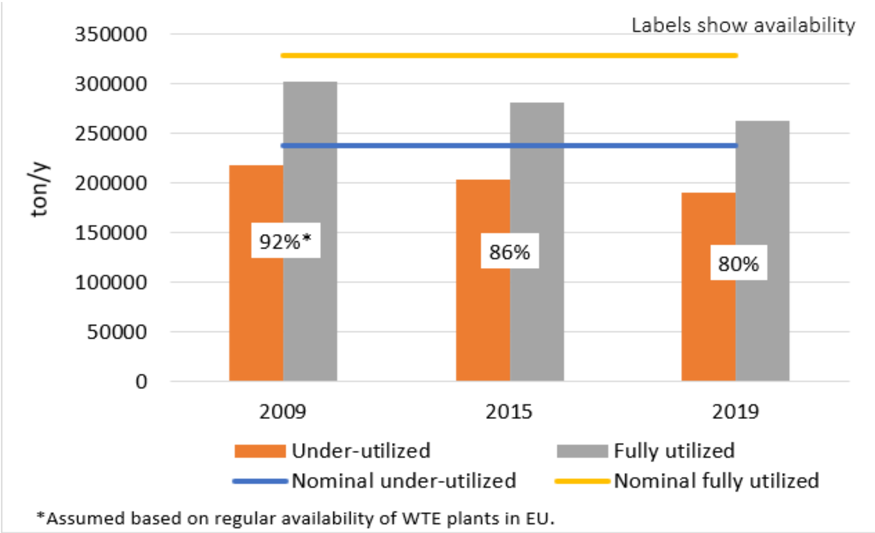


Figure 4.1: Decrease in the MSW treatment capacity in a WTE plant in Italy throughout the years [91, 92].

up year (2008) up to the present time (2019) due to underutilized capacity and availability decrease. The availabilities (percentages) shown in the labels of fig. 4.1 are reported in [91, 92], except for 92%, which is an assumption based on the most frequent availability values reported by WTE plants in the European Union (EU). Availability is the ratio between the annual number of hours the plant operates generating power over the total number of hours in the year. It is never 100% because planned and unplanned stops are required mostly for maintenance, where the main reason for unplanned shutdowns are problems in the boiler, mostly in the radiation section [93]. The nominal capacity instead refers to the maximum capacity, i.e. if the plant operated with an availability of 100% and full load furnace. It can be observed from fig. 4.1 that the plant not only faces a reduced capacity due to an inoperative line but also due to a decrease in its availability. The reason for such reduction in the availability is not known but the possibilities are a lower demand of power from the grid or a decrease in the amount of disposed waste. Indeed, waste management institutions in the EU currently assume that the increase in MSW generation due to expected population growth would be balanced by a successful waste prevention effort [94]. This reinforces the prospects of a likely decrease in the amount of disposed waste in a near future, which would increase the search for repowering alternatives to the underutilized WTE plants. Another issue that arises from current discussions among the EU waste management experts is that the increasing of recycling to comply with the current EU targets may cause the plastic content in MSW to decrease, which might require adding methane or other supplementary fuel to allow MSW burn in WTE plants.

This is a side effect of increasing recycling targets, whose consequences have not yet been clearly studied. Anyhow, the actual trend points towards a context where developed countries will face a stagnation in the number of WTE plants or worse a downsizing in the capacity of the existing ones, such as the example shown in this work. In this sense, this work intends to anticipate providing useful information to engineers, designers and decision makers about the feasibility of *WTE repowering* through integration of a GT to Rankine cycles changing as little as possible the system's original layout.

4.2 State of the art

Developing ways of improving energy efficiency of WTE plants is the main goal of repowering studies such as the one by Bianchi et al. [26], which evaluates thermodynamic options to integrate a gas turbine into an existing middle-sized underutilized WTE configuration. As observed in fig. 4.2, the subject of their study is a WTE-GT plant whose thermodynamic cycle consists of a conventional Rankine cycle with a MSW grate-fired mass burn boiler. The cycle operates in off-design conditions because it was supposed to work with two steam lines (L1 and L2) but only L1 is operating (L2; dotted line). Due to such downsized waste input the system operates with reduced electric power output and the other components face overcapacity [26]. Bianchi' study [26] proposes the integration of a gas turbine (GT) and a heat recovery steam generator (HRSG) into the existing off-design WTE cycle with the concern to add as little modification as possible. Two arrangements (A and B) were studied by the authors, where the GT exhaust gases feed the HRSG. Arrangement A is the one presenting the fewest change to the existing structure compared to arrangement B, which involves a small extra change in the ST because excludes the existing ST bleed. Arrangement A is depicted in fig. 4.3, showing that it consists simply in dividing the deaerator exit flow into two streams: one already addressed to the WTE boiler and the "new stream" is addressed to the HRSG. In the HRSG, GT exhaust gases provide heat to generate additional superheated steam at medium pressure (20 bar, 360°C), which then expands into the steam turbine (ST).

Case B is depicted in fig. 4.4, it holds the same modification of case A but additionally proposes the elimination of the ST bleed, which originally feeds the deaerator, and substitutes it by a low-pressure saturated steam produced in the HRSG. Such arrangement involves some change in the ST operating condition, because eliminating the original bleed would cause an increase in the mass flow rate of low-pressure steam turbine. However, the authors affirm

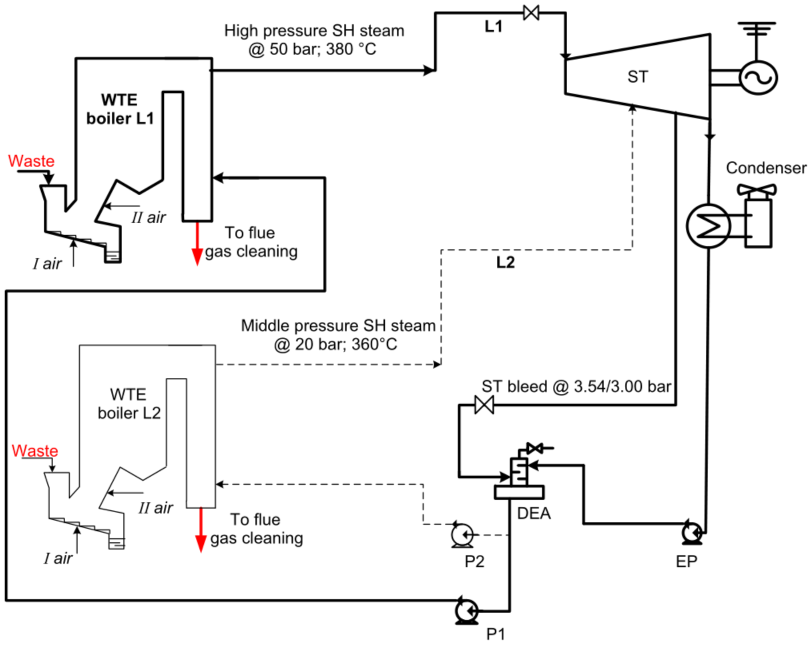


Figure 4.2: Schema of the original (non-repowered) WTE system [26].

that such modification is compatible with the regular ST operation. Both solutions do not require any modification to the WTE boiler or the energy recovery section. Bianchi et al. investigate in what way increasing the gas turbine size causes the amount of heat discharged with the GT flue gases to increase. Their study attempts to obtain an optimal GT integration to the existing WTE cycle by seeking to minimize the GT size to the given steam cycle parameters. Four GT models are tested, two in each option: case A is simulated with KAWASAKI GPB180D (GT1) and General Electric LM2500PH (GT2) and case B is simulated with Siemens SGT-600 (GT3) and General Electric LM2500+PK (GT4). The power output and gross energy efficiencies of such GTs are shown in tab. 4.1.

Table 4.1: Characteristics of the gas turbine machines.

GTS machines	**Gross eff. (%)**	**Power (MW_e)**
GT1: KAWASAKI GPB180D	33.1	17.5
GT2: General Electric LM2500PH	35.0	19.3
GT3- Siemens SGT-600	32.1	21.1
GT4- General Electric LM2500+PK	37.9	27.9

The exact same layout of arrangement A with the GT4 gas turbine is now subjected to a comprehensive energy-exergy-economic-environmental assessment using the 4E method proposed in ch. 3. In addition, all configura-

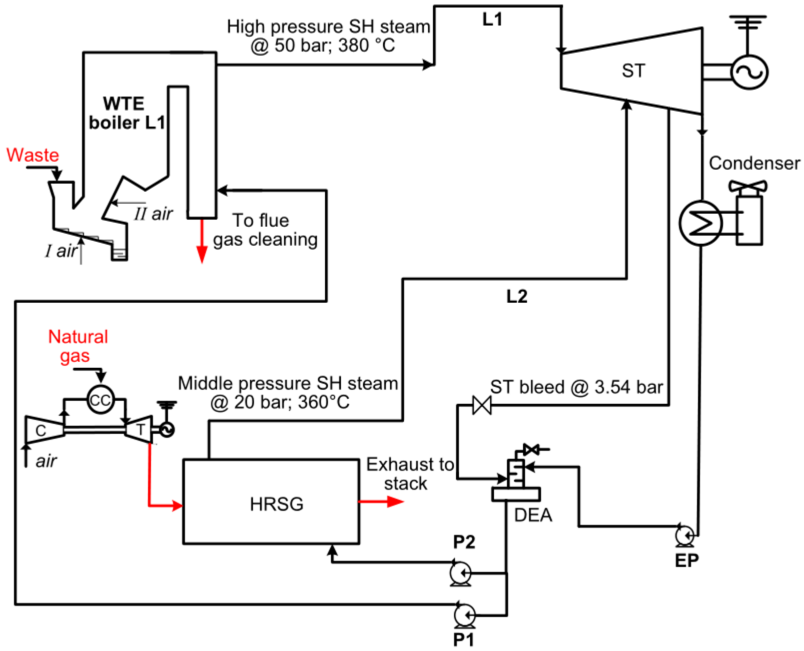


Figure 4.3: Schema of the repowered WTE-GT system (layout A) [26].

tions, i.e. arrangements A and B with the four gas turbines plus the original non-repowered plant designs (*in* and *off* design) are subjected to a complete exergy assessment, using the method proposed in ch. 2. Moreover, economic and environmental indicators, such as the *annual profit* and the *amount of* CO_2 *emitted per unit of power generated*, are also used aiming to confront the different alternatives. Based on the literature review described in section 1.2, which also applies here, no research has been found so far presenting a comprehensive assessement of GT repowering options to existing underutilized WTE plants including exergy, economic and environmental aspects concomitantly. In summary, this study aims to fill this gap by investigating the following cases with the respective methods:

- **GT4** - layout B repowered with 28 MW_e GT (model GE LM2500+PK): 4E analysis and 3E EXC tool.
- **GT1** - layout A repowered with 18 MW_e GT (model KAWASAKI GPB180D): 3E EXC tool.
- **GT2** - layout A repowered with 19 MW_e GT (model GE LM2500PH): 3E EXC tool.
- **GT3** - layout B repowered with 21 MW_e GT (model Siemens SGT-600): 3E EXC tool.
- **WTE-design** - original single fueled WTE plant operating with full capacity (two MSW firing lines): 3E EXC tool.

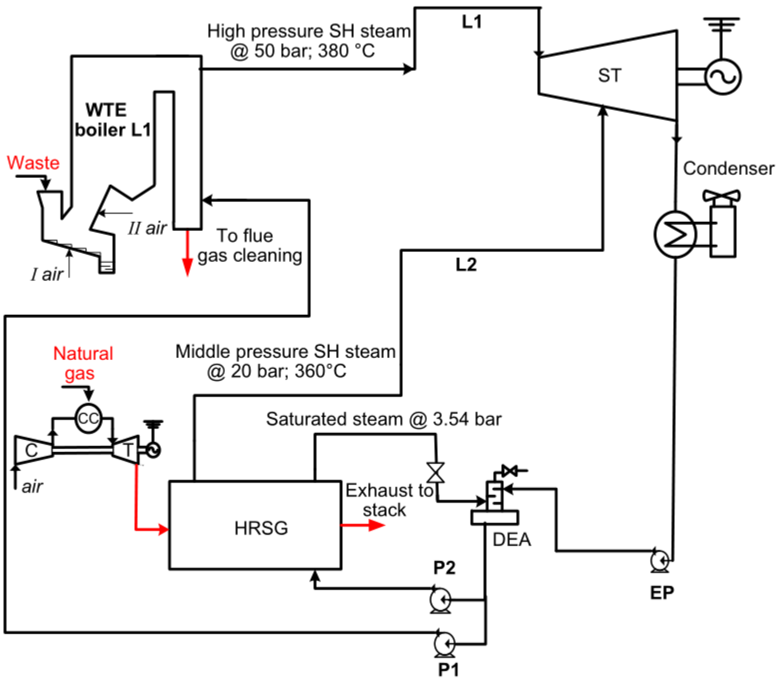


Figure 4.4: Schema of the repowered WTE-GT system (layout B) proposed by [26].

- **WTE-off-design** - original single fueled WTE plant operating with reduced capacity (one MSW firing line): 3E EXC tool.

where 4E analysis is the method described in ch. 3 and 3E EXC tool is the method described in ch. 2 that makes use of a code developed in Microsoft Excel® software.

4.3 Method

The first step consists in simulating the thermodynamic cycles in the commercial software Thermoflex® (TFX), version 28, in order to obtain the main outputs used in the analysis: properties (h, s, T, P, m), power inputs/outputs, 1st Law efficiency, gas mole composition and emissions of SO_2 , CO_2 and dust.

Schemas of the simulated arrangements, A and B, are shown in figs. 4.5 and 4.6 showing in more detail the main components. Even though such figures represent the same cycles modeled in Bianchi' study [26] (reproduced in figs. 4.3 and 4.4), such adapted schemas also show the pressure/temperature conditions at the main points of the cycles and in orange are highlighted the main differences between arrangements A and B. It can be noted that such figures include a feedwater heater (FWH) and an air pre-heater (APH) not

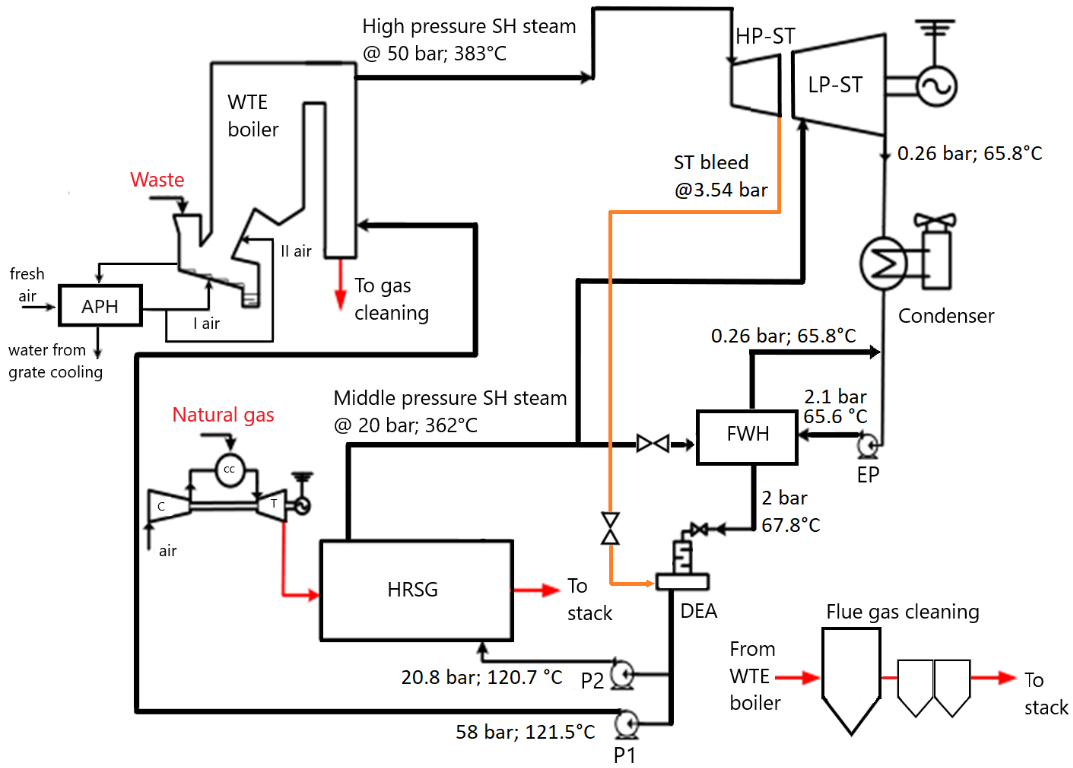


Figure 4.5: Schema of the repowered WTE-GT system (layout A) simulated in Thermoflex® (adapted from [26]).

present in Bianchi' schemas [26] (even though considered by the authors). In particular, the APH rectangle represents two water-gas heat exchangers that have in the hot side water used for cooling the furnace grate and in the cold side primary and secondary air flows. In particular, water at ambient temperature is sent to cool the furnace grate and has its temperature increased by 25 °C, whereas primary and secondary air flows are heated in about 22 °C and 17 °C, respectively. In addition, the flue gas cleaning system is considered to be of the type Dry FGD (v. section 2.1.10).

4.3.1
4E analysis of repowered WTE plants

The GT4 cycle is used as a case study to demonstrate the application of the 4E method to a repowered WTE-GT plant. All steps of the analysis as described in chapter 3 are applied, with the advantage that this time the energy analysis is shortened because many parameters are calculated automatically by Thermoflex® commercial software. Recapitulating, after the characterization of the cycle, the main steps of the 4E analysis are:

1. Energy analysis: obtain the properties at each point, the 1st Law efficien-

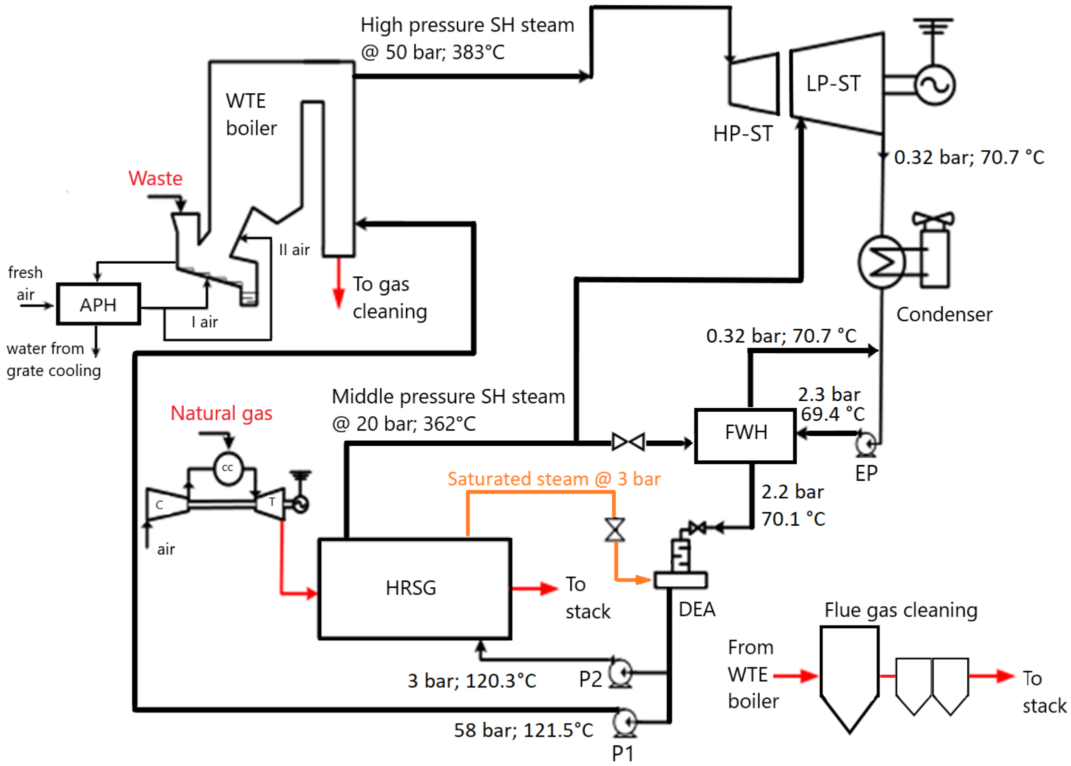


Figure 4.6: Schema of the repowered WTE-GT system (layout B) simulated in Thermoflex® (adapted from [26]).

cies and energy flows;

2. Exergy analysis: obtain the exergy efficiencies;

3. Environmental analysis: obtain the ecological efficiency (EE);

4. Economic analysis, aiming to obtain:

    a. The cost estimates of the main equipment using cost functions and eventual extra costs[1];

    b. The total equipment cost including the environmental control system;

    c. The total initial investment to build and start the operation of the plant (including contingencies, engineering and balance of the plant);

    d. The levelized cost of electricicy production (LCOE).

[1]Extra costs are not obtained w/ cost functions: e.g. dearator, fans, feedwater heater and air pre-heater.

Table 4.2: Energy parameters of the repowered system (GT4 case).

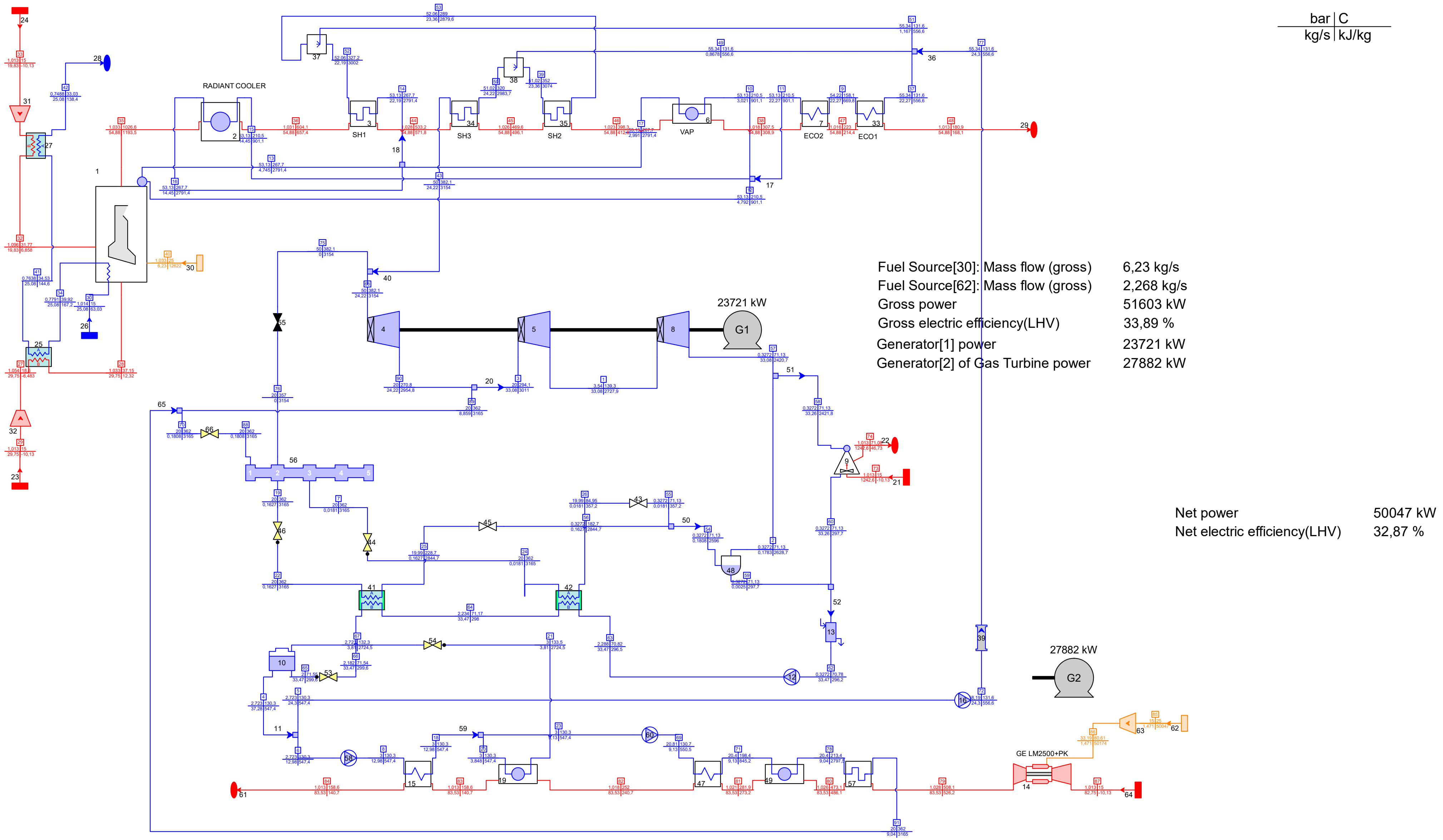
Parameter	Value	Parameter	Value
$\dot{W}_{tot}$	52 MW_e	$\dot{Q}_{MSW}$	78.6 MW_t
$\dot{W}_{net}$	50 MW_e	$\dot{Q}_{NG}$	73.6 MW_t
$\dot{W}_{ST}$	24 MW_e	η_{tot}	33.9%
$\dot{W}_{net,GT}$	28 MW_e	η_{net}	32.9%
MSW capacity	~196500 t/y	$\delta = \dot{Q}_{MSW}/\dot{Q}_{tot}$	51.6%

4.3.1.1 Energy analysis (GT4 case)

As mentioned, GT4 layout is modeled using Thermoflex® software (v. next page, where the flue gas cleaning system is not depicted) based on scheme of fig. 4.6. The bottoming cycle, i.e., the non-repowered plant represents the actual WTE facility of Modena, Italy. Methane (LHV_{CH4}=50 MJ/kg) is fueled to the GTS and MSW (LHV_{MSW}=12.6 MJ/kg) in fed to a grate-fired furnace coupled to a natural circulation boiler that consists of 1 radiative cooler (represents the radiative section of the furnace, v. section 2.1.8), 3 superheaters, 1 evaporator and 2 economizers. In addition, two desuperheaters are placed in the radiative section in order to keep the superheating temperatures controled. The HRSG is composed of 1 superheater, 2 evaporators and 2 economizers. The condenser is air cooled.

The software takes several inputs from the user along with many default variables from its library and allows to calculate mass and energy balances, as well as provides estimates of sizes, costs and some emissions. Almost all properties of the cycle and 1st Law parameters needed in the 4E analysis can be obtained directly from TFX. Variables that are not available from TFX outputs are assumed in the same way as described in section 3.2.2. In particular, since the gas turbine machine is modeled in TFX as a unique component, it is not possible to obtain the values of T/P at the GT combustion chamber outlet directly from TFX, which are necessary to calculate the GT costs using the functions in tab. 3.1. For this, the same procedure described in 3.2.2.2 regarding CC calculations is applied here[2]. The main parameters obtained from the simulation are shown in tab. 4.2.

[2] Here AF_1 and LHV_{NG} are taken as suggested by TFX and CC pressure drop is neglected.


bar | C
kg/s | kJ/kg
RADIANT COOLER
SH1
SH3
SH2
VAP
ECO2
ECO1
23721 kW
G1
Fuel Source[30]: Mass flow (gross) 6,23 kg/s
Fuel Source[62]: Mass flow (gross) 2,268 kg/s
Gross power 51603 kW
Gross electric efficiency(LHV) 33,89 %
Generator[1] power 23721 kW
Generator[2] of Gas Turbine power 27882 kW
Net power 50047 kW
Net electric efficiency(LHV) 32,87 %
27882 kW
G2
GE LM2500+PK

4.3.1.2
Exergy analysis (GT4 case)

Based on the cost functions used in the 4E method described in ch. 3, exergy efficiency values should be provided to at least four components: WTE boiler, HP-ST, LP-ST and pump[3]. Those will then be used in the economic part. Because the version of the software does not provide exergy outputs, the procedure described in ch. 2 is applied, that is, 3E EXC tool is used to obtain ϵ_{inc}, ϵ_{STA}, ϵ_{STB} and ϵ_{pp}. Attention should be paid to exceptionally consider e^{ch}_{MSW}=LHV_{MSW} when calculating ϵ_{inc} for economical purposes, as explained in section 3.2.3, otherwise eq. 2-8 should be used.

As mentioned, the HRSG and WTE boiler are two assemblies composed of several heat exchangers and boilers. Their exergetic efficiencies (ϵ_{HRB} and ϵ_{inc}) can be calculated as:

$$\epsilon_{HRB} = \sum_{cold}(\dot{E}_{outlet} - \dot{E}_{inlet}) / \sum_{hot}(\dot{E}_{inlet} - \dot{E}_{outlet}) \tag{4-1}$$

$$\epsilon_{inc} = \sum_{steam}(\dot{E}_{outlet} - \dot{E}_{inlet}) / \dot{E}_{MSW} \tag{4-2}$$

Particularly, for gas turbine systems there are two possibilities for calculating the exergy efficiency. In chapter 2, eq. 2-22 describes the expression for ϵ_{GTS} in case the exhaust gases are used as a heat source for the combined cycle (GT exhaust gases considered as a useful product). Another possibility is using eq. 4-3 to obtain ϵ_{GTS}, where the exergy of the GT flue gases is not accounted as a useful product.

$$\epsilon_{GTS} = \dot{W}_{GTS} / \dot{E}_{NG} \tag{4-3}$$

4.3.1.3
Environmental analysis (GT4 case)

Approach III of the energy-ecologic efficiency method, as described in section 5.3.3, is used to measure the environmental performance of the GT4 cycle through the calculation of the ecological efficiency (ε). Thermoflex® gives an estimate of CO_2 emissions for the GT combustion, as well as estimates of CO_2, SO_x and PM emissions from MSW combustion. Remembering that approach III assumes only *controlled emissions* (treated) in the calculation of f_{CO2eq}, attention should be paid to measure them at the FGS outlet, i.e., at stack. Since NO_x emissions are not provided by TFX, NG controlled emissions

[3]Ideally, all pumps could have their costs estimated individually through cost functions but, for simplicity, and because those represent a small contribution to the total plant's cost, only the one with the highest power consumption is considered.

are assumed from tab. 5.2 and NO_x emission *at stack* are assumed from the Modena WTE plant (100 mg/Nm^3) [95].

4.3.1.4
Economic analysis (GT4 case)

The same procedure described in section 3.2.5, i.e. using cost functions described in [50], is applied here to estimate the cost of the following equipment: WTE boiler, GTS, ST and pump. Because the condenser is not water cooled, the functions described in [50] do not apply, hence it is taken Z_{UC}=6.8 Mi US$, as suggested by TFX. In this chapter, all costs are updated to US$ of 2017.

The following formulation is used to estimate the costs of the waste and ash handling devices [88]:

$$C_{wh} = [3 \times 10^6 \,(S_w/1600)^{0.65}] \cdot Conv_{Euro-US\$} \tag{4-4}$$

where C_{wh} is the waste handling cost in [US$ of 2017]; $Conv_{Euro-US\$}$=1.1 is the conversion factor from Euro to American Dollars of 2017; S_w is the waste processing capacity in [ton/day].

$$C_{ah} = [2.5 \times 10^6 \,(S_w/350)^{0.7}] \cdot Conv_{Euro-US\$} \tag{4-5}$$

where C_{ah} is the ash handling cost in [US$ of 2017]. Those are included in the extra costs (Z_{extra}) along with the TFX estimated values for the fan, APHs, deaerator and FWHs costs. Then, the same procedure described in section 3.2.5 is applied to obtain first Z_{in} (total initial investment to build and start the plant operation) then the LCOE. Recapitulating, the procedure starts with obtaining Z_{eq}, followed by estimating Z_{EPC} using ε, then obtaining Z_{equip} and at last Z_{in}. The LCOE is obtained through eq. 3-19 using assumptions described in sections 3.2.5.4 to 3.2.5.7. Remembering to calculate two values for all those parameters in order to obtain a range (minimum-maximum). In order to test the ability of the method in estimating the total initial investment of the plant, Z_{in} is also calculated using five other methods: Viganò's [88], Consonni's [16], Thermoflex® 's (estimates of Z_{TPC} by TFX[4] + 26.5%), graph of figs. 3.9 and graph of fig. 3.8.

A cost that is particularly interesting in the case of repowered plants is the cost of the *modification*, i.e., how much it would cost to transform the plant from single-fueled to dual-fueled. This was estimated by calculating the costs

[4]Sum of Costs for Equipment and PEACE Components given by Thermoflex® is considered to be the cost of equipment and installation, which corresponds to Z_{TPC}.

of the repowered plant and the single-fueled plant then diminishing the first of the second.

4.3.2
Comparison between repowered and original WTE-plants

Cases GT1, GT2, GT3, WTE-design, WTE-off-design are also simulated in TFX. Then they are investigated using the 3E EXC tool aiming to confront their exergy, economic and environmental performances. What differs the four repowering cases is the size of the gas turbine and the layout. The influence of the GT size is investigated separately by comparing cases with the same layout, for instance, GT1 and GT2. However, the influence of the layout, which would require to use the same GT in layouts A and B, is outside the scope of this chapter. Layout influence on energy performance of hybrid plants is investigated in a subsequent chapter. Hence, the influence of the GT size + layout is investigated here for instance by comparing GT1 and GT3. Figure 4.7 summarizes the GT efficiencies and power outputs of each case. As observed, GT3 does not perform as well as the other GTs, since its gross electric efficiency is the lowest, even though its power output is the second highest. As it will be shown in the next section, this has a strong impact in the energy performance of the plant, making evident that the choice of the GT machine is of paramount importance.

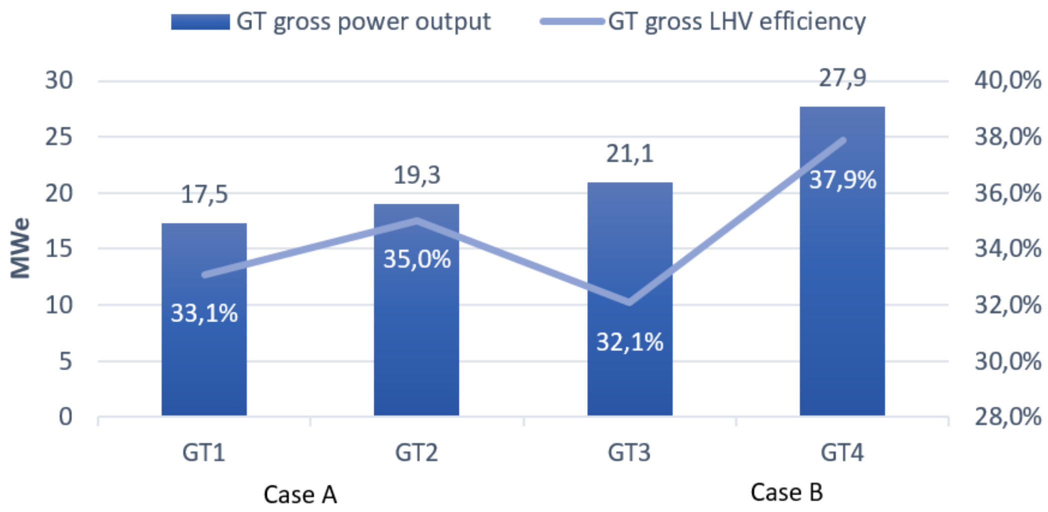


Figure 4.7: Power output and energy efficiencies of the gas turbines used in the repowering layouts.

In parallel, the original (non-repowered plants), i.e., WTE-design and WTE-off-design, are confronted against each other and also against the repowered options in order to answer the following questions:

Table 4.3: Expenses and incomes considered in the yearly cost balance of repowered plants in the actual context of Europe.

Expense/income	Value	Unit
Electricity sale	60	Euro/MWh
MSW gate fee	78	Euro/ton
NG purchase	4	Euro/BTU
Fly ash disposal	200	Euro/ton
Bottom ash disposal	78	Euro/ton
O&M variable	6	Euro/MWh
Operating hours	8000	h/y

– How much profit is being lost by operating the plant under full capacity (off-design) in comparison with the full capacity (on-design)?

– If the repowering options were implemented, what is the annual extra gain that could be expected financially?

For this, a cost balance is calculated assuming each case as installed and operative. The annual profit (A_{sup}) is calculated by diminishing the incomes of the charges. Because the main goal is to compare very similar layouts, it is assumed that differences between fixed costs, amortization and taxes are small. Hence, the cash flows considered in this simplified analysis are only the expenses from natural gas cost, operation and maintenance (O&M) and ash disposal (fly and bottom ash), and the incomes from electricity sale and waste gate fee. A_{sup} is obtained as:

$$A_{sup} = A_{ele} + A_{MSW} - (C_{NG} + C_{O\&Mvar} + C_{desI} + C_{desII}) \tag{4-6}$$

where A_{ele} is the income from electricity sale; A_{MSW} is the gate fee; C_{NG} is the cost of NG; $C_{O\&Mvar}$ is the variable cost of operation and maintenance; C_{desI} is the cost of fly ash disposal in a class I landfill[5] and C_{desII} is the cost of bottom ash disposal in a class II landfill. All of those in US$ of 2017 per year. Because the repowering of WTE plants is currently applicable to the European context, the values assumed for the calculation of A_{sup} reflect the European reality, as shown in Table 4.3.

A simplified environmental analysis is done also aiming to compare the different cases. The mass of carbon emitted by the plant, considering total net power output, is calculated in kg CO_2 per kWh_e.

[5]Brazilian landfills are classified in I and II, depending on the toxicity of the disposed material. Class I is for dangerous refuse and class II is for non-dangerous.

4.4
Results and discussion

The results are divided in three parts. First are presented the results of the energy and exergy comprehensive assessments, followed by the brief analyses showing the estimates of CO_2 emissions and annual profit for all cases. Then, the results from the exergy assessment of GT4 case are shown identifying the main sources of exergy destruction. Finally, the results from the application of the 4E analysis to the GT4 case are presented showing the estimate of the total initial investment compared to those obtained from other methods.

4.4.1
Comparing all cases

In hybrid WTE-plants the key parameter that influences the performance of the plant is the ratio between the MSW thermal input and the total thermal input (or simply MSW thermal input percentage or MSW share), i.e. $\delta = \dot{Q}_{MSW}/(\dot{Q}_{NG} + \dot{Q}_{MSW})$, thus, most results are shown accordingly. Figure 4.8 shows the 1st Law efficiency (η_{net}) of the repowered options placed

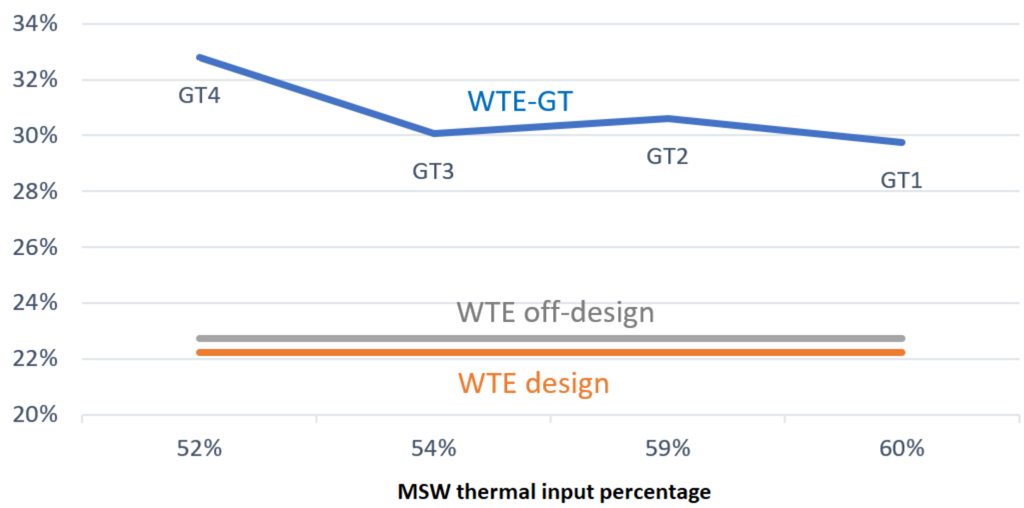


Figure 4.8: Net energy efficiency of the repowered options and single-fueled WTE designs.

according to δ, where the original WTE plants (on-design and off-design) are not to be correlated to x axis (only shown as a reference). From fig. 4.7 it is clear that, in terms of the GT size, the repowered cases are ranked as GT1 < GT2 < GT3 < GT4, which means $\delta_{GT4} < \delta_{GT3} < \delta_{GT2} < \delta_{GT1}$ (axis x in fig. 4.8). Under the influence of only the GT size, i.e. comparing the cases with the same layout (GT1 x GT2 and GT3 x GT4), it can be observed from fig. 4.8 that the greater δ the smaller η_{net}, as expected. However, under the influence of both the layout and the GT size (comparing all cases concomitantly), the result is not predictable. As observed in such figure, GT3 presents a slightly worse energy performance than GT2, even though GT3 has a smaller δ. Another interesting observation is that the off-design condition has a slightly greater η_{net} than the on-design case. This is explained by Bianchi et. al [26]: *"On the contrary, WTE overall plant [off-design] efficiency shows an increase of about 1.6% points compared to design value. The efficiency gain can be explained considering that the power output decrease due to off-design operation is lower than the decrease in waste input capacity due to L2 line shortage".* In fact, results from the energy analysis show that the underutilized (off-design) facility processes 25% less MSW but produces only 23% less power than the original full capacity (on-design). This means that under-utilizing the plant in terms of the overall energy efficiency was good, but as it will be shown later, it translates also into a worse economic performance. Another disadvantage of operating in off-design mode is pointed out by Bianchi et al. [26]: *"It must be outlined that ST off-design operation, due to reduced steam mass flow rate, also causes a decrease in ST isentropic efficiency".* This was also observed in the results of the exergy assessment, where the ST exergy efficiency obtained for the WTE-design is 83.7%, being slightly higher than the value obtained for the WTE-off-design (82.1%).

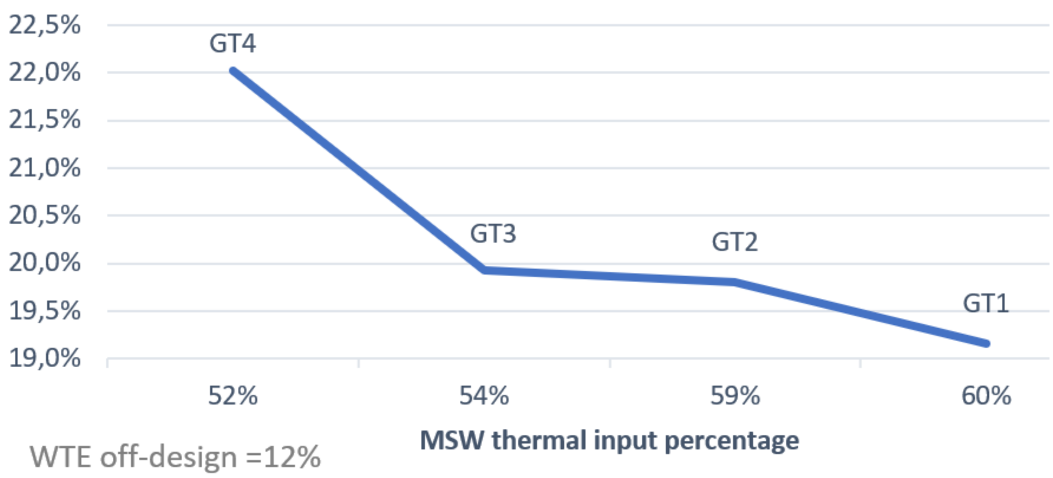


Figure 4.9: Exergy efficiency of the repowered options.

Figure 4.9 shows the overall exergy efficiency of the repowered systems. It can be observed that the higher δ the lower ϵ, showing that the overall exergy efficiency is actually governed by the fuel input ratio, independently of the layout. As in the energy results, GT4 is by far the best option, presenting an exergy efficiency of 22.0% against 19.2% of the worst option GT1, whereas the actual underutilized mode has ϵ=12%. Both GT2 and GT3 cases present similar results, with GT3 being slightly higher than GT2. This contradicts the observed energy results from fig. 4.8, which showed that GT2 has a better energy performance than GT3. The reasons for this become clear by evaluating the performance of equipment individually. Figure 4.10 shows the

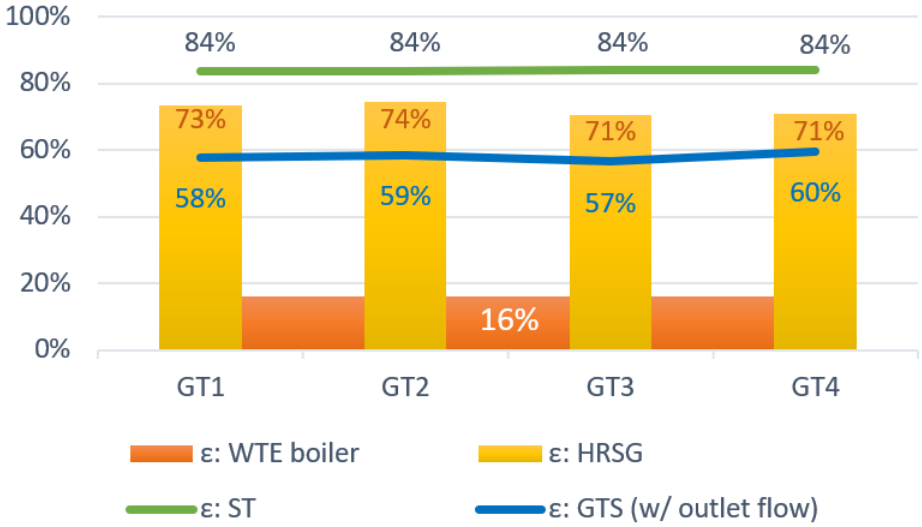


Figure 4.10: Exergy efficiency of the assemblies in the repowered options.

exergy efficiencies of WTE boiler (ϵ_{inc}), HRSG (ϵ_{HRSG}), ST (ϵ_{ST}) and gas turbine system (ϵ_{GTS}) obtained for the repowering cases. As observed, the WTE boiler and ST exergy efficiencies are constant in all cases, with ϵ_{inc}=16%, which is certainly one of the lowest exergy efficiencies among all equipment of the system, and ϵ_{ST}=84% being the highest of all. This is an expected result, since combustion processes are highly irreversible, resulting in lower exergy performances of combustors. The GTS exergy efficiency is calculated according to eq. 2-22, achieving a maximum value of 60% in GT4, which can be explained by the fact that the gas turbine machine used in case 4 has the greatest energy efficiency of all GT machines. On the other hand, GT2 is the case presenting the highest ϵ_{HRSG} of all, approximately 74%. It is interesting to notice that ϵ_{HRSG} obtained for layout A cases (GT1 and GT2) are slightly greater than in layout B cases (GT3 and GT4). This can be explained by the fact that layout B has an extra evaporator in the HRSG compared to layout A. As it will be shown later, evaporators are the heat exchangers with the highest exergy destruction, thus contributing more to decrease the average performance of the HRSG. Hence, the unexpected better energy performance of GT2 observed in fig. 4.8 can be explained by the better exergy performances of the HRSG and GTS compared to GT3. Given those results, one could wonder if GT machine 2 placed in layout B would result in a better alternative than GT3 case. Indeed, Bianchi et al. [26] evaluate this alternative and concludes that NG relative Synergy Index presents the highest value with GT2 machine and layout B. This makes even more evident the importance of choosing a GTS that performs well as a prior criterion to repowering, because a poor choice of the GT machine can "ruin" a potentially good repowering layout.

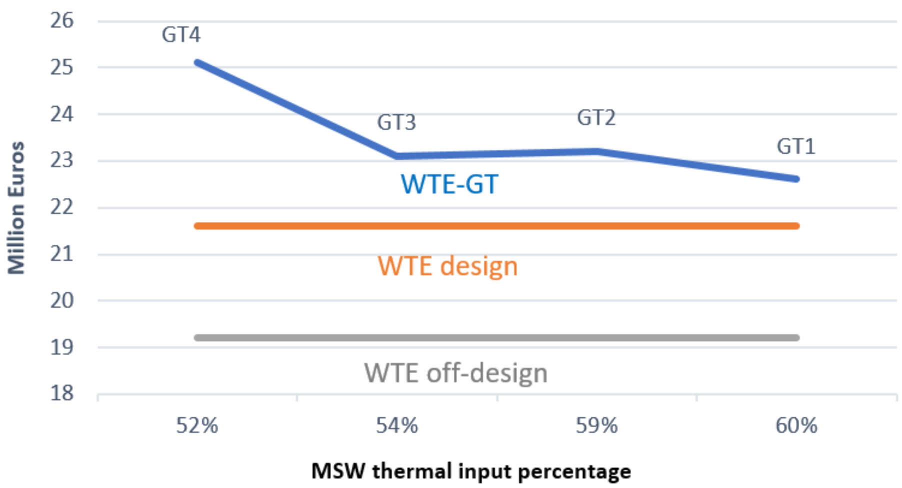


Figure 4.11: Annual profit of the repowered options and single-fueled WTE designs.

Figure 4.11 shows the estimated annual profit that could be gained if the plants were operating with and without repowering. It can be observed that the WTE-on-design plant presents a higher potential profit than the WTE-off-design, which is expected, meaning that running the plant below capacity is not interesting economically. It is also shown that all repowering options present higher potential profit than both original and underutilized WTE plant. This demonstrates that if the facility had been built with GT4 configuration instead of the planned single-fueled WTE design, an additional annual income of about 3.5 million Euros could be earned. Moreover, if compared with the actual underutilized plant, the extra income would be of 5.9 million Euros per year. This means that the choice of building a single-fueled WTE plant and operating it with a 25% smaller capacity (less 59 thousand tons/year) instead of operating a dual-fueled GT4 option results in an annual "loss" of almost € 6 million. Obviously, this estimate applies only after the investment required for building the repowered configuration has been paid back, which was estimated as 21 million dollars (US$ of September 2018). This is equivalent to 18% of the total plant's cost (Z_{TPC}) before repowering.

Figure 4.12 shows the results from the environmental assessment in terms of emission of carbon dioxide. It can be observed that the repowered options

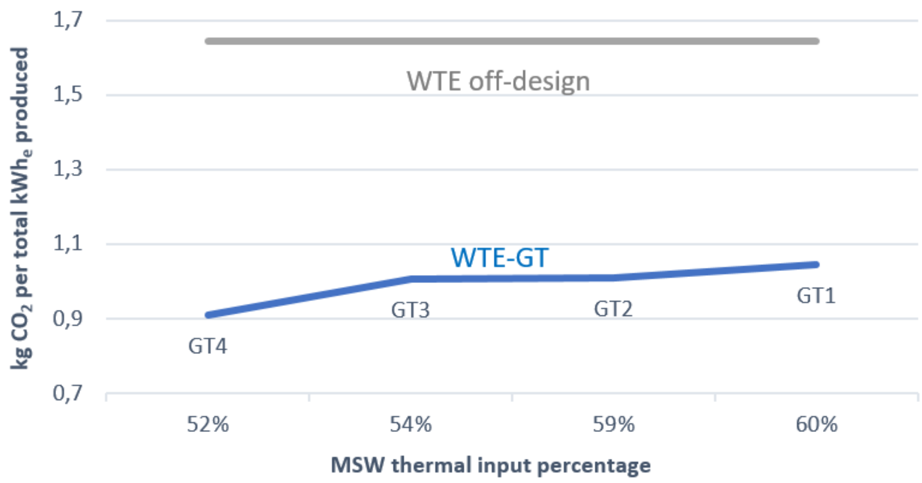


Figure 4.12: Emissions of carbon dioxide per electrical power output produced in the repowered options and under utilized WTE plant.

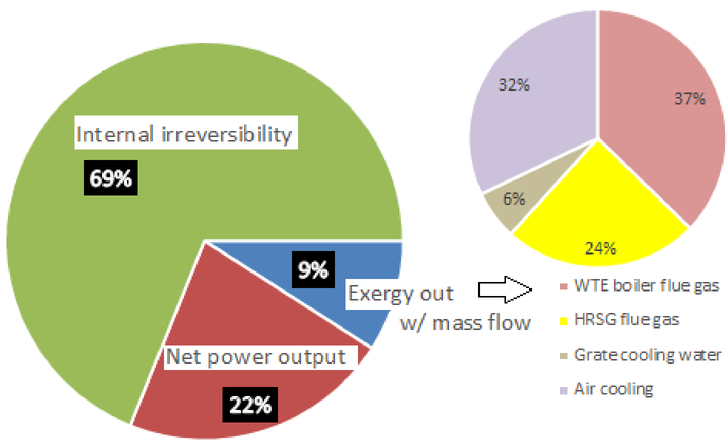


Figure 4.13: Overall exergy balance of the GT4 case detailing exergy out with mass flows.

present smaller emissions compared to the single-fueled WTE plant, since the higher the δ, the greater the CO_2 specific emission. This is expected since natural gas has a lower CO_2 emission potential than MSW and a higher calorific value. One more time GT4 arrangement presents the best performance, i.e. the lowest emissions of all. GT3 and GT2 present very similar environmental performances, again indicating that GT3 is not a good alternative, since having a much lower δ than GT2 one can expect GT3 to have significantly smaller CO_2 specific emissions than GT2.

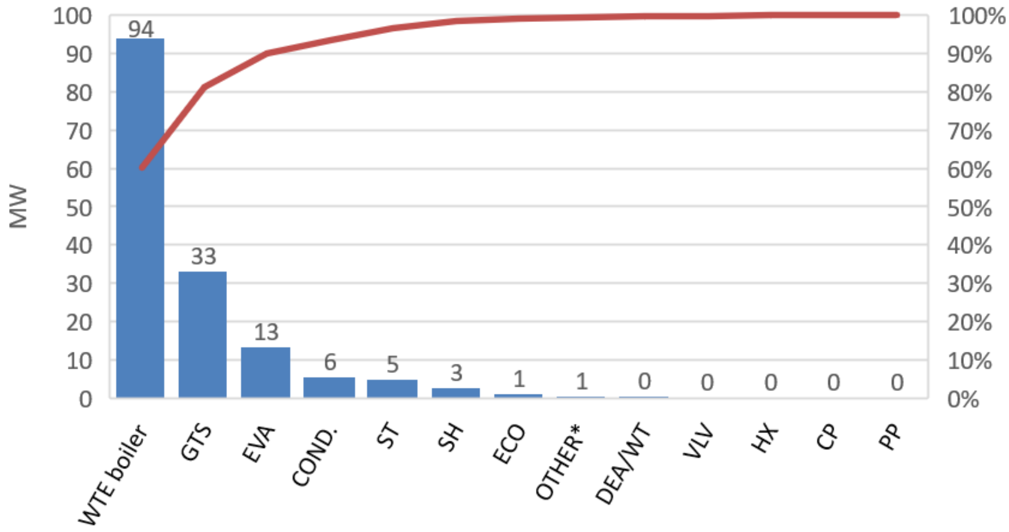


Figure 4.14: Distribution of internal irreversibilities in the repowered plant GT4. *Other: splitter, mixer, fan, pipe, makeup/blowdown.

4.4.2
Exergy results - GT4 case

Without a doubt, GT4 has shown to be the best repowering option in terms of energy, exergy, environmental and economic aspects. Investigating the GT4 case deeper in order to quantify all exergy losses and efficiencies of individual components could help improve the system's performance even more in the future. The results from applying eq. 2-18 to the overall system is shown in fig. 4.13. It can be observed that 69% of exergy is loss due to internal irreversibilities ($\dot{I}$), 22% is electricity exported to the grid and 9% of exergy leaves the system with exiting mass flows, from which 37% exits with the WTE boiler flue gases and 32% with the condenser air cooling. As shown, this means that the exergy efficiency of the GT4 system (ϵ_{tot}) is 22%. A pareto graph shows in fig. 4.14 the distribution of exergy losses due to irreversibilities within the system. It can be observed that more than 80% of internal irreversibities occur in the combustion devices, with 60% occurring in the WTE boiler and 21% in the gas turbine system. After those, exergy is mostly destroyed in evaporators, accounting for 9% of all internal irreversibilities. For comparison, Behzadi [59] finds that "biomass burner contributes to 55% of the total exergy destruction". Behzadi's value is smaller than the one obtained here due to the fact that the author takes into account the ash outflow from the MSW combustor, which is neglected here as per hypothesis v in ch. 2. Since the exergy of ash outflow is being accounted here together with $\dot{I}_{inc}$, it is expected that our result to be greater than that of Behzadi.

Figure 4.15 shows GT4 exergy efficiencies of each equipment individually compared to the average exergy efficiency of its class, where are only shown equipment with more than one exemplar in the class. This information is useful to identify potential for improvement. It can be observed that exergy efficiencies of economizer 47 (ECO-47) is below the average of economizers in the system, meaning it could probably be improved. Similarly, EVA-2 is below the average of evaporators, PP-58 is below the average of pumps, SH-3 and SH-57 are below the average of superheaters. APH-25 and APH-27 stand for air pre-heaters of primary (underfire) and secondary (overfire) air streams, respectively, while FWH-41 and FHW-42 are the two feedwater heaters. The small values of ϵ observed for APHs, FHWs and also for the condenser (ϵ_{cond}=41%, not shown in the figure) are explained by the fact that they operate close to the reference temperature, i.e. $T_u \simeq T_0$, which as per eq. 2-19 causes ϵ to be small.

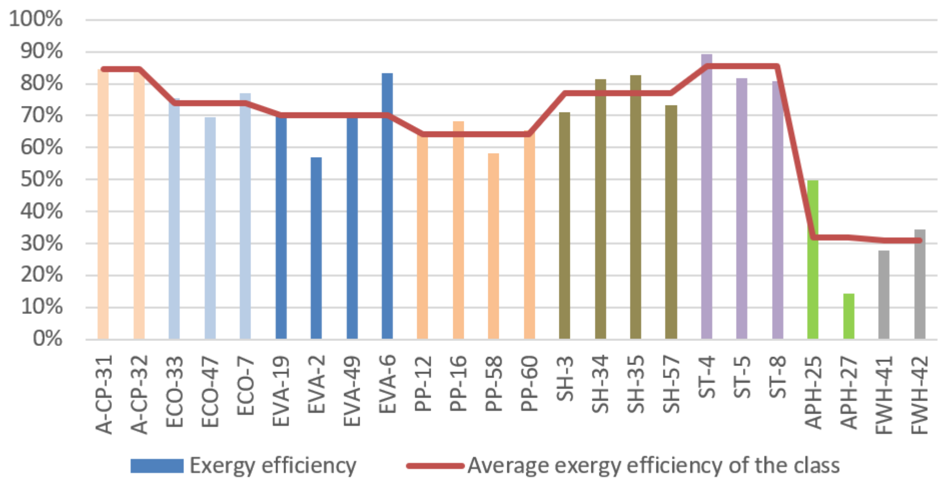


Figure 4.15: Exergy efficiencies in the repowered GT4 layout showing each equipment compared to the average (red plot) of its class. Every class of equipment has a different color and are only shown the classes with more than one exemplar. The class acronym is followed by the equipment identification number, e.g. A-CP stands for air-compressor, which has 2 exemplars 31 & 32.

4.4.3
4E analysis of the GT4 case

As mentioned, the method explained in ch. 3 was applied this time to case GT4. The main goals of the 4E analysis are to determine the total initial investment and the LCOE in the Brazilian context. The results from the 4E analysis not shown yet are:

– Exergy efficiency of the WTE boiler calculated as $\epsilon_{inc} = \Delta\dot{E}/LHV_{MSW}$ for economic purposes, as explained in 3.2.3, is about 29%.

– The ecological efficiency of the GT4 repowered case is $\varepsilon \simeq$89%. This means, as per eqs. 3-11 and 3-10, that the environmental cost is about 11% (1-0.89) of the total initial investment (Z_{in}), which is within the expected range for WTE plants (6-13%).
– The total initial investment range is estimated between $Z_{in}^{min} \simeq$90 Mi and $Z_{in}^{max} \simeq$170 million Dollars of 2017.
– The range estimated for the LCOE in the Brazilian context and according to assumptions in tab. 3.2 is between $LCOE_{min} \simeq$81 and $LCOE_{max} \simeq$119 US$ of 2017 per MWh (average US$ 97/MWh), with $\mathrm{LCOE_{max}}$ assuming the worst possible scenario as if the waste is treated for free (null gate fee).

In order to validate the method, Z_{in} is also estimated using different methods and the results are:

- Thermoflex's estimate of Z_{TPC} is 135 Mi US$ of 2017, as per eq. 3-13, it gives $Z_{in} \simeq$171 Mi US$.
- Consonni's estimate is also $Z_{in} \simeq$171 Mi US$ of 2017.
- Viganò's estimate is $Z_{in} \simeq$173 Mi US$ of 2017.
- Graph shown in fig. 3.9 for hybrid WTE plants gives an estimate of about 2960 US$/$\mathrm{kW_e}$, which is equivalent to $Z_{in} \simeq$153 Mi US$ of 2017.
- Graph shown in fig. 3.8 gives an estimate of 536 US$/ton/y, which is equivalent to $Z_{in} \simeq$102 Mi US$ of 2017.

As observed, the above-mentioned methods appraise 102< Z_{in}< 173 Mi US$, which is very similar to the range obtained with the 4E analysis (90 <Z_{in}< 170 Mi US$), showing that the method is reliable also for small scale WTE-GT plants, such as the GT4 case.

Performing a sensitive analysis with the assumptions from tab. 3.2, figure 4.16 shows how the maximum LCOE is influenced by the gate fee, where LCOE is calculated with $Z_{in} = Z_{in}^{max}$ and assuming this time a non null gate fee. As observed, LCOE decreases with the gate fee, as expected, because a greater income payed by the Municipality to treat the waste would cause the electricity production cost to decrease. An increase of about 6 times in the gate fee causes the LCOE to decrease by 32%. Another important variable is the natural gas cost, whose influence on the LCOE is shown in fig. 4.17. As observed, LCOE increases with NG cost, as expected, because a greater expense causes the electricity production cost to increase as well. An increase of about 1/3 in the NG cost causes the LCOE to increase by 36%. Hence, it can be concluded that, for the GT4 repowering case, the NG cost has a much more important influence on the electricity price than the gate fee.

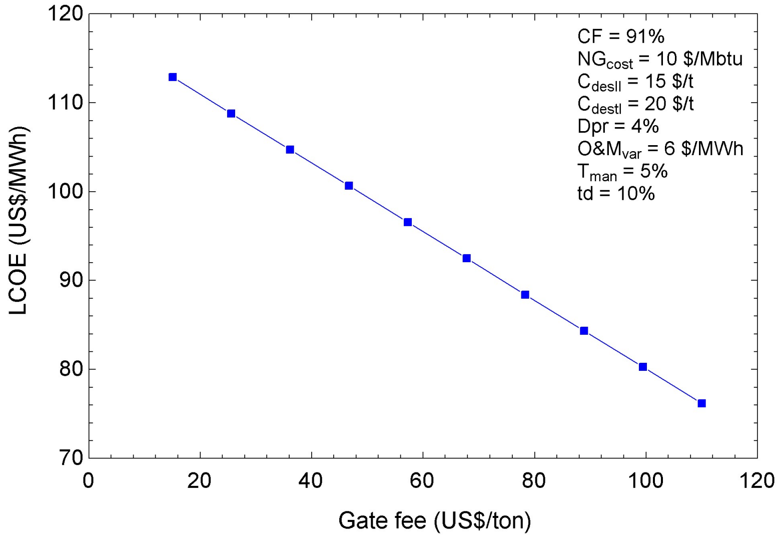


Figure 4.16: LCOE in US$ of 2017 per MWh calculated with the Z_{in}^{max} of the GT4 repowering option as a function of the MSW cost (gate fee).

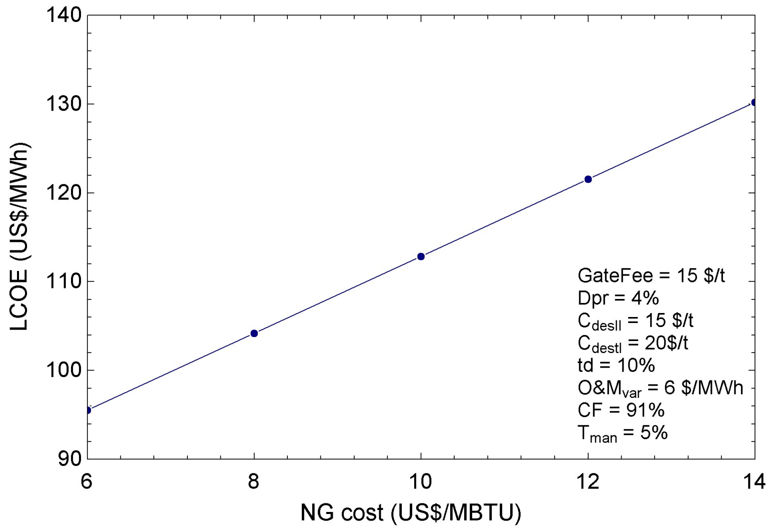


Figure 4.17: LCOE in US$ of 2017 per MWh calculated with the Z_{in}^{max} of the GT4 repowering option as a function of the NG cost.

4.5 Conclusive remarks and future work

It can be concluded that all repowering options present higher profit potential than the single-fueled WTE plant. The choice of building a single-fueled WTE plant and operate it with a capacity 25% smaller over building a hybrid WTE-GT plant with GT4 layout prevents an extra gain of almost € 6 million per year. In terms of energy, exergy, economic and environmental (4E) performance, comparing all four repowering alternatives indicates that the best option is GT4 (case B). Since layout A requires less modification to the original design, if layout A were preferred over B the best case would be GT2. Thus, it can be concluded from the investigated cases that the best alternative is always the larger GT because it represents a smaller MSW thermal input percentage, which offers a higher potential profit, a better overall exergy performance and smaller carbon emission. However, the choice of the gas turbine should not be based only in the capacity, the exergy/isentropic efficiency should guide this crucial choice to avoid ruining a potentially good repowering layout. Moreover, GT2 is preferable over GT3 in both energy and economic aspects, even though GT3 has a smaller MSW thermal input percentage than GT2. Future work suggestions are to calculate the LCOE of the other repowering options; try to improve the performance of the equipment with exergy efficiencies below the average of their class; investigate the influence of the layout on the 4E performance of the repowering options; calculate the cost of the changes required to the existing facility in order to implement the other repowering options, and obtain some economic indicators such as the internal rate of return and payback time.

5
Energy-ecologic efficiency

This chapter presents both an original contribution and a comprehensive technical review of the existing methods of “energy-ecologic efficiency” and “ecological efficiency”. As an original contribution, this chapter aims at investigating for the first time, through the above-mentioned approaches, power plants fueled by municipal solid waste and hybrids with natural gas. A novel strategy is proposed to solve the main flaws of the existing methods and allow the evaluation of multi-fuel plants in a more practical way. The strategy allows to quantify through a single-score indicator the human health toxicity and climate change potential along with resource depletion due to fuel misuse. It has the advantage of being simple and applicable to any thermal system, allowing quick comparisons between different sources. The major contributions of this chapter are: i) to perform a critical analysis, pointing out nuances and limitations of the investigated techniques; ii) to evaluate systems fueled by urban waste, ranking them among fossil and renewable sources including biomass, biofuel and natural gas; iii) to study the influence of emission abatement and biogenic carbon offset due to biomass regrowth regarding waste fired plants. As unique findings of this chapter, are shown the advantages of using urban waste as fuel integrated to gas turbines in combined cycles as an alternative to conventional single-fueled waste-to-energy plants, namely: i) hybrid plants perform 12-24% better than single-fueled plants; ii) the only way a single-fueled plant can overcome a hybrid plant is if it performs co-generation with a thermal efficiency 27% superior.

5.1
Introduction

Evaluating the environmental performance of thermal plants has become an indispensable procedure in engineering assessments. Developing simple and accessible strategies, preferably capable to quantify environmental impacts through single-score indicators, has been attracting the attention of researchers worldwide. In the last decades, several authors have proposed general methods to evaluate all kinds of thermal systems. However, some of them may require long and complex techniques, be dependent on an excessive number of assumptions or allow comparisons only between specific cases. Such difficulties may discourage the application of a method even more when complicated contexts are involved or whenever the environmental issue is not a priority. An attractive arrangement proposed by Romanian/American authors in the late 90's, called "Energy-Ecologic efficiency", has the interesting advantages of being simple and general (applicable to any thermal system). It bases on the idea of joining the impacts of toxicity and fuel depletion by combining three parameters: fuel burning emissions, fuel calorific power and thermal efficiency.

The original concept aimed at developing an indicator that could place a combustible within a range set at the upper limit by the "cleanest fuel" and at the bottom limit by the "dirtiest fuel". However, a shortcoming relies on the fact that the criterion for choosing such range limiting fuels is based only on their combustion emission potential, not taking into account their energy carriers, i.e. the emissions from their production pathway. Such hypothesis may cause a significant impact whenever the overall electricity generation pathway is considered. Nevertheless, this is an intrinsic characteristic of the approach that does not invalidate it, but contributes notably for its facility compared to other methodologies such as the life cycle assessment (LCA). Hence, it should be highlighted that this method presents limitations, further detailed in the text, along with which is the fact that the environmental effects gauged by the indicator disregard life cycle burdens.

Certainly, the method's versatility and uncomplicated employment are its major appealing features, making it an interesting tool for researchers and engineers who would like to supplement their feasibility studies with an energy-environmental perception. Up to the present, it has been applied to different fuel conversion technologies and the initial proposal has been subject of further improvements and modifications developed by its own creators and other researchers worldwide. However, a comprehensive technical review including all those arrangements has never been performed and the indicator has never been calculated to single-fueled municipal solid waste (MSW) fired plants. Moreover, hybrid combined cycles fueled by MSW and natural gas (NG) have never been deeply investigated in the light of such approaches. In addition, the influences of emission abatement and offset of biogenic carbon in MSW due to biomass regrowth have never being studied using such strategies. As a result of the critical analysis, the main issues involving the existing methods are identified and a novel strategy is proposed to solve them. Besides that, an alternative technique is proposed to employ the method to plants fueled concomitantly by two or more different combustibles.

5.2 State of the art

"Energy-Ecologic efficiency" is the name of the method/indicator proposed by Cardu & Baica [96] in 1999. "Ecological efficiency" is the name of the method/indicator proposed by Villela & Silveira [30] as a variation of the original method. Since both have a common root, they will be referred in this work simply as "EE". The main idea of EE is to consider simultaneously a plant's air pollution potential and efficacy in converting fuel into useful energy, giving a measure of its potential harm to humans and the environment due to fuel misuse. It aims at measuring throughout a percentage-score the environmental performance of an energy system based on its thermal efficiency and main pollutants emission. Later, Villela & Silveira [30] included a modification to the original method increasing the number of pollutants considered in the analysis. Then, the author in [66] used it to propose an alternative application for economical purposes, where the percentage-score is used as an indirect measure of the environmental costs of electricity production. The main journal publications about the Energy-Ecologic efficiency and Ecological efficiency are summarized here below, where non-peer-reviewed references were excluded (conference proceedings, theses and dissertations).

– Cardu & Baica [96] present the original method of energy-ecologic efficiency.

– Cardu & Baica [97] present the first variation of the original method and apply it to several types of coal and oil fired plants.

– Cardu & Baica [98] present a second variation of the original method to account for controlled emissions of nitrogen and sulfur oxides.

– Cardu & Baica [99] present an analogy between the energy-ecologic efficiency of flue gases and the seismic activity of the earth's crust.

– Cardu & Baica [100] apply the method to circulating fluidized bed boilers comparing it to conventional coal boilers.

– Cardu & Baica [101] propose the first approach to dual-fuel thermopower plants, evaluating the combination of coal and fuel-oil with methane and hydrogen.

– Silveira et al. [102] use Villela's method for the first time in a comparative study between a 1000 MW combined cycle power plant and 1000 kW Diesel power plant.

– Villela & Silveira [30] evaluate the ecological efficiency of natural gas and Diesel combustion in combined cycle power plants.

- Cardu & Baica [103] introduce a new criterion called "SONOX" in order to take into account the European Union norms regarding the respective emissions limits of sulfur dioxide and nitrogen oxides.
- Coronado et al. [31] study the ecological efficiency of biodiesel, Diesel and blends with both fuels considering for the first time the fuel's energy carriers.
- Silveira et al. [104] increment Villela's method and apply it to natural gas burn in a hospital co-generation system.
- Filho et al. [85] apply it to sugar cane bagasse burn in a circulating fluidized bed gasifier. They investigate for the first time how EE is influenced by the offset of biogenic carbon emissions from biomass burn due to CO_2 capture from plant regrowth.
- Santos et al. [87] evaluate it for natural gas combined cycles.
- Carneiro & Gomes [50] quantify the ecological efficiency and use it to estimate the cost of pollution abatement equipment of a case study regarding a hybrid combined cycle fueled by municipal solid waste and natural gas.

As noticed, this is the first time the EE method undergoes a comprehensive technical review in order to compare both original and derived approaches, and present their simultaneous application to two MSW conversion technologies. As original contributions of this chapter one can highlight: (i) to propose a novel strategy for the EE method maintaining its uncomplicated employment and single-score indicator idea but eliminating the major shortcomings of the existing approaches; (ii) to quantify the EE value of single-fueled waste-to-energy (WTE) plants; (iii) to quantify the EE value of a class of hybrid power plants fueled by MSW and NG; (iv) to investigate the influence of pollutants abatement on the EE value; (iv) to investigate the influence of carbon dioxide (CO_2) offset from biomass growth on the EE value; (v) to propose an alternative technique for EE evaluation in multi-fuel plants.

5.3 Method

At first, the method developed by Cardu & Baica [96] is technically reviewed. It proposes a dimensionless indicator (energy-ecologic efficiency) ranging between 0 and 1, analogously to the 1^{st} Law efficiency (η), v. eq. 5-1, where zero means an undesirable condition (maximum pollution) and 1 indicates the most desirable condition (null pollution). The logic is that EE should be large when η is large, because the more efficient a plant is the smaller its impact on the environment (less waste of fuel). Moreover, EE should be large when equivalent carbon dioxide (CO_{2eq}) is small, because the fewer emissions the better. For the extreme case when the plant's efficiency is zero $\eta = 0$, EE=0 (for LHV>0), i.e., a very inefficient plant consumes fuel, generates pollution without producing useful energy. Since $\eta = 1$ is not achievable in real conditions, the other extreme case is EE=1, which can be obtained only if CO_{2eq}=0, as for instance from burning a fuel that only emits water vapor, such as hydrogen. In this sense, an empirical formula was proposed according to eqs. 5-1 and 5-2 [96]:

$$\eta = \frac{\omega}{q} \therefore \omega = \eta \cdot q \tag{5-1}$$

where η is the gross 1st Law efficiency, ω is the specific useful energy produced by the plant [MJ/kg] and q is the energy consumed per unit mass of fuel [MJ/kg].

$$\begin{aligned}\varepsilon &= \frac{\omega}{(\omega + 2\ f_{CO2eq})} \\ &= \frac{\eta \cdot q}{(\eta \cdot q + 2\ f_{CO2eq})} \\ &= \frac{1}{1 + \frac{2\ f_{CO2eq}}{\eta\ q}}\end{aligned} \tag{5-2}$$

where ε is the energy-ecologic efficiency and f_{CO2eq} is the emission factor of equivalent carbon dioxide (kg of equivalent CO_2 per kg of fuel).

The "pollution indicator" is defined by Cardu & Baica [96] according to eq. 5-3:

$$\Pi = \frac{f_{CO2eq}}{q} \tag{5-3}$$

where Π corresponds to kg of equivalent CO_2 per MJ of fuel and $q = LHV_{fuel}$ is the lower heating value of the fuel (MJ/kg);.

Substituting 5-3 into 5-2, the general formula of EE is proposed by [96]:

$$\varepsilon = \frac{1}{1 + 2\ \Pi/\eta} \tag{5-4}$$

Evaluating ε (Π,η), the authors observed that eq. 5-4 resulted in graphs too scattered to be evaluated with curves, being necessary to introduce a variant to the model. This results in the subsequent study by Cardu & Baica [97], in which the authors consider a first approximation for ε to be:

$$\varepsilon \propto [\ \phi(\eta) \cdot \psi(\Pi)\ ]^n \tag{5-5}$$

Moreover, they assume two extreme cases based on two fictitious fuels: hydrogen and sulfur. Here is an important hypothesis of the method, that is, *hydrogen is assumed as the least polluting fuel and sulfur as the most polluting* *fuel*. They [97] consider that eq. 5-5 has to satisfy the three following hypotheses:

1. For hydrogen: $\Pi_H = 0$ and $\varepsilon = 1$, for all values of η;

2. For the weakest existing fuel "Rovinari lignite": $\Pi_{RL} = 2.045$ kg/MJ (because $f_{CO2eq,RL} = 12.99$ kg/kg and $LHV_{RL} = 6.35$[1] MJ/kg), for $\eta_{RL} = 0.3 - 0.4$ and $\varepsilon_{RL} = 0.3 - 0.4$;

3. For sulfur (and hydrogen): $\Pi_S = 134$ kg/MJ (because $f_{CO2eq,S} = 1400$ kg/kg v. eq. 5-12 and $LHV_S = 10.45$ MJ/kg) and $\varepsilon = 0$, for all values of η.

Since Π varies from 0 to 134 (large interval) and ε varies from 0 to 1 (small interval), the authors considered that the sub-function ψ in eq. 5-5 must be in logarithmic form:

$$\psi(\Pi) = \ln(K \pm \Pi) \tag{5-6}$$

where K is a constant. At the same time, according to eqs. 5-2 and 5-4, the sub-function ϕ in eq. 5-5 "must give to the function ε a property that if η increases, $\phi(\eta)$ and ε also increase" [97]. Therefore ϕ must be a function of both η and Π, i.e. $\phi = \phi(\eta, \Pi)$, which, in order to satisfy condition 1, must be:

$$\phi(\eta, \Pi) = \frac{\eta}{\eta + \Pi} \tag{5-7}$$

Substituting 5-6 and 5-7 into 5-5, one has:

$$\varepsilon = \left[c\,\frac{\eta}{\eta + \Pi}\,\ln(K \pm \Pi)\right]^n \tag{5-8}$$

where c is a constant.

In order to accomplish with condition (3.) (ε=0 for $\Pi = 134$, one must have $ln(1)$, which is obtainable for instance with $\ln(K - 134) = 0 \therefore K = 135$. Replacing η and Π according to condition (1.) into eq. 5-8, the constant $c = \frac{1}{\ln 135} \simeq 0.204$ can be obtained. The value of the exponent n can be obtained using condition (2.), thus, $n \simeq 0.5$. Finally, the proposed function for the *energy-ecologic efficiency* is given by eq. 5-9:

$$\varepsilon = \left(0.204\frac{\eta}{(\eta + \Pi)}\ln(135 - \ \Pi)\right)^{0.5} \tag{5-9}$$

It can be easily observed that for the extreme cases of H and S as fictitious fuels eq. 5-9 works as intended:

[1] This value is much lower than the LHV range reported for lignite in the USA, which is between 11.63-17.45 MJ/kg [105].

$$\begin{aligned}\varepsilon_{hydrogen} &= \left(0.204\frac{\eta}{(\eta+0)}\ln(135-0)\right)^{0.5}\\ &= [0.204\cdot\ln(135)]^{0.5}\\ &= 1\end{aligned} \tag{5-10}$$

$$\begin{aligned}\varepsilon_{sulfur} &= \left(0.204\frac{\eta}{(\eta+134)}\ln(135-134)\right)^{0.5}\\ &= \left[\frac{0.204\eta\ln(1)}{\eta+134}\right]^{0.5}\\ &= 0\end{aligned} \tag{5-11}$$

The choice over the limiting fuels seems to be motivated based on the energy conversion context of the original study, which may sound somewhat arbitrary or site-specific. However, the method is valid as long as the user employs it to investigate a case within such limits in a context where there is no fuel better than hydrogen or worse than sulfur.

5.3.1 Determining an equivalent pollutant (Approach I)

As observed in the description so far (v. eq. 5-3), the authors in [96] consider a formulation that bundles the different pollutants using as equivalent substance the carbon dioxide, i.e. an equivalence formulation gives as result the CO_{2eq} emission factor (f_{CO2eq}) according to eq. 5-12. The polluting gases taken into account in such approach are CO_2, sulfur dioxide (SO_2) and nitrogen oxides (NO_x), which have been chosen for being emitted by thermopower plants in large quantities, whereas some other harmful pollutants (e.g. heavy metals, dioxins, etc.) have been disregarded because of their very small concentrations.

$$f_{CO2eq} = f_{CO2} + 700\cdot f_{SO2} + 1000\cdot f_{NOx} \tag{5-12}$$

In eq. 5-12 f_{CO2}, f_{SO2} and f_{NOx} represent the CO_2, SO_2 and NO_x content resulting from burning one kilogram of fuel (kg/kg_{fuel}, or denoted kg/kg_f) and the coefficients 1, 700 and 1000 were obtained from the maximum accepted concentrations allowed in the work place affecting human health (according to specialized literature/standards from Romanian context). For comparison purposes, it is taken the example of Brazil, where an equivalent actual literature is “NR 15 – Unhealthy activities and operations” [106], which indicates the maximum concentrations allowed for a 48-hour week work journey, as described in table 5.1.

Table 5.1: Maximum concentrations (tolerated limits) of CO_2, SO_2 and NO_x allowed in the work place in $mg_{substance}\ m_{air}^{-3}$.

Pollutant	Brazilian legislation [106]	Hypotheses from Cardu [97]
CO_2	7020 mg m^{-3}	10000 mg/m^3
SO_2	10 mg/m^3	15 mg/m^3
NO_2 or NO_x	7 mg of NO_2/m^3	10 mg of NO_x/m^3

The logic used by the authors is that the smaller the concentration limit allowed for a pollutant in the air, the stronger its effect on human health, thus its contribution to equivalent CO_2 emissions should be greater. Since the importance of CO_2 in the total CO_2 equivalent emissions should be 1, the values from table 5.1 for SO_2 and nitrogen dioxide (NO_2) are inversely proportional to CO_2. Hence, the coefficients of eq. 5-12, according to approach I, are $1 = 10000/10000$; $700 \simeq 10000/15$ and $1000 = 10000/10$, for CO_2, SO_2 and NO_x, respectively. It is interesting to notice that if the coefficients were obtained from the Brazilian legislation [106], they would also be $1 = 7020/7020$; $700 \simeq 7020/10$ and $1000 \simeq 7020/7$ for CO_2, SO_2 and NO_x, respectively (assuming NO_x composed only by NO_2). The CO_2 emission per mass unit of fuel (f_{CO2}) is easily obtained if the chemical formula of the fuel is known. For instance, taking the example of methane, its stoichiometric chemical reaction with dry air (79% nitrogen, 21% oxigen) considering complete combustion is:

$$1\ CH_4 + 2\ O_2 + 3.76\ N_2 = 1\ CO_2 + 2\ H_2O + 3.76\ N_2 \tag{5-13}$$

Knowing the molecular weights of $CH_4 = 16$ kg/kmol and $CO_2 = 44$ kg/kmol, f_{CO2} is easily calculated by: $f_{CO2} = 44/16 = 2.75$ kg/kg$_f$. However, it is not always simple to estimate emission factors this way, being easier to collect data from the literature as shown in tab. 5.2 and discussed in the following subsection.

In summary, approach I proposes to determine the EE value by quantifying the emissions of each one the above-mentioned gases separately due to burning of a fuel, then using equations 5-3, 5-9 and 5-12 to obtain the energy-ecologic efficiency indicator (ε). The first studies which introduced the method applied it to natural gas, oil and to six different types of coal in the context of Romania: "lignite Rovinari" - 0.7% sulfur, "pit coal" - 2% sulfur, "oil with high S content" - 3% sulfur, etc.)[96]. The objective was to rank the fuels in terms of their air pollution potential using emissions estimated from the fuels' chemical reactions, except for NO_x (based on literature data). Figure 5.1 shows the influence of η on ε.

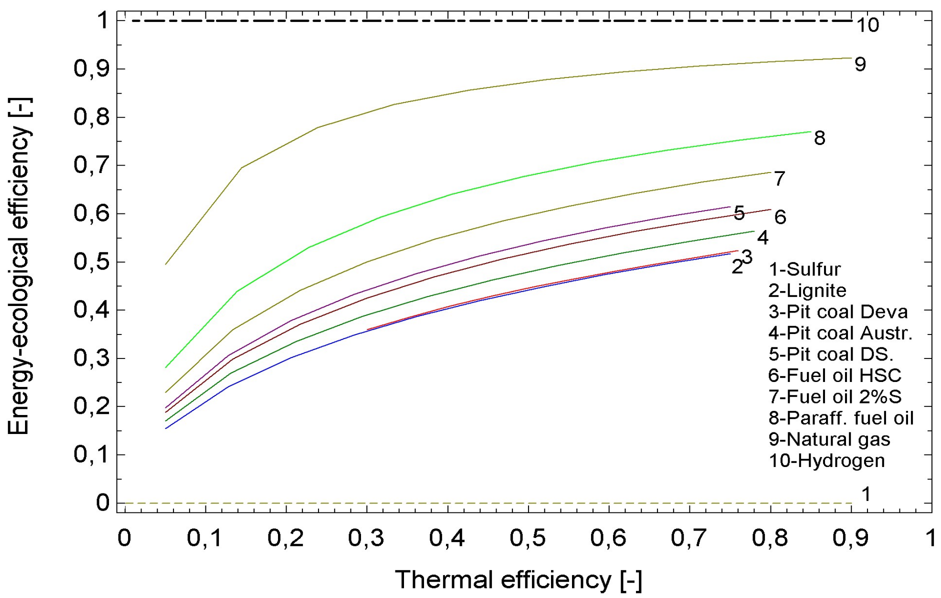


Figure 5.1: Energy-ecological efficiency as a function of thermal efficiency for different fuels [97].

5.3.2 Accounting for controlled emissions (Approach II)

Up to this point, employing the method originally as it was proposed presumes the use of uncontrolled emissions, i.e. those that are directly emitted from fuel burn without any emission control procedure or post combustion treatment. In order to account for the possibility of pollutant abatement technology, indispensable in many cases to comply with emission regulations, Cardu&Baica [98] propose a II approach that consists of adding two variables to eq. 5-12: the desulphurization efficiency (σ_s) and the nitrogen oxides abatement ($DeNO_x$) efficiency (σ_n). For instance, assuming σ_n=60% and σ_s=25%, it can be obtained:

$$\begin{aligned} f_{CO2eq} &= f_{CO2} + 700\ (1-\sigma_s)\cdot f_{SO2} + 1000\ (1-\sigma_n)\cdot f_{NOx} \\ &= f_{CO2} + 525\cdot f_{SO2} + 400\cdot f_{NOx} \end{aligned} \tag{5-14}$$

As observed, the NO_x and SO_2 contributions to the total CO_{2eq} are less important here than in approach I, with the reduction factor σ being a crucial variable in the determination of EE. Thus, all emission factors in eq. 5-14 are supposed to consider uncontrolled emissions (otherwise abatement would be counted twice), however one should be aware that the results derived from it regard a plant applying $DeNO_x$ and desulphurization procedures. This leads to the conclusion that employing approach I with controlled emissions should be equivalent to approach II if the proper values of f and σ are used.

It is known that diverse abatement techniques may be applied to the different pollutants. For instance, control of SO_2 regards pos-combustion equipment for acid gas control, such as spray drying or dry sorbent injection, and control of NO_x requires specific procedures employed *during combustion.* Nitrogen oxides are a combination of nitric oxide (NO), nitrogen dioxide (NO_2) and nitrous oxide (N_2O), where the primary component is NO; NO_2 and N_2O being formed in smaller amounts [107]. In this sense, it is a hard task to find specified emission factors of NO_2 and N_2O separately and even harder to find good data for uncontrolled emission factors. The USA Environmental Protection Agency (EPA) [108, 107] suggests the following regarding NO_x abatement techniques for NG burn in boilers/gas turbines and MSW combustion:

- *NG burn*: The two most prevalent combustion control techniques used to reduce NO_x emissions from NG-fired boilers are flue gas recirculation (FGR) and low NO_x burners. NO_x emission reductions of 40-85% (relative to uncontrolled emission levels) have been observed with low NO_x burners. An FGR system is normally used in combination with specially designed low NO_x burners, in this case, they are capable of reducing emissions by 60-90% [109]. The majority of GTs currently manufactured are "lean-premix" staged combustion turbines. In a lean-premix combustor, fuel and air are thoroughly mixed in an initial stage resulting in a uniform mixture addressed to a secondary stage where the combustion reaction occurs. GTs using staged combustion are also referred to as "dry low NO_x combustors" [108]. Reductions of 60-90% in NO_x using such technology are reported [109]. In fact, from tab. 5.2 it can be noted a 70% reduction between uncontrolled NG combustion and lean-premix GT.
- *MSW burn*: Selective non-catalytic reduction (SNCR) is a technique where ammonia (NH_3) or urea is injected into the furnace along with chemical additives to reduce NO_x without the use of catalysts, where reductions of 45% are achievable [107]. Another technique is selective catalytic reduction (SCR), where NH_3 is injected into the flue gas downstream of the boiler and passes through a catalyst bed, where NO_x is reduced up to 80% [107]. From experimental data in Korea, SNCR processes present NO_x reduction efficiency of 40–75% while SCR achieves up to 90% [110].

In this sense, attention should be paid to distinguish between controlled and uncontrolled emissions, specially when collecting data for emission of

NO_x or other pollutant whose control occurs during combustion, which may go unnoticed. Table 5.2 summarizes the emission factors (controlled and uncontrolled) of the main pollutants for MSW mass burn and NG burn in gas turbines based on specified heating values (HV) of such fuels. The conversion formula to the desired fuel heating value (HV) is:

$$f = f_{ref} \frac{HV}{HV_{ref}} \tag{5-15}$$

where HV is the desired heating value, HV_{ref} is the reference heating value, f is the desired emission factor and f_{ref} is the reference emission factor. Reduction factors (σ) values can also be obtained from data in tab. 5.2.

Table 5.2: Emission factors for NG burn in gas turbines and mass burn of MSW [107, 108, 111].

	Natural gas[a]		Municipal solid waste[b]	
Pollutant	Uncontrolled (kg/MJ)	Controlled[c] (kg/MJ)	Uncontrolled (kg/ton)	Controlled (kg/ton)
CO_2	4.73×10^{-2}	-	9.85×10^{2}	-
SO_2	1.46×10^{-6}	-	1.73	2.77×10^{-1}[d]
NO_x or NO_2	1.38×10^{-4} (NO_x)	4.26×10^{-5} (NO_x)	1.83 (NO_x)	1.00 (NO_2)[e]
PM	2.84×10^{-6}	-	12.6	3.11×10^{-2}[d]
CH_4	3.70×10^{-6}	-	-	-
N_2O	1.29×10^{-6}	-	$\frac{1.05 \times 10^{-1}}{(1-\sigma_n)}$ [f]	1.05×10^{-1}[e]

[a] Emission factors of stationary GTs; NG with $HHV_{NG} = 38$ MJ/m^3 @ 15.6°C/1 atm [108].
[b] All factors in kg/ton of refuse combusted ($LHV_{MSW} \simeq 10.5$ MJ/kg) [107, 111].
[c] DeNO_x technique: lean-premix GT [108].
[d] Emission control equipment: spray dryer + fabric filter [107].
[e] DeNO_x technique: SNCR with urea [111].
[f] σ_n is the reduction factor of nitrogen oxides for SNCR with urea (suggested 0.4-0.75 [110]).

The disadvantage of approaches I and II, according to the authors, is that synergy effects are not taken into account. However, there are two more problems with these arrangements. The major issue regards a very common misconception between atmospheric pollution and contamination in confined environments. Atmospheric pollution or contamination of the ambient air is the main source of environmental pollution of thermal systems due to emissions at stack, which are supposed to be measured away from the polluting source (large scale). On the other hand, indoor contamination affects exclusively the workers and is supposed to be measured in the vicinity of the polluting source (small scale). Atmospheric emissions can contribute to several environmental impacts such as climate change, ozone layer depletion, smog, etc., whereas pollution in confined environments can contribute only to human health issues. In the way the calculation of equivalent emissions is proposed, such concepts are merged, which might cause confusion. The second issue is that the arrangement does not take into account the difference between energy produced as electricity and heat. Since it considers the "general 1st Law efficiency", it does not distinguish whether or not the system applies co-generation (combined heat and power - CHP). Nevertheless, the method is legit, but the user should be aware of those issues whenever applying or improving the technique.

5.3.3 Adding new pollutants (Approach III)

Subsequent studies [30, 85, 112] propose to include other pollutants in the analysis and adapt the coefficients of eq. 5-12 based on air quality standard values. Villela & Silveira [30] have been identified as one of the precursors implementing this strategy, proposing a III approach based on two air quality standards. The first one is the Brazilian legislation for air quality "CONAMA/90" [113], which by the way has been updated recently [114]. The second one is the international Air Quality Guidelines by World Health Organization [115]. The method consists of including the concentration of particulate matter (PM) in the equivalence formulation as described in eq. 5-16 where the logic for obtaining the new coefficients was to divide the original CO_2 concentration by the values indicated in tab. 5.3.

$$f_{CO2eq} = f_{CO2} + 80 \cdot f_{SO2} + 50 \cdot f_{NOx} + 67 \cdot f_{PM} \tag{5-16}$$

That is, the coefficients of eq. 5-16 are $1 = 10,000/10,000$; $80 \simeq 10,000/125$; $50 = 10,000/200$ and $67 \simeq 10,000/150$; for CO_2, SO_2, NO_x and PM, respectively. However, it is important to note that there is a crucial issue regarding the units of these coefficients. The tolerated limits shown in tab. 5.1 are much higher compared to the air quality standards shown in tab. 5.3; the first ones being in $\mathrm{mg\,m^{-3}}$ and the second ones being in $\mathrm{\mu g\,m^{-3}}$. Thus, the coefficients of SO_2, NO_x and PM proposed in this approach have units of $\mathrm{mg\,\mu g^{-1}}$, while originally they had units of $\mathrm{mg\,mg^{-1}}$. Therefore, for eq. 5-16 to be coherent it should be dimensionless, i.e. such coefficients should be $80,000 \simeq 10,000/125 \times 10^{-3}$; $50,000 = 10,000/200 \times 10^{-3}$ and $67,000 \simeq 10,000/150 \times 10^{-3}$. This would make the contribution of SO_2, NO_x and PM to f_{CO2eq} much superior than in approaches I and II (v. eqs. 5-12 and 5-14), which would cause the pollution indicator (Π) to be also greater and the EE value (ε) to be smaller than suggested. Anyhow, the main problem with this approach is that it mixes even more the concepts of pollution in a confined environment (which is used in the original approach to balance the pollutants) and pollution in ambient air (which contextualizes the air quality standards).

Another unclear issue is if this arrangement presumes only the use of uncontrolled emissions or could also be employed with controlled emission

Table 5.3: Pollution concentrations limits adopted in approach III [66].

Pollutant	Tolerated limit or Air Quality Standards	Unit	Source
Carbon dioxide	10000	$mg\,m^{-3}$	Cardu&Baica[96]
Sulfur dioxide	125	$\mu g\,m^{-3}$	WHO[115]
Nitrogen dioxide	200	$\mu g\,m^{-3}$	WHO[115]
Particulates	150	$\mu g\,m^{-3}$	CONAMA[113]

factors. As noted from eq. 5-16, NO_x coefficient is smaller than SO_2's, i.e. NO_x contributes less to total CO_{2eq} than SO_2. This gives the idea that this approach is more in accordance with approach II (eq. 5-14) than with approach I (eq. 5-12), which indicates that the results derived from it would be more in line with a plant applying at least $DeNO_x$ procedures. Investigating another study [66] that uses this method to estimate the environmental cost of electricity production in thermopower plants, one can conclude that only uncontrolled emission factors should be used. This is because the author in [66] proposes a correlation between EE and the cost of the pollution abatement procedures, which only makes sense if EE is calculated considering uncontrolled emissions. Somehow, the results from such study [66] demonstrate that there is a correlation between the ecological efficiency value calculated through eq. 5-16 and the costs of pollution abatement. Yet, the user should be aware of those nuances while using EE for economical purposes.

Another reservation that can be highlighted from this approach with respect to the original arrangement is that the constants K, c and n in eq. 5-8 would have to be recalculated taking into account eq. 5-16. This is because originally they had been determined using eq. 5-12, which is no longer valid in this approach. An additional misleading issue is to take into account the emissions from the fuel production pathway (fuel's energy carriers) using the EE method original arrangement, as performed by Coronado et al. [31]. The authors [31] investigate the EE value of biodiesel, among others, through approach II accounting for the biodiesel life cycle emissions (before and at the tailpipe). However, in order to do so, the general formulation should be completely changed, starting by re-evaluating the "worst" and "best" fuels hypotheses and determining the emissions from their life cycle as well. Because, for instance, even though hydrogen does not emit any of the considered pollutants during its burn, its production pathway probably presents non negligible carbon emissions.

Even though those issues are not made explicit by the authors, some conclusions can be reached from the critical analysis of approaches I-III:

1. The results obtained from approach III are not directly comparable to

those obtained from approaches I and II.

2. A general rule for employing approaches I-III is that they presume the use of emission factors regarding "raw" (untreated, i.e. uncontrolled) emissions, although the effect of NO_x and SO_2 abatement may be considered through approach II.

3. If willing to employ the EE method originally as it was proposed, the user should not account for emissions from the fuel's life cycle, i.e. only emissions directly derived from the fuel burn should be considered.

With this, the first objective of this chapter is achieved, which was to present a comprehensive technical review of the energy-ecologic efficiency method filling the existing gap in the literature. Next, it is proposed an innovative strategy to eliminate the main flaws identified so far.

5.3.4 Proposing an innovative method (Approach IV)

In order to solve the main shortcomings of the approaches described previously, a novel strategy is proposed to evaluate the energy-ecological efficiency of thermopower systems, here called approach IV. It may be applied using uncontrolled or controlled emissions from fuel burn, as desired. Even though the idea borrows some concepts from the LCA methodology, as it will be noted further, it continues to consider only direct emissions derived from burning of a fuel without accounting for the fuel's energy carriers/production pathways. It consists of performing the calculation of the pollution indicator (Π) separately in two parts, considering the emissions contributing to: i) climate change and ii) human toxicity. In this sense, it is proposed to have Π_{GW} to gauge "global warming potential" and Π_{HT} to gauge "human toxicity potential". Expressing both Π in the most general way possible, one has:

$$\Pi_{GW} = \frac{\sum_{i=1}^{p} x_i \cdot f_{pol,i}}{LHV_{fuel}} \tag{5-17}$$

$$\Pi_{HT} = \frac{\sum_{j=1}^{r} y_j \cdot f_{pol,j}}{LHV_{fuel}} \tag{5-18}$$

with f_{pol} being the pollutant emission factor in [kg pollutant/kg fuel], x and y being the characterization factors[2], p and r being the total number of pollutants considered for global warming (GW) and human toxicity (HT), respectively.

[2] In the LCA methodology substances that contribute to an impact category are multiplied with a characterization factor (coefficient) that expresses the relative contribution of the substance.

In order to maintain the simplicity of the method, it is proposed to restrict the number of pollutants in six (three in each category, i.e. $p = r = 3$), and to consider x_i measured in [kg CO_{2eq}/kg pollutant i] and y_j in [kg 1,4-DCB_{eq}/kg pollutant j][3]. In addition to the four substances already considered by approaches I-III, it is proposed to include two other substances in the analysis: methane (CH_4) and nitrous oxide (N_2O). They will compose the GW indicator along with CO_2, while the other substances (PM, SO_2, NO_x or NO_2) will contribute to HT. Table 5.4 presents the assumed values of x and y, which have been based on theories used in many LCA studies (HT factors have been adapted from [116]).

Table 5.4: Characterization factors for calculation of Π_{GW}[1] and Π_{HT} [116, 117].

Pollutant	Characterization factor	Unit	Applied to:
CO_2	1	kg CO_{2eq}/kg CO_2	GW
CH_4	28	kg CO_{2eq}/kg CH_4	GW
N_2O	265	kg CO_{2eq}/kg N_2O	GW
PM[2]	38.75	kg 1,4-DCB_{eq}/kg PM	HT
SO_2	4.54	kg 1,4-DCB_{eq}/kg SO_2	HT
NO_x or NO_2	56.71	kg 1,4-DCB_{eq}/kg NO_x	HT

[1] GW potentials regard a time horizon of 100 years [117].
[2] Assumed here as total particulate matter.

Thus, the proposed indicators Π_{GW} and Π_{HT} can be expressed as:

$$\Pi_{GW} = \frac{f_{CO2} + 28\, f_{CH4} + 265\, f_{N2O}}{LHV_{fuel}} \tag{5-19}$$

$$\Pi_{HT} = \frac{4.54\, f_{SO2} + 56.71\, f_{NOx} + 38.75\, f_{PM}}{LHV_{fuel}} \tag{5-20}$$

From eq. 5-3, one can observe that the two proposed equivalent emission factors, f_{CO2eq} [kg CO_{2eq}/kg_f] and $f_{1,4DCBeq}$ [kg 1,4DCB_{eq}/kg_f], can be expressed separately as:

$$\begin{aligned} f_{CO2eq} &= f_{CO2} + 28\, f_{CH4} + 265\, f_{N2O} \\ f_{1,4DCBeq} &= 4.54\, f_{SO2} + 56.71\, f_{NOx} + 38.75\, f_{PM} \end{aligned} \tag{5-21}$$

As observed, both Π_{GW} and Π_{HT} respect the original unit of Π [kg of equivalent pollutant per MJ of fuel], preserving Π_{GW} in [kg CO_{2eq}/MJ] but including Π_{HT} in [kg 1,4-DCB_{eq}/MJ]. Two possibilities derive from this:

1. Split the EE indicator into two, namely: $\varepsilon_{GW} = \phi_1(\eta, \Pi_{GW})$ and $\varepsilon_{HT} = \phi_2(\eta, \Pi_{HT})$.

[3] 1,4-Dichlorobenzene is a substance commonly used to calculate human toxicity impact indicator in the LCA methodology (expressed in kg 1,4-DCB per 1 inhabitant).

2. Maintain the single-score EE indicator as a function of Π_{GW} and Π_{HT}, namely: $\varepsilon = \phi_3(\eta, \Pi_{GW}, \Pi_{HT})$.

Both options have advantages and disadvantages. Option 1. allows the possibility to make a distinction between these two indicators and evaluate each impact at a time. That is, one would be able to say for instance if a system presents higher potential to affect the climate than the human health. Option 2. maintains the form of a single-score indicator, with the advantage of being able to rank different energy conversion systems through a unique eco-index. The major disadvantage of option 1. is that a new assumption would have to be made about the "worst fuel" affecting global warming. This is because the hypothesis used in the original approach, that considered sulfur as the worst theoretical fuel, becomes invalid for ε_{GW}, since $\Pi^S_{GW} = 0$ ("sulfur" combustion generates only SO_2, which contributes exclusively to HT). Meanwhile, option 2. has the main disadvantage of requiring to define a relative importance between climate change and human toxicity through weighting factors.

Hofstetter et al. [118] propose a method to determine single-score environmental indicators using the weighted sum of three "damage indicators": "damage to human health", "damage to ecosystem quality" and "damage to resource availability". The original EE strategy already considers the potential of fuel misuse through η, which can be considered to gauge the potential to cause damage to resource availability. With the new proposed approach, Π_{GW} will gauge the potential to cause damage to ecosystem quality and Π_{HT} will gauge the potential to cause damage to human health. If a function could be determined to define $\Pi = \varphi(\Pi_{GW}, \Pi_{HT})$, ε would embrace all three damage indicators, thus being what the study [118] calls an "eco-indicator" or "eco-index", as represented in fig. 5.2. This seems to be a sound solution to maintain the original idea of the EE method, which was to gauge the environmental performance of thermopower plants allowing to rank diverse energy conversion technologies.

Hofstetter et al. [118] propose to correlate the above-mentioned damage indicators using as relative weights the "number of premature deaths per one million European inhabitants per year", based on data from European emission inventories. The study states that from a total of 165 deaths, 122.5 can be associated to damage to human health and 42.5 can be associated to damage to ecosystem quality. Hence, it is here proposed to consider 74.2% as the weighting factor of Π_{HT} and 25.8% as the weighting factor of Π_{GW}, which allows the new pollutant indicator Π to be expressed as a sum of Π_{GW} and Π_{HT} :

$$\Pi = 0.742\,\Pi_{HT} + 0.258\,\Pi_{GW} \tag{5-22}$$

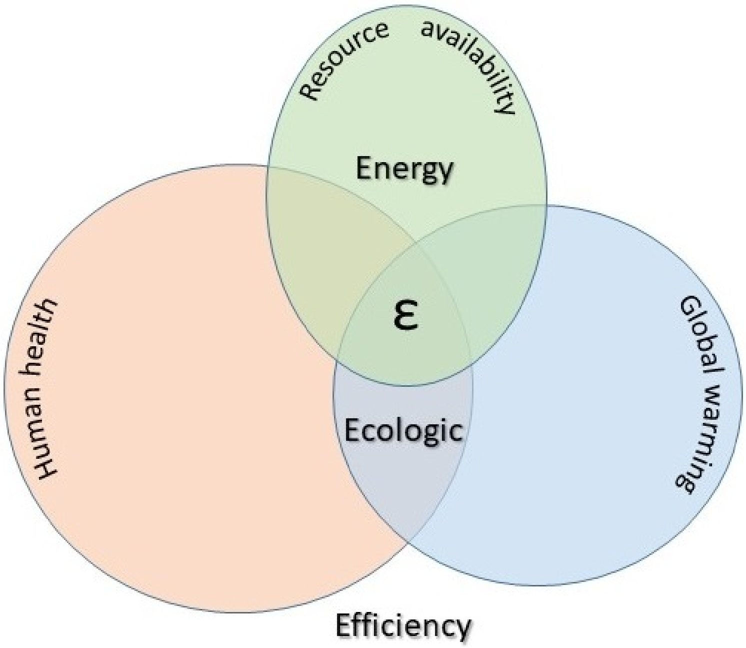


Figure 5.2: Scheme of the new energy-ecologic efficiency concept.

and the total equivalent emission factor f_{eq} [kg/kg$_f$] can be expressed as:

$$f_{eq} = 0.742\, f_{CO2eq} + 0.258\, f_{1,4DCBeq} \tag{5-23}$$

This leads to the need of recalculating the constants K, c and n in eq. 5-8, for which the three original criteria have to be satisfied; (v. hypotheses 1. to 3. after eq. 5-5). For this important task, the values of Π_S and Π_{RL} have to be re-determined according to eqs. 5-19 to 5-22, for which are used the characteristics of Rovinari lignite based on study [96] (20% of C, 0.7% of S, 24.5% of ash, LHV=14.775 MJ/kg[4]) and the emission factors for uncontrolled lignite combustion in stokers[5] described by the EPA [105]. EPA describes also that "N_2O emissions for lignite combustion are not significant, except for fluidized bed combustion. Methane emissions vary with the type of coal being fired and firing configuration, but are highest during periods of incomplete combustion, such as the start-up or shut-down cycle". For which emissions of CH_4 and N_2O for Rovinari lignite are considered null. Finally, the new formulation of ε is proposed:

$$\varepsilon = \left(2.01 \frac{\eta}{(\eta + \Pi)} \ln\left(1.645 - \Pi\right)\right)^{1.7} \tag{5-24}$$

As observed, this equation preserves the extreme cases of hydrogen as the best fuel ($\varepsilon_{hydrogen}$=1) and sulfur (ε_{sulfur}=0) as the worst fuel, which makes

[4]LHV of lignite assumed as 14.775 MJ/kg (instead of 6.352 MJ/kg) to accord w/ [105].
[5]Firing configuration assumed as "pulverized coal, dry bottom, tangencial".

it valid for fuels with Π within the range $\Pi_H < \Pi < \Pi_S$. However, if one desires to investigate a fuel outside of this range, the method can still be applied by adapting the original hypotheses (changing the extreme fuels) and recalculating the constants of eq. 5-5. Moreover, this approach allows the inclusion of more pollutants as desired, as long as the same procedures are respected: determining all characterization factors, verifying the hypotheses of extreme fuels and the need to recalculate the constants of eq. 5-5. A constraint that arises from this approach is that, such as in the LCA, each substance can only be related to one type of impact. Moreover, as mentioned, the approach can be applied taking into account controlled or uncontrolled emissions, being sufficient to use the proper emission/reduction factors (a correlated discussion is presented in the end of section 5.3.6).

This closes the main innovative methodological contribution of this chapter solving the main flaw of the EE method and making it no longer questionable about merging of concepts regarding pollution in confined environments and ambient air. The new strategy here proposed accounts for all three damage indicators and formulates an eco-index that measures the environmental performance of thermal systems in terms of potential to cause climate change, fuel depletion and damage to human health.

5.3.5
Accounting for multi-fuel plants

In this section is presented another original contribution in the methodological level. First, a review is presented of the existing method for evaluating dual-fuel facilities proposed by Cardu&Baica [101]. They suggest to consider the plant fueled by a fictitious "hybrid fuel" composed by fuel 1 (fuel with greater mass = primary fuel) and fuel 2 (supplementary fuel), whose lower heating value (LHV_d) can be obtained through a weighted average of each fuel's LHV, according to eq. 5-25:

$$LHV_d = LHV_{f1} + n_d \cdot LHV_{f2} \tag{5-25}$$

where n_d is the ratio between secondary and primary fuel mass flows with respect to 1 unit mass of the primary fuel: $n_d = \dot{m}_{f2}/\dot{m}_{f1}$, where always $n_d < 1$. Thus, LHV_d does not refer to 1 kg of dual-fuel, but to a "package" formed of 1 kg of primary fuel and n kg of supplementary fuel [101]. The equivalent emission factor of the "dual fuel" $(f_{eq})_d$ can be calculated generally for all approaches as:

$$(f_{eq})_d = \sum_{k=1}^{m} a_k \, (f_{pol,k})_d \tag{5-26}$$

$$(f_{pol,k})_d = (f_{pol,k})_{f1} + n_d \cdot (f_{pol,k})_{f2} \tag{5-27}$$

where $f_{pol,k}$ is the emission factor of pollutant k, a_k is its characterization factor/coefficient used in the equivalence formulation of the chosen approach and m is the total number of pollutants. Then, the pollution indicator of the hybrid fuel is calculated analogously as in eq. 5-3:

$$\Pi_d = \frac{(f_{eq})_d}{LHV_d} \tag{5-28}$$

As observed, this strategy imposes that the mass ratio of the fuels (n_d) needs to be known previously. However, this is not a practical procedure, since it is often more convenient to indicate one of the fuels thermal input percentage (δ) instead of the mass fuel ratio (n_d). Therefore, it is here proposed a novel alternative strategy to obtain Π_d without the need to use eq. 5-25, by assuming the value of δ as an input (ranges between 0 and 1). Moreover, this arrangement facilitates the use of the method in multi-fuel systems, as described later. In the case of a dual-fuel plant, Π_d can be calculated through eq. 5-29 in replacement of eq. 5-28:

$$\Pi_d = \delta \; \Pi_{f1} + (1 - \delta) \; \Pi_{f2} \tag{5-29}$$

where Π_{fl} is the pollution indicator of fuel l. For instance, if δ is the thermal input percentage of fuel 1, it is sufficient to know that the definition of δ is:

$$\delta = \frac{Q_{f1}}{Q_{f1} + Q_{f2}} \tag{5-30}$$

where $Q_{fl} = \dot{m}_{fl} \cdot LHV_{fl}$ and $\dot{m}_{fl}$ is the mass flow of fuel l, which does not need to be calculated or assumed, because δ is assumed as an input variable. Both eqs. 5-28 and 5-29 should give the same value of the energy-ecological efficiency if using proper values of n_d and δ. Hence, generalizing for multi-fuel plants gives:

$$\Pi_L = \sum_{l=1}^{L} \Pi_{fl} \cdot \delta_l \tag{5-31}$$

$$\delta_l = Q_{fl} \Big/ \sum_{l=1}^{L} Q_{fl} \tag{5-32}$$

where Π_L is the pollution indicator of the multi-fuel plant, L is the total number of different fuels for which $\sum_{l=1}^{L} \delta_l = 1$.

5.3.6 Accounting for biogenic carbon offset

In this section it is investigated for the first time how the EE method is influenced by the consideration of CO_2 uptake due to biomass regrowth regarding MSW burn. The main advantage of using vegetable biomass as fuel instead of fossil fuels is that, in terms of life cycle, plants absorb CO_2 during their growth. In other words, it can be recognized that the *CO_2 emitted from biofuel burn may not increase the total atmospheric CO_2 concentration* because emissions may be offset by the uptake of carbon due to regrowing biomass. Such carbon that *does not* contribute to increase atmospheric carbon concentration is called "biogenic", whereas the other type is called "antropogenic" or "fossil". It should be highlighted that *avoided emissions* in general are outside the scope of this work. In the particular case of MSW, it means that it is not considered here that burning MSW could be preventing methane emission if MSW were landfilled (instead of burned). This is because, again, the fuel's life cycle is not considered by the original method, which means that the analysis presented in this chapter includes accounting only for carbon emission *during combustion*, being it fossil or biogenic + fossil (total).

However, as mentioned earlier, other authors used different approaches such as applying the EE method considering the fuel's life cycle, as in the case of fluidized bed gasification of sugar cane bagasse investigated by Filho et al. [85]. They consider that approximately 81% of CO_2 emitted is biogenic, i.e. 19% is fossil carbon. Obviously, this is only possible if one does take into account emissions from the life cycle, since bagasse is 100% biogenic. EPA reports some emission factors for bagasse burn in sugar mills (tab. 5.5), where data is available for only half of the considered pollutants. Applying 81% reduction to the emission factor of CO_2 showed in tab. 5.5 results in $632\,\mathrm{kg}_{\,(\mathrm{bioCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{bagasse})}$, which means the emission factor with the accounting for CO_2 offset would be about $148\,\mathrm{kg}_{\,(\mathrm{fossilCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{bagasse})}$ instead of $780\,\mathrm{kg}_{\,(\mathrm{totCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{bagasse})}$. An evaluation of the EE value as a function of the thermal efficiency for bagasse burn ($LHV_{bagasse} = 7.32\,\mathrm{MJ\,kg^{-1}}$), calculated using approach III and data from tab. 5.5, is carried out and results are shown in fig. 5.13. Anticipating briefly the discussion presented later in section 5.4, it can be observed in such figure that the consideration of CO_2 offset results in higher values of ε, i.e. a better performance of the plant, as expected.

Similar logics can be applied to MSW, since it is partially composed of biomass. Because MSW composition may vary significantly it is not straightforward to estimate the biogenic carbon content of waste. Moreover, the type of carbon contained in MSW is also determinant to the value of LHV_{MSW},

Table 5.5: Emission factors[1] for sugar cane bagasse-fired boilers in $\mathrm{kg\,t^{-1}}$ [119].

Pollutant	Uncontrolled	Controlled[2]
CO_2	780.0	-
SO_2	-	-
NO_x	0.6	-
PM	7.8	0.7-4.2
CH_4	-	-
N_2O	-	-

[1] Units are $\mathrm{kg\,t^{-1}}$ of wet as-fired bagasse containing $\simeq$ 50% moisture by weight, assuming $1\,\mathrm{kg_{bagasse}}$ produces $2\,\mathrm{kg_{steam}}$. Average $LHV_{bagasse} = 8141\,\mathrm{kJ\,kg^{-1}}$, wet as fired.
[2] Lower value regards wet scrubber control and higher value regards mechanical collector.

i.e. the greater the fossil carbon content (more plastics), the greater the LHV. Ryu [120] states that 51% of the carbon content in MSW can be considered biogenic for a $LHV_{MSW} = 12.99\,\mathrm{MJ\,kg^{-1}}$ (total carbon content of 32.6%), while Giugliano et al. [111] suggest a biogenic carbon content of 71% for a $LHV_{MSW} = 10.5\,\mathrm{MJ\,kg^{-1}}$ (total carbon content of 23.7%). Since the CO_2 emission factor in tab. 5.2 ($985\,\mathrm{kg}_{\,(\mathrm{totCO_2})}\,\mathrm{t}^{-1}_{\,(\mathrm{MSW})}$) is also referenced to $LHV_{MSW} \simeq 10.5\,\mathrm{MJ\,kg^{-1}}$, to estimate the fossil emission factor of MSW burn it is sufficient to multiply 985 by 0.29, giving $285.65\,\mathrm{kg}_{\,(\mathrm{fossilCO_2})}\,\mathrm{t}^{-1}_{\,(\mathrm{MSW})}$. Therefore, 0.71 can be seen as a “reduction factor” to CO_2 emissions in this case.

In this sense, a comparison can be made between accounting for biogenic carbon offset and the formulation described in approach II (section 5.3.2), which introduced the reduction factors σ_s and σ_n to account for controlled emissions of SO_2 and NO_x. That is, stating σ_c as the reduction factor that can be applied to f_{CO2} in eq. 5-14, gives:

$$f_{CO2eq} = (1-\sigma_c)\cdot f_{CO2} + 700\,(1-\sigma_s)\cdot f_{SO2} + 1000\,(1-\sigma_n)\cdot f_{NOx} \tag{5-33}$$

Hence, a general formulation for the total equivalent emission factor (f_{eq}) that includes the possibility of considering or not the emission control of all pollutants in whichever approach is:

$$f_{eq} = \sum_{k=1}^{m}(1-\sigma_k)\cdot a_k \cdot \left\langle f_{pol,k}\right\rangle_{uncontrolled} \tag{5-34}$$

$$= \sum_{k=1}^{m} a_k \cdot \left\langle f_{pol,k}\right\rangle_{controlled} \tag{5-35}$$

where σ_k is the reduction factor of pollutant k, $f_{pol,k}$ is the emission factor of pollutant k, a_k is coefficient used in the equivalence formulation of the chosen approach and m is the total number of pollutants. Obviously, if one desires to employ the EE method to plants without any pollution control it is sufficient to use $\sigma_k = 0$ in eq. 5-34.

5.4
Results and discussion

This section presents the results from the application of approaches I-IV to three types of thermopower plants "WTE", "GT" and "WTE-GT", identified as cases A, B and C, respectively:

A. Waste-to-Energy plant (WTE): is a conventional incineration plant (Rankine cycle) fueled by MSW combusted in a grate furnace;

B. Gas turbine (GT): is a conventional gas turbine system, i.e. compressor, NG combustor and gas turbine.

C. Hybrid waste-to-energy (WTE-GT): is a combined cycle composed of a MSW incineration plant (bottoming cycle) integrated to a GT (topping cycle) through a heat recovery steam generator (HRSG), i.e. with MSW and NG burned in separate combustors.

In addition, the influence of biogenic carbon offset, as described in section 5.3.6, is also investigated to systems A and C, identified as sub-cases $\mathrm{A_{off}}$ and $\mathrm{C_{off}}$, where it is assumed $\sigma_c = 51\%$ as a reference for the CO_2 reduction factor (offset) of MSW burn.

Recapitulating the four approaches described in section 5.3 and their status with respect to emission control:

I. Approach I is Cardu's method (without any emission control)

II. Approach II is Cardu's method (with emission control)

III. Approach III is Villela's method (unclear)

IV. Approach IV is the new strategy (with/without emission control)

For all approaches, emission factors are obtained from tab. 5.2 and converted using eq. 5-15. Approaches I-III make use of uncontrolled emission factors and approach IV uses either uncontrolled or controlled.

Other specific assumptions based on the literature and in the author's knowledge are:

a. $LHV_{MSW} = 12.622\,\mathrm{MJ\,kg^{-1}}$ is assumed as a reference for cases A and C.

b. $LHV_{NG} = 47.141\,\mathrm{MJ/kg}$, $HHV_{NG} = 52.225\,\mathrm{MJ/kg}$, $v_{NG} = 1.405\,\mathrm{m^3/kg}$ @ $T = 25\,°\mathrm{C}$ and $P = 101.325\,\mathrm{kPa}$ [121].

c. $\sigma_{n,MSW} = 0.45$ (in tab. 5.2) is the reference DeNO$_x$ reduction factor of MSW burn [107].

d. $\sigma_{n,NG} = 0.69$ is the reference DeNO$_x$ reduction factor of NG burn [108].

e. η_{WTE} range is 22-30% for plants producing only electricity and 30-60% for CHP [49]; where it is assumed $\eta_{WTE} = 26\%$ as a reference for case A.

f. $\eta_{NG} = 45\%$ is assumed as a reference for case B [5].

g. η_{WTE-GT} range is 30-45% for plants producing only electricity and 30-60% for CHP; where it is assumed $\eta_{WTE-GT} = 38.1\%$ as a reference for case C.

h. $n_d = 0.2361$ or $\delta = 51.6\%$ are assumed as references for case C.

5.4.1
Case A: Single-fueled waste-to-energy plants

The results of case A are shown in fig. 5.3. As mentioned from the EE concept, the EE values differ according to the thermal efficiency and emissions. For the same η, higher emissions result in lower ε. On the other hand, if η increases it means the plant has a lower impact on the environment due to fuel misuse, being expected a higher ε. Indeed, this is shown in fig. 5.3 where, considering fixed the other parameters, it can be observed that by triplicating η, ε increases about 1.3 times to approaches I-III and 1.7 times to approach IV. It can also be observed that approach III is the one presenting the highest values of EE, as predicted in section 5.3.3, the EE value estimated through such approach is indeed more similar to approach II. Comparing approaches I and II it is noted that applying DeNOx and desulfurization procedures causes a significant increase of about 11 percentage points (p.p.) in ε. As observed, approach IV is the most conservative, presenting the lowest EE estimates of all, where ε differs of about 6 p.p. when emission control is considered compared to uncontrolled emissions.

From combustion chemical reactions, it is expected that by increasing the hydrocarbon heating value the CO_2 emissions would increase as well, because a greater number of carbons in the hydrocarbon molecule generates proportionally a higher number of CO_2 moles. However, since Π is the ratio between f_{eq} and LHV, whenever both f_{eq} and LHV increase by the same amount Π stays constant. This is observed in fig. 5.4, which shows the influence of MSW LHV on Π, ε, f_{CO2eq} and $f_{1,4DCBeq}$ calculated through approach IV. Therefore, for a fixed thermal efficiency (26%), increasing LHV of MSW results in constant EE of the single-fueled WTE plant, because even though f_{eq}

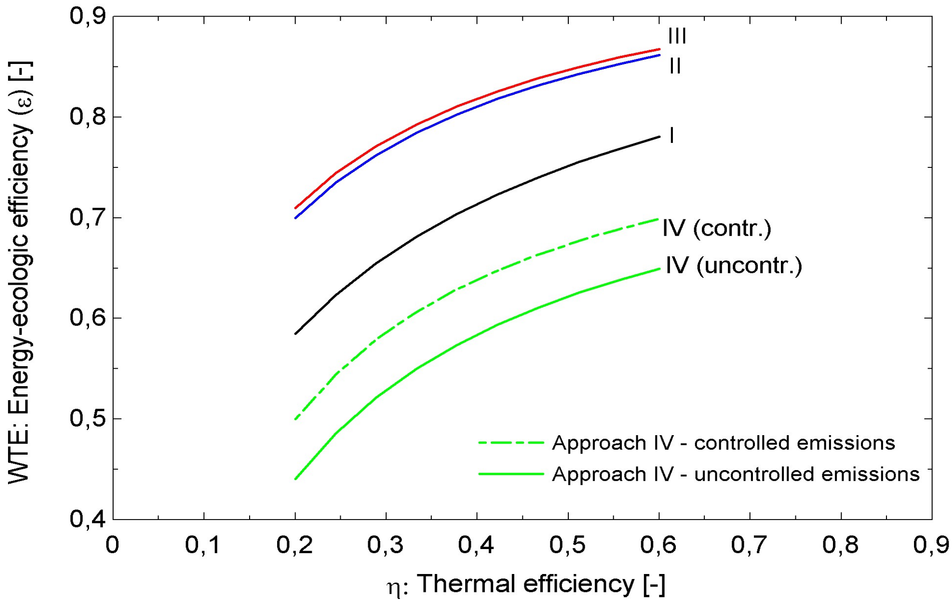


Figure 5.3: EE of WTE plant (case A) as a function of η through all approaches.

increases per unit mass of fuel it is constant per unit energy of fuel. Another observation regards the difference in the curves of f_{CO2eq} and $f_{1,4DCBeq}$; which increase at different rates.

Figure 5.5 compares the cases A and A_{off} under the influence of η. It can be noted that when carbon offset is considered, ε is greater, as expected. Approach IV is the most affected by the CO_2 offset. ε increases by 2 p.p. in approach I, about 5 p.p. in approaches II and III and more than 15 p.p. in approach IV when comparing cases with and without carbon offset. All approaches present a smaller influence of CO_2 offset on EE with η, meaning that discounting the biogenic carbon in a plant with a high thermal efficiency causes less impact on its environmental performance than it does on a plant with low η. This makes sense, since the worse the situation of a system, the greater its potential for improvement.

Another important conclusion from observing the EE values obtained through approach IV is that, for all variables fixed, accounting for biogenic carbon causes a greater increase in ε_{WTE} than applying emission control. This can be easily observed by comparing the EE values in figs 5.3 and 5.5. From tab. 5.6, which shows the results for the reference cases, the upgrade in ε_{WTE} is:

– 5.7 p.p. when considering emission control compared to null control;

– 15.4 p.p. when considering biogenic carbon offset compared to null carbon offset;

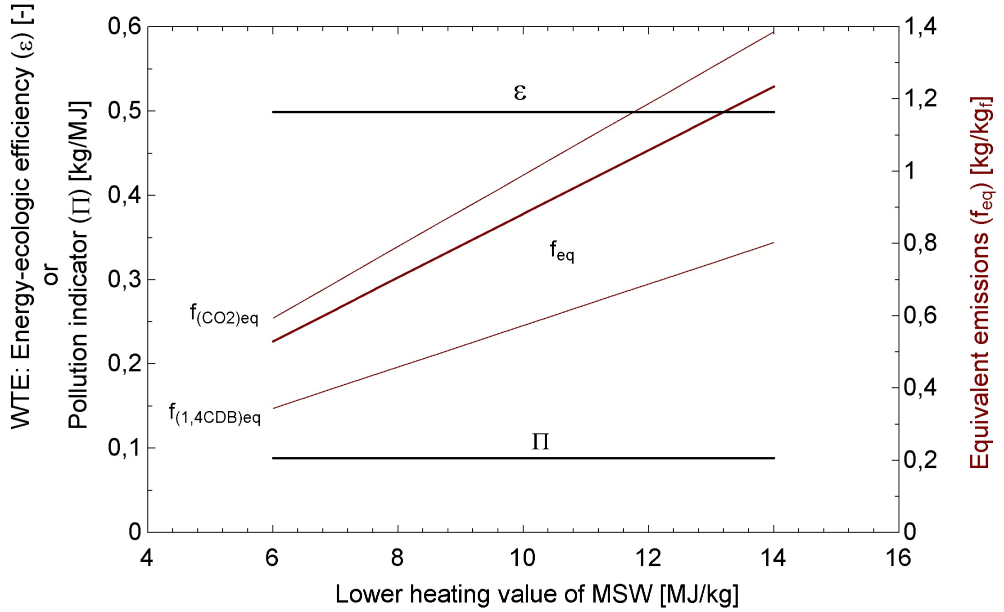


Figure 5.4: Influence of LHV_{MSW} on EE parameters of case A through approach IV.

– 23.3 p.p. when considering emission control + carbon offset compared to null control + null carbon offset.

Hence, for improving the EE performance of WTE plant, is more effective to account for the biogenic carbon offset than to apply emission control of the considered pollutants. Obviously, the EE value is the highest when accounting for both carbon offset and emission control, which for the reference case A is $\varepsilon_{WTE} = 78\%$.

5.4.2
Case B: Gas turbines

The results of case B are shown in fig. 5.6. A similar pattern as noted in case A regarding the influence of η on EE is repeated here (EE increases with η), however some differences can be highlighted. Considering approach IV it can be noticed that emission control influences very little the EE value, meaning that applying DeNOx procedures represents little gain to the plant's environmental performance. This is actually interesting because as known (also demonstrated in fig. 5.1), NG emissions are intrinsically low compared to the other fossil fuels, which does not justify significant effort in abatement. This means that improving the performance of the best existing fuel is a harder task than improving the performance of a worse fuel such as MSW. On the contrary, when comparing approaches I and II, it is observed that applying

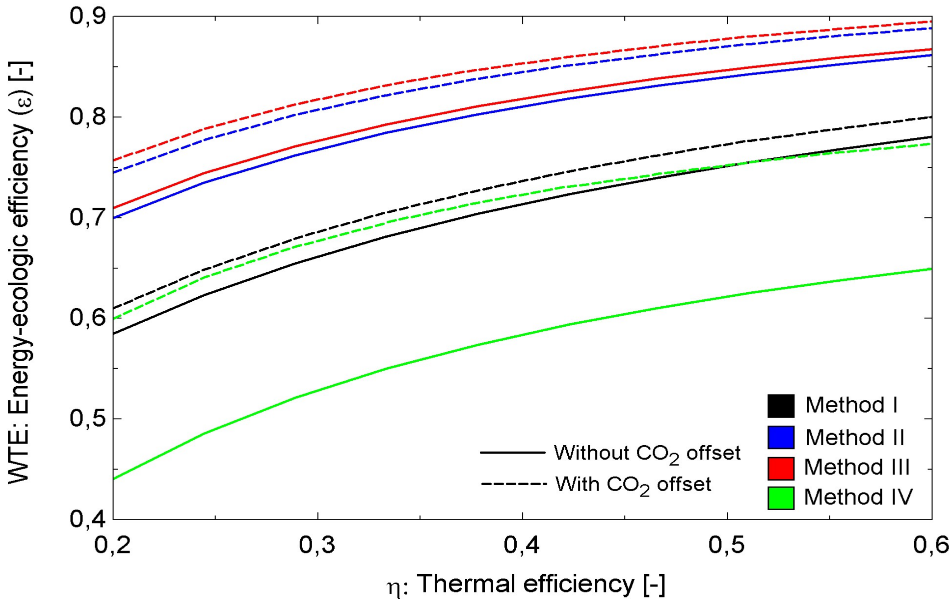


Figure 5.5: EE of WTE plant (case A) as a function of η through all approaches - with and without CO_2 offset.

DeNOx procedures causes a significant increase in ε of about 10 p.p. Moreover, ε obtained through approaches II and III differ by 4.5 p.p., which is much larger than observed in case A (less than 1 p.p.).

5.4.3 Case C: Hybrid waste-to-energy plants

The results of case C are shown in fig. 5.7, which depicts the influence of η on ε considering fixed the other parameters. It is observed that doubling η makes ε increase a little more than 1.1 times (less than case A, with 1.3). Discriminating the EE increase with η for each approach, from fig. 5.7 one has: about 11 p.p. for approaches I and IV-uncontr. and 8 p.p. for approaches II and III. When comparing approach IV with and without emission control, a difference of 4 p.p. is observed for the lowest η and 3 p.p. for the highest η. Thus, an interesting conclusion is that ε increases less with η for the approaches assuming emission control (methods II, III and IV-contr.) than when no control is applied (method I and IV-uncontr.). It means that, increasing the thermal efficiency of a plant that already performs some sort of pollution control causes less impact on its energy-ecologic performance than it does on a plant that does not apply any pollution control measures. Alternatively, it signifies that making an effort to control emissions in a plant with high thermal efficiency causes less impact on its environmental performance than it does on a plant with lower η.

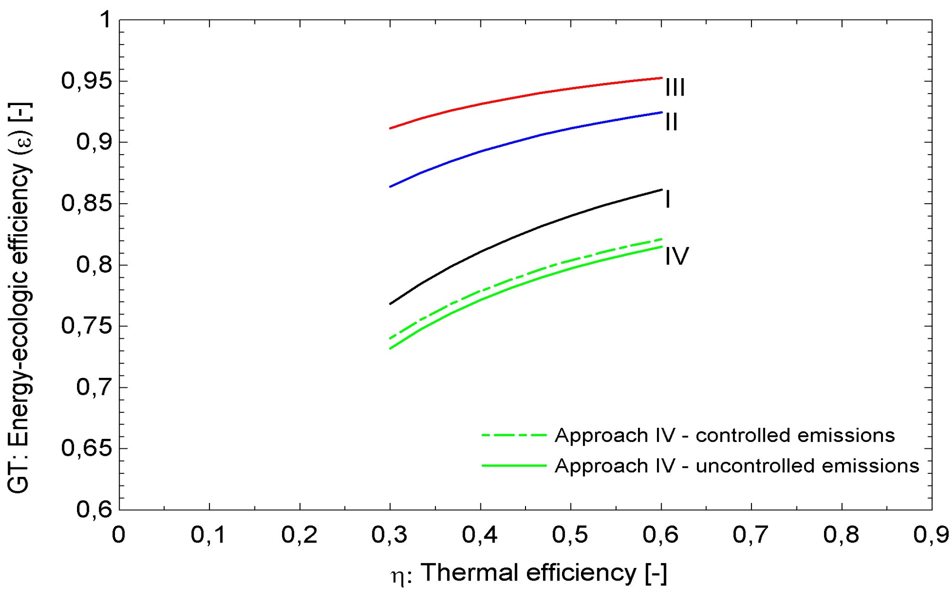


Figure 5.6: EE of GT plant (case B) as a function of η through all approaches.

This explains the observed differences between cases A and C. That is, since case C has already a better ε than case A its potential for improvement through the increase of η is expected to be lower than case A. Once again those results make sense because, as also noted from cases A and B, the worse the situation of a system, the greater its potential for improvement.

Considering the primary fuel as MSW and the supplementary fuel as NG, the LHV of the "hybrid fuel" is calculated through eq. 5-25, resulting in $LHV_d = 23.75$ MJ/kg; as expected $LHV_{MSW} < LHV_d < LHV_{NG}$. Figure 5.8 depicts the influence of MSW LHV on f_{eq}, Π and ε, while the other parameters are considered to be fixed, and the evaluated interval corresponds to LHV_d varying between 17 and 25 MJ/kg. As known, increasing the LHV of a fuel (higher carbon content) causes its emissions (f_{eq}) to increase (more CO_2 is generated). Since Π is a ratio between f_{eq} and LHV, it may increase or stay constant when LHV increases. In case C the fuel is "hybrid", hence LHV_d is what counts. In this case, f_{eq} and LHV_d increase at different rates causing Π to increase as well (in case A Π is constant with $\mathrm{LHV_{MSW}}$). It means more pollution is generated per unit energy of the "hybrid fuel", making the energy-ecologic efficiency to decrease. Another interesting observation from fig. 5.8 is that f_{CO2eq} and $f_{1,4DCBeq}$ increase at different rates, with the first presenting a larger derivative than the second. The effect of such behavior will be explained further in the text.

Figure 5.9 shows the influence of ε with η for cases C and $\mathrm{C_{off}}$ for all

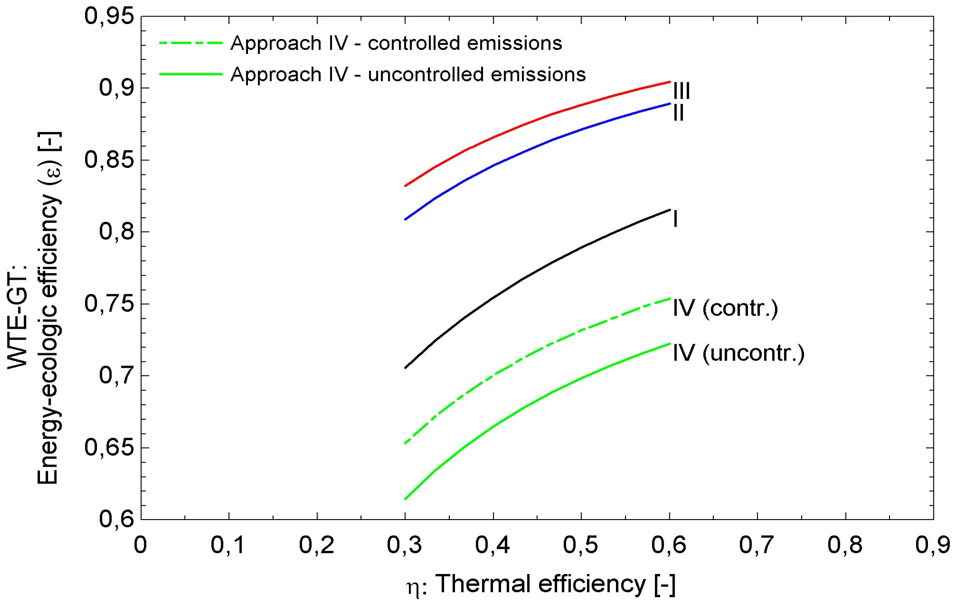


Figure 5.7: EE of WTE-GT (case C) as function of η through all approaches.

approaches. Two coincidences are: i) case C through approach III coincides with case C_{off} through approach II; ii) case C through approach I coincides with case C_{off} through approach IV for η tending to 30%. This leads to an interesting result that for η_{WTE-GT}<33%, approach I and approach IV with carbon offset present very similar values of ε_{WTE-GT}. This is unexpected since both approaches present very different arrangements. Another interesting result is that approach IV presents the largest difference in the value of EE with and without CO_2 offset, 7-9 p.p., while the other approaches present 1-3 p.p. In addition, it can also be noted from fig. 5.9 (and also in fig. 5.11) that the curves built with approaches I-III also present a slight approximation for greater values of the x variables. An explanation for such curves presenting a much more prominent approximation (even crossing in fig. 5.11) when built with approach IV is due to the characteristics of the method. This becomes clearer by observing the plots of f_{CO2eq} and $f_{1,4DCBeq}$ in fig. 5.8. The fact that f_{CO2eq} derivative is almost 2 times greater than $f_{1,4DCBeq}$ indicates that a reduction in carbon emissions (affecting only f_{CO2eq}) results in a greater impact on f_{eq} of approach IV (which reflects on ε) than it does on f_{eq} of the other approaches (which do not have such different “weights” influencing f_{eq}).

Figure 5.10 shows ε_{WTE-GT} calculated through approach IV under the influence of NG/MSW mass ratio, with all other parameters fixed. Three curves are shown: with and without emission abatement and considering uncontrolled emissions with biogenic carbon offset. As observed, increasing n_d 2.5 times

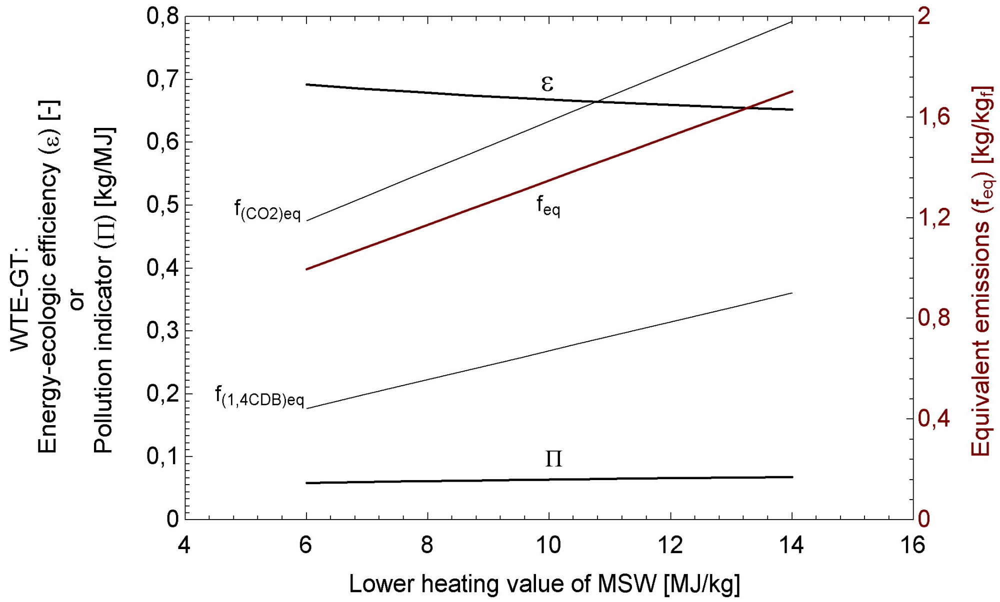


Figure 5.8: Influence of LHV_{MSW} on EE parameters of case C through approach IV.

causes the EE performance to increase 4.3 p.p. for uncontrolled emissions, 3.2 p.p. for controlled emissions and 1.1 p.p. for the scenario with carbon offset. This leads us to conclude that increasing the amount of NG, maintaining fixed the MSW capacity, causes a greater effect in plants without any flue gas treatment than in plants with emission control or that consider CO_2 offset. In addition, as concluded also in case A, considering the CO_2 offset of biogenic carbon in WTE-GT is more effective in improving ε than applying emission control. From tab. 5.6, which shows the results for the reference cases, the upgrade in ε_{WTE-GT} according to approach IV is:

- 3.6 p.p. when considering emission control compared to null control;
- 8.2 p.p. when considering biogenic carbon offset compared to null control + null carbon offset;
- 12.3 p.p. when considering emission control + carbon offset compared to null control + null carbon offset.

Hence, for improving the EE performance of WTE-GT plant, is more effective to account for the biogenic carbon offset than to apply emission control of the considered pollutants. Obviously, the EE value is the highest when accounting for both carbon offset and emission control, which for the reference case A is $\varepsilon_{WTE} = 78\%$.

Finally, in the specific cases of dual-fuel, it would make sense to take into account that whenever the LHV of one of the fuels increases, η and/or n_d

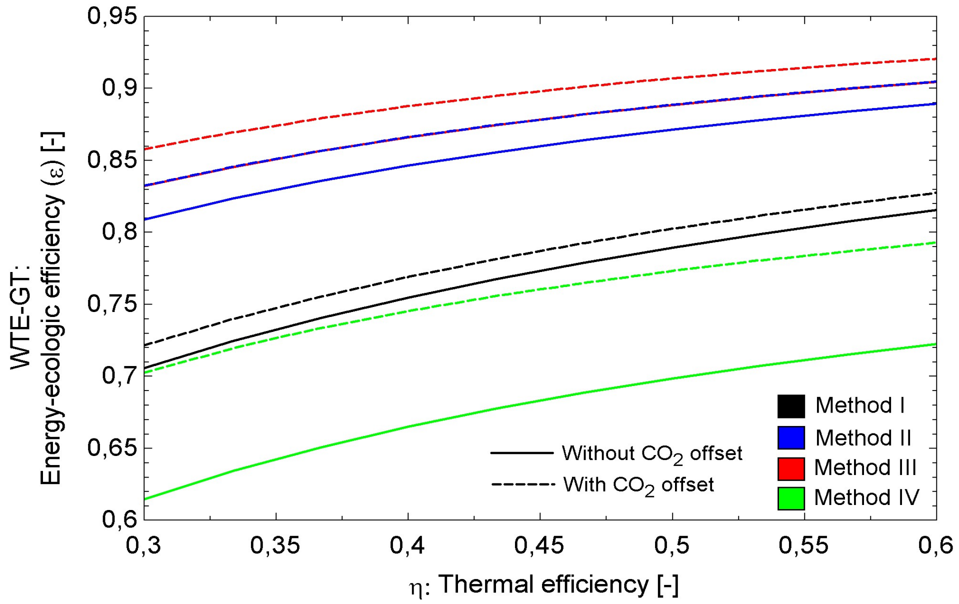


Figure 5.9: EE of WTE-GT (case C & C$_{\text{off}}$) as function of η through all approaches

could vary as well. Such analysis is outside the scope of this work, but a future interesting study would be to consider simultaneous variations of η and n_d.

5.4.4
Ranking Waste-to-Energy plants

Table 5.6 presents the values of f_{eq}, Π and ε obtained for the reference scenarios of cases A, A$_{\text{off}}$, C and C$_{\text{off}}$. It can be noted that the EE performance of the single-fueled WTE is 12-24% worse than the WTE-GT, depending on the approach. Approach IV is the one presenting the highest difference of performances between cases A and C (ε differs of 15.7 p.p.) with WTE-GT performing 24% better than WTE. Another interesting observation is to compare ε of cases A-contr. and C-uncontr., or alternativelly A$_{\text{off}}$-contr. and C$_{\text{off}}$-uncontr. In both comparisons $\varepsilon_{WTE-GT} > \varepsilon_{WTE}$, which is a great advantage for the hybrid technology because no emission control would be needed for this plant and it would still have a better environmental performance than the single fueled plant. Of course, this does not take into account the need to comply with emission regulations, which is the main reason for applying emission control in the first place. Morevover, even more similar EE performances are noted between cases A$_{\text{off}}$ and C, both without emission control; meaning that operating a WTE-GT even disregarding the carbon offset is better than operating a WTE accounting for the carbon offset.

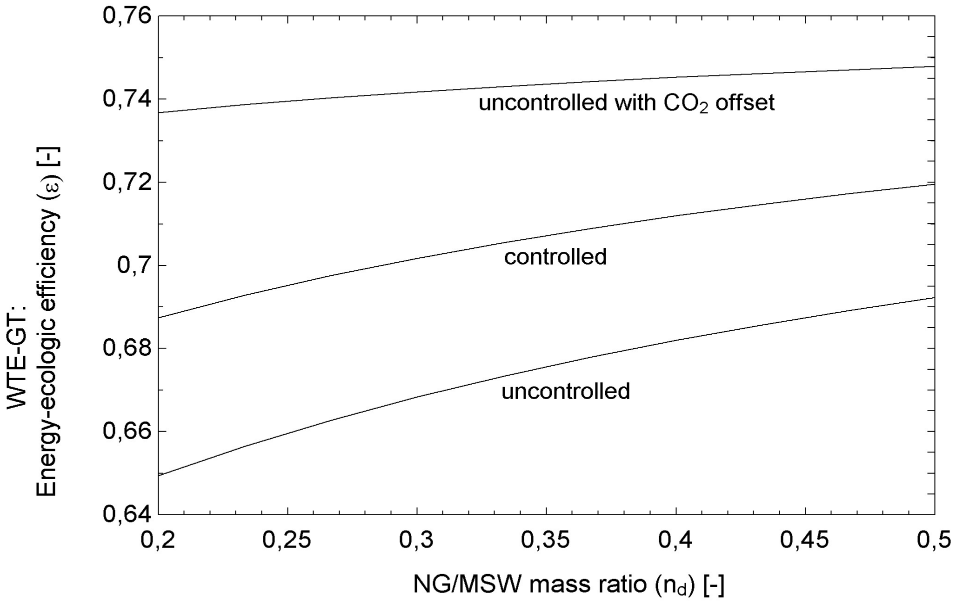


Figure 5.10: Influence of n_d on EE of WTE-GT with and without emission control/CO_2 offset through approach IV

Table 5.6: Main parameters of the EE method obtained through approaches I-IV applied to the reference cases A, A_{off}, C and C_{off}.

	f_{eq}[-]				Π [kg/MJ]				ε [%]			
Method	A	A_{off}	C	C_{off}	A	A_{off}	C	C_{off}	A	A_{off}	C	C_{off}
I	4.86	4.25	7.16	6.55	3.85×10^{-1}	3.37×10^{-1}	3.01×10^{-1}	2.76×10^{-1}	63.49	65.99	74.71	76.14
II	3.53	2.93	4.81	4.20	2.80×10^{-1}	2.32×10^{-1}	2.02×10^{-1}	1.77×10^{-1}	69.39	72.70	80.81	82.63
III	2.48	1.88	3.16	2.56	1.97×10^{-1}	1.49×10^{-1}	1.33×10^{-1}	1.08×10^{-1}	75.44	79.75	86.08	88.30
IV-uncontr.	1.11	0.66	1.58	1.13	8.79×10^{-2}	5.23×10^{-2}	6.65×10^{-2}	4.76×10^{-2}	50.00	65.39	65.70	73.88
IV-contr.	0.93	0.48	1.37	0.92	7.33×10^{-2}	3.77×10^{-2}	5.79×10^{-2}	3.89×10^{-2}	55.74	73.34	69.32	78.00

Figure 5.11 shows how ε is influenced by the fossil carbon emission factor ($\mathrm{kg}_{\,(\mathrm{fossilCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{MSW})}$) of MSW burn in cases A and C. A general range for the biogenic carbon reduction factor (σ_c) of 0-100% is adopted, where the extreme case of MSW composed exclusively by biogenic components (e.g. food, vegetables, plants) gives $0\,\mathrm{kg}_{\,(\mathrm{fossilCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{MSW})}$ ($\sigma_c = 100\%$). It can be observed from such figure that ε increases with σ_c, as expected, meaning that a smaller offset of biogenic emissions results in higher fossil emissions and lower EE performance of the system. For approaches I-III ε_{WTE-GT} is always greater than ε_{WTE}, with the first being about 2 times less influenced by the CO_2 offset than the second. An interesting result is obtained from approach IV: when $(1-\sigma_c) \simeq 10\%$ it is observed for the first time a situation when $\varepsilon_{WTE} = \varepsilon_{WTE-GT}$. Moreover, when $(1-\sigma_c) < 10\%$ results in $\varepsilon_{WTE} > \varepsilon_{WTE-GT}$, denoting that in the extreme case when the WTE plant emits $0\,\mathrm{kg}_{\,(\mathrm{fossilCO_2})}\,\mathrm{t}^{-1}_{(\mathrm{MSW})}$ it is environmentally worse to have a WTE-GT

plant (always emits non null fossil carbon) instead. It is worth mentioning one more time that such results do not take into account the whole life cycle of the fuels, which should also be evaluated in order to reach deeper conclusions about the advantage of running a hybrid plant with biomass 100% biogenic over a single-fueled one.

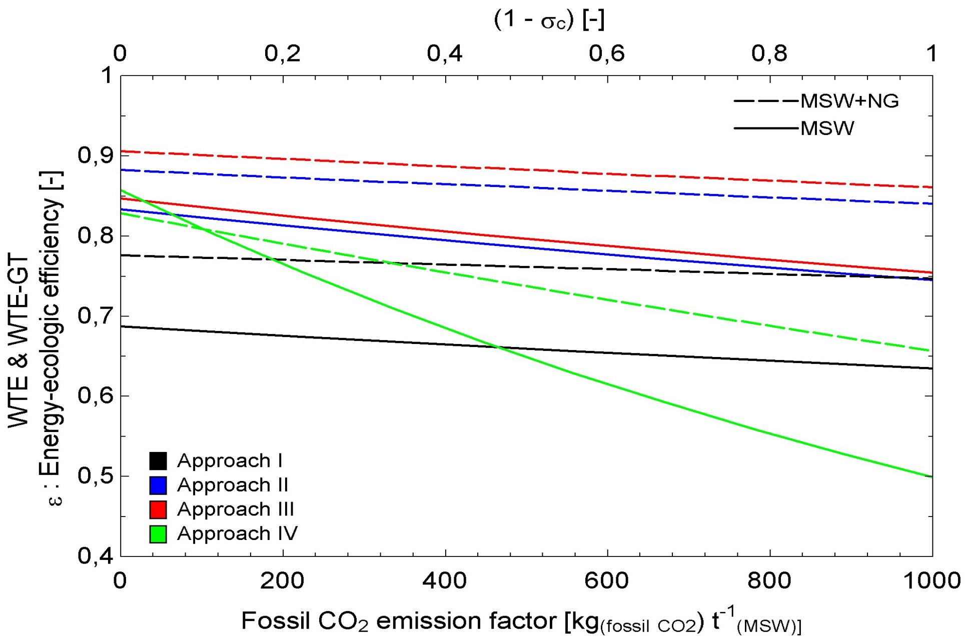


Figure 5.11: EE of MSW and MSW+NG (cases A, C) as a function of fossil CO_2 emission factor.

Figure 5.12 shows the influence of thermal efficiency on the energy-ecologic efficiency of different fuels through approach I, where it can be observed that MSW curve is above all fossil fuels, except NG. The evaluated hybrid MSW+NG, which has about 52% of total thermal input derived from MSW, presents the second highest curve only after NG. This means that, fixing a value of the thermal efficiency for all fuels, MSW+NG presents a better performance than coal and oil, but a worse performance than both types of NG. The difference observed between the evaluated NG and Romanian NG is due to the LHV values (Romanian NG has LHV=35.7 MJ/kg). It can also be observed in fig. 5.12 that MSW+NG with η of 30% has ε equal to MSW with $\eta = 38\%$ or a paraffin-oil fired plant with $\eta = 57\%$. On the other hand, MSW with $\eta = 30\%$ presents the same ε as paraffin oil with $\eta = 45\%$ or as fuel-oil (with maximum 2% sulfur content) with $\eta = 69\%$.

Hence, it can be concluded from fig. 5.12 that for a fixed MSW/NG mass ratio:

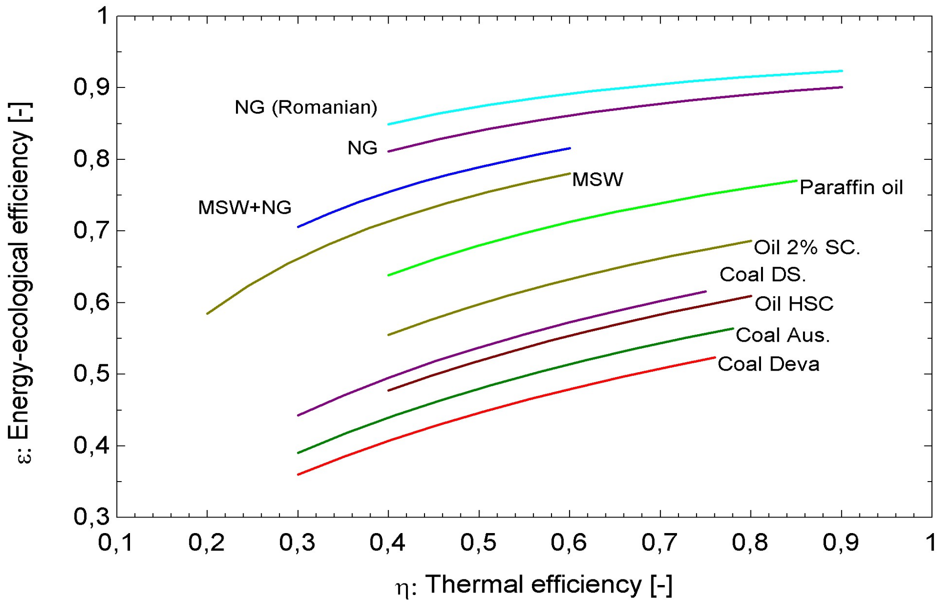


Figure 5.12: EE as a function of η for different fossil fuels through approach I. Coal/oil/NG Romanian from [97]. "Oil 2% SC": fuel-oil w/ max. 2% S content; "Coal DS.": coal (Doiceşti-Şotânga); "Coal Aus.": coal (Australia); "Oil HSC": fuel-oil w/ high S content.

– The EE performance of a WTE-GT with $\eta \geq 47\%$ will always overcome that of a single-fueled WTE plant with maximum achievable η of 60% (both operating as CHP systems).

– The only way EE of a single-fueled WTE plant can overcome a WTE-GT plant is if $\eta_{WTE} \geq 38\%$ and $\eta_{WTE/GT} < \eta_{WTE}$ in about 27%.

Figure 5.13 also shows the influence of η on ε of different fuels but considering approach III and additionally presenting the effect of CO_2 offset (when applicable). As it can be noticed, MSW performs worse than all other fuels even if accounting for carbon offset. Sugar cane bagasse curve is superior to MSW+NG only if considering carbon offset. The upgrade in biodiesel's performance is the most significant of all fuels when comparing emissions with and without carbon offset, which makes it the best fuel (approaching hydrogen) when considering only fossil emissions. Comparing the two renewable fuels, bagasse and biodiesel, it can be observed that the EE performance of biodiesel is much superior than bagasse, this is because even though both fuels are almost 100% composed of vegetable biomass, bagasse is a lower quality fuel (more emissions and lower LHV). Moreover, the only way Diesel and the evaluated case of MSW+NG present competitive EE performances is whenever Diesel has η=40% and MSW+NG has $\eta_{WTE-GT} > 54\%$ (without carbon offset) or $\eta_{WTE-GT} > 44\%$ (with carbon offset).

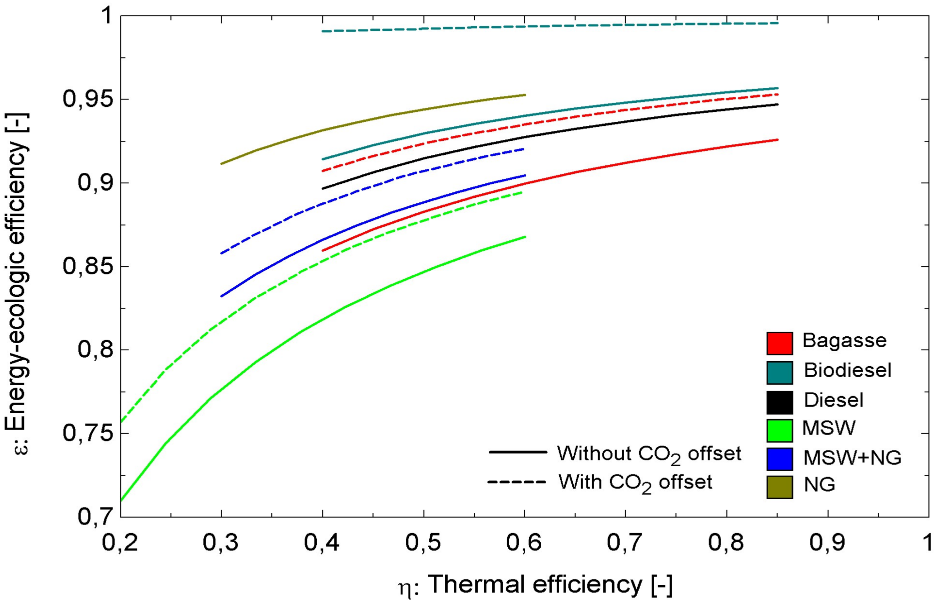


Figure 5.13: EE as a function of η for different fuels through approach III - renewables with and without CO_2 offset. Diesel/biodiesel from [31]. Sugar cane bagasse from [85].

5.5 Conclusive remarks and future work

This chapter resulted in an article[1] published in *Energy Conversion and Management* journal. Even though approached by several publications in peer-reviewed journals, the different arrangements to calculate the energy-ecologic efficiency have never been subject of a comprehensive technical review. It can be concluded that, even if they present weaknesses and strengths, the general strategy is certainly a useful tool for researchers to easily compare diverse energy conversion technologies through an uncomplicated technique. The main reservation identified in the existing approaches regards the merging of concepts involving the pollution in confined environments and ambient air. Therefore, it has been proposed a novel technique that keeps the original advantages of the method but eliminates such problem. It consists of evaluating the indicator in such a way that it accounts for the three key environmental damage potentials: global warming, human health and resource availability. A major outcome of this chapter is that the methodology has been proposed in

[1]https://doi.org/10.1016/j.enconman.2019.05.098

its most generalized form, describing the pathway to admit its application not only as presented in its final mold, but also considering as many pollutants as desired and employed to multi-fuel systems.

As unique findings of this chapter, MSW fired plants have been quantified in terms of their energy-ecological efficiency, demonstrating how significant are the differences between the existing strategies. Three cases were investigated: MSW burn, NG burn in gas turbines and a hybrid of both. The results from the proposed approach have shown to be consistent and very coherent with the context of real thermal systems. Additionally, the influences of pollutant abatement and biogenic CO_2 offset due to biomass regrowth have also been discussed for the first time for MSW conversion technologies. The results show that whenever the plant's biogenic emissions are discounted or emission control is applied, its potential for improvement in terms of the EE indicator is smaller compared to a plant without any of those reductions. It can also be concluded that the investigated cases of MSW-fired plants do not perform as well as bagasse, biodiesel nor Diesel, but perform better than all types of coal and oil. Finally, considering the possibility of combined heat and power, it can be concluded that, for the investigated cases, the only way a single-fueled WTE plant can overcome a WTE-GT in terms of EE is if the first has an energy efficiency of about 38% (CHP system) and the second operates with a thermal efficiency 27% smaller than the first.

This work can be continued by performing further sensitivity analysis, including the variation of two or more parameters simultaneously, for instance MSW/NG mass ratio & thermal efficiency in the hybrid case or waste's LHV & biogenic carbon offset in both MSW and MSW+NG cases. Other continuation could involve expanding the method to account for more pollutants such as carbon monoxide and dioxins or upgrading it to be able to account for the fuel's energy carriers (life cycle perspective).

6
Advanced hybrid waste-to-energy plants

This chapter presents the evaluation of energy, exergy, economic and environmental indicators of advanced hybrid waste-to-energy cycles modeled in Thermoflex®. Starting from the reference layout presented in chapter 3, several changes in the design of the combined cycle are applied in order to make it more realistic, increase its energy efficiency and waste share in the hybrid fuel. The fact that increasing efficiency is beneficial to save fuel costs by avoiding fuel wastage, increasing efficiency too much may not be worth because equipment costs might become excessively high. Thus, an optimal arrangement is the one presenting the best performance with acceptable costs and pollution. This explains why an integrated evaluation of thermodynamic, economic and environmental parameters must be proceeded. In this sense, the goal of this chapter is to propose an engineering design of an optimized large sized power plant fueled by municipal solid waste and natural gas considering energy, exergy, economic and environmental aspects. The levelized costs of electricity production and waste treatment are evaluated in the context of a developing country, where urban waste has low calorific power, landfilling costs are very low, electricity prices are variable and the weather is warm all year long.

6.1
Introduction

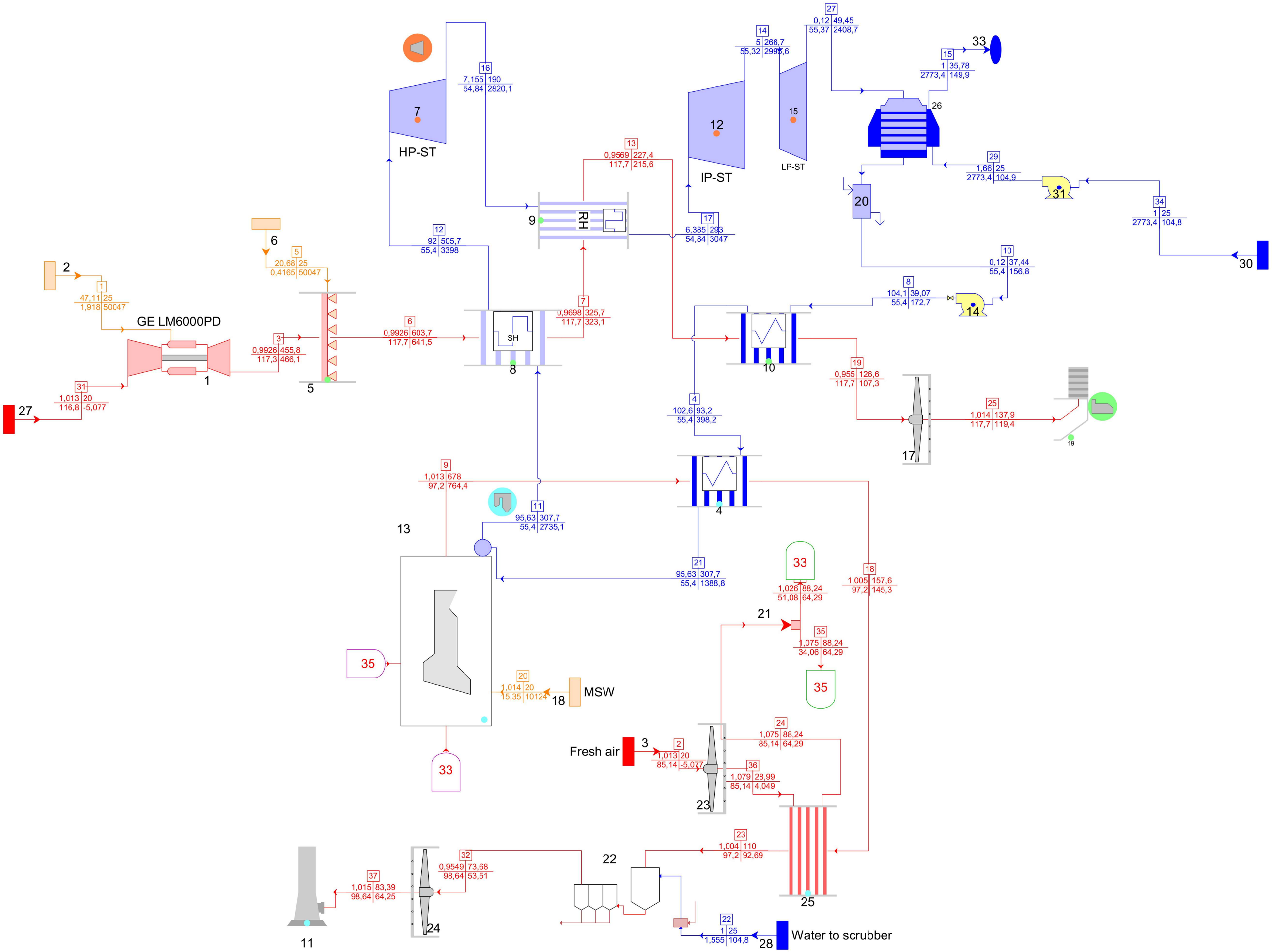
According to Leckner [13], in a combined heat and power plant without system limitations on the demand of heat, such as in large energy systems in cold countries, there is no need to make an effort to increase the electric power share of the energy production. The problem related to WTE is rather found in regions where there is insufficient or no heat demand, such as the case of Brazil. In such situations the efficiency of the electricity production should be promoted within economic restrictions. As mentioned in chapters 2 and 4, Thermoflex® (TFX) is a commercial software for simulation of complex thermodynamic cycles. It performs automatic calculations of mass and energy balances taking as inputs reference data from its library and user-added variables. The software facilitates tremendously simulations and engineering design of energy systems, specially sizing and costing. Besides that, it has the additional advantage of estimating also some pollutant emission. Nevertheless, it has some limitations in the chemical part (does not estimate NO_x emissions) and some problems have been related regarding economic estimates of large-sized incinerators. In summary, it is a closed software that gives thermodynamic outputs, efficiencies and purchase/installation cost. Parametric analyses are also possible by using the integration link with Excel®, although not as easily. In comparison, Engineering Equation Solver® (EES) is an equation solver with some thermodynamic functionalities that also presents optimization functions, with the additional advantage of facilitating parametric analysis and plot building. To model a thermodynamic power cycle in EES, the user is supposed to input all the mass and energy equations which are then solved by the software. Unlike TFX, EES requires the user to know several specific design parameters in order to accomplish sizing procedures. For instance, to model a heat exchanger and obtain its cost, Thermoflex® has several library data requiring only the user's inputs for temperature/pressure conditions. In Engineering Equation Solver® such detailed simulation would require many other inputs such as material type, tube roughness, number of tube passes, etc.

Another interesting advantage of TFX regards determination of gases dew points. Since water is formed when hydrocarbon fuels are burned, the mole fraction of water vapor in the gaseous products of combustion can be significant. If those are cooled at constant mixture pressure, the dew point temperature is reached when water vapor begins to condense. Since water deposited on duct work, mufflers, and other metal parts can cause corrosion, knowledge of the dew point temperature is important. TFX always alerts when a component of the flue gases is below its dew point. In this sense, the cycles should be designed with all flue gases temperatures above their dew point, which by the way, constitutes an important limitation to the plant's performance. Since we are interested in modeling engineering designs as realistic as possible, Thermoflex® is used in the so called "engineering design mode", a more sophisticated simulation process that takes into account pressure losses, leaks/blowdowns and internal consumptions of the main equipment. In particular, the WTE boiler simulation in TFX can be, in theory, much more accurate than in EES because it takes into account most of the processes occurring in real life. It considers the possibility of adding two air flows to simulate combustion, one for the primary air (under grate) and other for the secondary air (over grate), and a third inlet to simulate the recirculation of flue gases. In addition, it allows to simulate grate cooling, blowdown of the evaporator and the effect of partial screening of the radiant flux to waterwall due to fouling (dirtiness) inside the furnace.

Both software have been used in the development of this thesis. In particular, Thermoflex® access was limited to a small period while EES access was unlimited. Since, a priori, EES modeling requires more strict simplification hypotheses than TFX, it is interesting to know if the hypotheses assumed in the EES simulation cause significant impact on the results compared to simulation in TFX. An ideal validation of the EES/TFX models would require access to experimental data, which is not available. Thus, what can be verified, at least, is if the differences between simulations in these two software are significant. For this, the cycle shown in fig. 3.2 is modeled in both TFX and EES. TFX simulation is done first, as shown in the **next page**. EES model is built subsequently, using the same assumptions described in section 3.2.2 (chapter 3) and taking, as much as possible, the inputs from TFX (those that cannot be taken from TFX are assumed from literature). The resulting properties are compared in tab. 6.1.

Table 6.1: Properties of the reference layout modeled in both software: Thermoflex® and Engineering Equation Solver®

			Thermoflex®				**Engineering Equation Solver®**			
Point	**Description**	**Fluid**	**T (°C)**	**P (bar)**	**x (-)**	**m (kg/s)**	**T (°C)**	**P (bar)**	**x (-)**	**m (kg/s)**
1	Compressor inlet	Dry air	20.0	1.00		116.8	20.0	1.00		116.8
2	Compressor outlet	Dry air					584.5	29.19		116.8
3	GT inlet	Flue gas					1248.8	29.19		118.7
4	GT outlet	Flue gas	455.8	0.99		117.3	461.7	0.99		118.7
5	Duct burner outlet	Flue gas	603.7	0.99		117.7	616	0.99		119.1
6	HRSG flue gas outlet	Flue gas	126.6	0.96		117.7	124.1	0.96		119.1
7	Condenser outlet	Water	37.4	0.12		55.4	38.0	0.12		55.4
8	Pump outlet	Water	39.1	104.1		55.4	39.1	104.1		55.4
9	Furnace ECO inlet	Water	93.2	102.6		55.4	88.1	102.6		55.4
9'	Evaporator inlet	Water	307.7	95.6	0.0	55.4	307.7	95.6	0.0	55.4
10	Superheater inlet	Water	307.7	95.6	1.0	55.4	307.7	95.6	1.0	55.4
11	HP-ST inlet	Water	505.7	92.0		55.4	510.0	92.0		55.4
12	HP-ST outlet	Water	190.0	7.16		54.8	197.0	7.16		55.4
13	LP-ST inlet	Water	293.0	6.39		54.8	293.0	6.39		55.4
14	LP-ST outlet	Water	49.5	0.12	0.92	55.4	49.5	0.12	0.92	55.4
15	Furnace outlet	Flue gas	157.6	1.00		97.2	157.6	1.00		85.1
16	Furnace inlet	Dry air	88.24	1.08		85.1	88.6	1.08		85.1
17	Env. Pol. Cont. inlet	Flue gas	110.0	1.00		97.2	107.4	1.00		100.5



bar | C
kg/s | kJ/kg
GE LM6000PD
HP-ST
IP-ST
LP-ST
RH
SH
MSW
Fresh air
Water to scrubber
1
2
3
5
6
7
8
9
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It can be observed that the most significant differences between the properties obtained in the two models are among the temperature values. The differences in gaseous streams can be explained mainly by the simplification hypotheses adopted in the EES model that consider flue gases composition identical to dry air, and dry air composed of N_2, O_2 and Ar (v. hypothesis iii in section 3.2.2). Since gaseous stream temperatures interfere in the cold side streams of heat exchangers, it is expected that temperatures of water/steam also differ between the two models. In fact, water/steam temperatures differ mainly in the heat exchangers of the heat recovery boiler (SH, RH, ECO)[1]. The differences between simulations of the WTE boiler in both models explain the observed temperature difference in point 17. On the other hand, differences in the mass flow values between the two models can be explained mainly by the fact that TFX considers a blow-off in the GT compressor and leakage in the ST, whereas those are not considered in the EES model (hypothesis xiii in section 3.2.2). Moreover, the observed difference in the mass flow of flue gases at furnace outlet is explained by the fact that TFX does consider ash outflow, contrarily to the EES model (v. hypothesis 7 in section 3.2.2). In particular, TFX considers the following weight composition of the waste ash: SiO_2 45%, Al_2O_3 26%, Fe_2O_3 8%, CaO 11%, MgO 2%, Na_2O 5%, K_2O 1%, SO_3 2%, TiO_2=P_2O_5 = 0%.

Table 6.2 shows the main energy results from the two models. It can be noticed that, despite of the observed contrasts, the main energy outputs are quite similar. It can also be noted that the main deviation is in the gross power output, which explains the difference also in the gross electric efficiency. The contrast between the two GT power outputs is due to TFX assuming a pressure drop at the air inlet filter of 9.963 mbar, a leakage/miscellaneous blow off of 1.404 kg/s, an internal consumption by GT miscellaneous auxiliaries of 84.98 kW and a GT generator efficiency of 98.48% (GT generator loss of 611.1 kW). All this aspects were neglected in the EES model. Moreover, despite of looking unlike in tab. 6.2, the internal consumption values appraised in the two models represent 7% and 8% of the total power output in TFX and EES, respectively. This shows that hypothesis vi (in section 3.2.2) is valid, but could be slightly adjusted to the case of hybrid plants.

As mentioned, this chapter aims at simulating diverse configurations of WTE-GT plants in TFX, having the above mentioned cycle as a reference, in order to achieve the following goals:

[1]Those differences were minimized not by using the exact same approach temperatures given by TFX, but rather choosing them in a way to minimize the differences between the models.

Table 6.2: Thermodynamic performance of the reference layout built with Thermoflex® and Engineering Equation Solver® .

Parameters	**TFX**	**EES**
Total fuel thermal input	272.34 MW_t	272.44 MW_t
NG fuel input	116.97 MW_t	117.04 MW_t
MSW fuel input	155.37 MW_t	155.40 MW_t
MSW th. input % (δ)	57%	57%
LHV MSW	10.12 MJ/kg	10.12 MW_t
LHV NG	50 MJ/kg	50 MJ/kg
Air/MSW fuel ratio	5.5	5.5
Gross power output	105.43 MW_e	106.95 MW_e
Net power output	98.49 MW_e	97.89 MW_e
Internal consumption	6.93 MW_e	8.29 MW_e
GT power output	39.49 MW_e	40.01 MW_e
ST power output	65.94 MW_e	66.85 MW_e
Gross electric efficiency	38.71 %	39.26 %
Net electric efficiency	36.17 %	35.93 %
GT LHV efficiency	41.78 %	41.77 %
Boiler efficiency	83.33 %	83.96 %

A. Obtain a more realistic design based on processes often applied in existing WTE plants and suggested by the specific literature.

B. Obtain the highest possible 1st Law efficiency (η_{tot}) producing electricity derived as much as possible from the waste, that is, with a maximum MSW thermal input percentage (δ).

C. Obtain an optimal configuration that is able to produce electricity at the lowest possible cost with acceptable environmental pollution.

For this, the reference layout was modified to include mainly a deaerator, air pre-heaters, regenerators, re-heaters, recirculation of MSW flue gases into the furnace, grate cooling and reuse of boilers' blowdown steam in the feedwater heaters. Then, an analysis of key energy, exergy, environmental and economic indicators allows to identify the best performing configuration. In addition, the possibility of using landfill gas as fuel in the supplementary firing HRSG is also investigated. This is the first time such comprehensive analysis is performed to advanced hybrid WTE-GT systems.

6.2 Method

Taking into account practical restrictions regarding WTE plants and the fact that the study aims at proposing an optimal arrangement that could operate in the context of developing countries with warm weather, such as Brazil, the following premises are assumed:

a. Temperature of the flue gases should not be greater than 160 °C at stack in order to avoid excessively small values of η.

b. Temperature of the flue gases should not be smaller than 115 °C nowhere in the cycle (except flue gas cleaning system outlet) to avoid condensation of acid gases or water.

c. Feedwater temperature should be $\geq$125 °C (sulfur dew point) before being introduced in the MSW furnace economizer in order to prevent corrosion.

d. Superheated steam temperature should be $\leq$450 °C in the WTE boiler in order to avoid corrosion problems.

e. Minimum pinch point temperature in the heat exchangers should be 10 °C in order to avoid excessively large heat transfer surfaces.

f. Oxygen content in MSW dry flue gases should not be smaller than 4% in order to guarantee complete combustion.

g. Considering the Brazilian context (hot weather), steam pressure at condenser inlet should not be smaller than 0.056 bar [9] for condensers operating with cooling towers, in order to avoid requiring excessively low temperatures of cooling water.

h. Considering the Brazilian context, the LHV of MSW should not be lower than 10.133 MJ/kg.

i. Considering the Brazilian context, the average ambient temperature should not be lower than 25 °C.

The methodology is structured in order to study some key aspects regarding the design and operation of WTE-GT plants:

1. Deaerator pressure;
2. Supplementary firing (duct burner);
3. Biogas use;

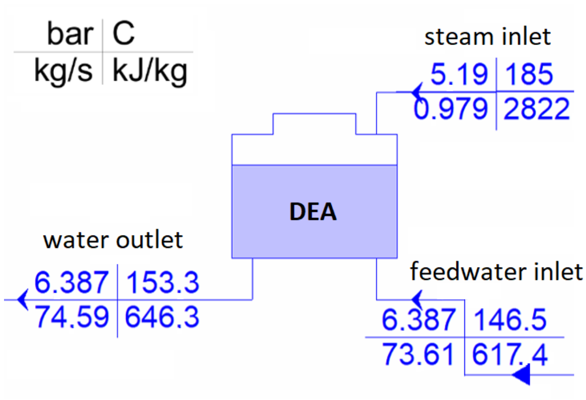


Figure 6.1: Deaerator example simulated in Thermoflex®

4. Gas turbine size;
5. Recirculation of MSW flue gases;

The motivation for investigating such aspects and the detailed methodology are presented in the following subsections.

6.2.1 Deaerator pressure

A first look at the reference layout aiming to identify possible changes towards a more realistic design reveals that a deaerator is missing. Deaerators are necessary for practical reasons. It is an equipment that aims at removing dissolved gases (mainly oxygen and CO_2) from the water (working fluid of steam cycles) up to tolerated levels in order to prevent steel corrosion in the pipes [122]. The motivation for investigating the influence of the deaerator pressure on the performance of hybrid WTE plants comes from experts' opinion that affirm typical values for DEA steam inlet pressure (P_{dea}) are around 5 bar. However, other references, including personal consultation, point out lower values for operating plants: WTE plant of Modena (Italy) has $P_{dea} \sim$ 2 bar; WTE-GT plant of Bilbao has P_{dea}=0.89 bar. Thermoflex® establishes a possible range for P_{dea} as 6.895 mbar to 34.47 bar. Figure 6.1 illustrates a deaerator simulation in TFX with P_{dea} around 5 bar. Given the fact that the steam flow feeding the deaerator comes from a turbine extraction, the choice of P_{dea} as 5 or 2.5 bar, for instance, requires the extraction to occur at different stages of the steam turbine, which may lead to modifications in the whole arrangement of the cycle. Hence, two cases are investigated: (i) P_{dea} ~5 bar, where steam is extracted from the high-pressure steam turbine (HP-ST); and (ii) P_{dea} ~3 bar, where steam is extracted from an intermediate stage, between the HP and LP-ST.

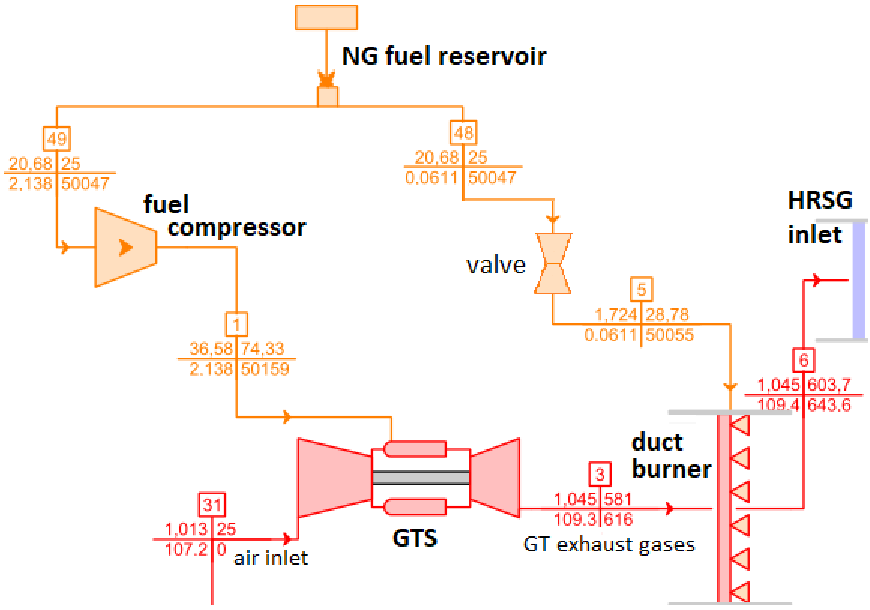


Figure 6.2: Example of a natural gas system fueling the duct burner and the gas turbine (GTS) simulated in Thermoflex®

6.2.2 Supplementary firing (duct burner)

The presence/absence of a supplementary firing device (or duct burner) at the HRSG inlet is also investigated. The motivation for this study is explained by a general opinion from experts in the field that suggest that duct burners are not necessary in hybrid WTE-GT plants. However, the plant of Bilbao has a supplementary fired HRSG and the OCC system proposed by [6] also makes use of a duct burner. Hence, two scenarios are confronted: HRSG with and without supplementary firing. The case with supplementary firing is illustrated in fig. 6.2, where it can be noticed a more realistic fuel feeding system where NG (stocked in a single tank ~20 bar) is addressed to the two combustors at different pressures (pumped to the GT combustion chamber at 37 bar and to DB at 1.7 bar).

6.2.3 Biogas/Landfill gas as an alternative

As suggested by some Brazilian studies [123], [6], biogas could be used instead of natural gas in WTE-GT systems. Biogas or landfill gas is a mixture of gases, mainly composed of methane, produced during the anaerobic decomposition of organic waste. In theory, there would be two possibilities for using biogas/landfill gas in the reference system: (i) replace 100% of NG, feeding both the GT and the duct burner; (ii) replace partially the NG use, feeding only the duct burner (GT continues to be fed with NG). Thermoflex® has in its library a fuel model for landfill gas but it does not allow simulation with such gas feeding the gas turbine, thus the only available option is the second. This is because the majority of commercial GTs are not manufactured to operate with a fuel other than natural gas and, even though the conversion of landfill gas into methane is possible, this procedure is outside scope of this work. Hence, two identical cycles are confronted, one fueled with NG in both GT and DB and the other with NG feeding the GT and landfill gas feeding the DB.

6.2.4 Gas turbine size

The gas turbine capacity (or size) regards the GT power output. It determines the scale of the WTE-GT plant. This means that large turbines generate greater power output because admit larger amounts of air, also generating greater amounts of exhaust gases at higher temperature. As a consequence, it would require a higher steam mass flow coming from the WTE boiler, which by its turn would demand a large-sized MSW feeding system. Within the context of a metropolis, four theoretical GT scales are investigated *small*, *medium*, *large* and *extra large*. Taking as a reference the MSW disposed in the city of Rio de Janeiro (9917 ton/day [124]), the "extra large" scale would stand for a giant plant that could hypothetically treat all the waste generated in the city and neighboring towns in a single WTE-GT unit. The "large scale" plant would stand for a unitary WTE-GT system hypothetically able to receive almost the totality of Rio's waste, whereas the "medium" and "small" scales would be able to process only a fraction of the city's waste. It should be highlighted that, even though it is suggested that each waste feeding line should have a maximum MSW thermal input capacity of 120 MW_t [5], this is not considered here, representing a limitation of the model. This is because the goal of the analysis is to compare different configurations within the same size scale; thus, for simplicity, all cycles are simulated with only one WTE boiler line. Hence, the appraised electricity production and waste treatment costs should not be taken as accurate values, but instead shall be used carefully as a reference. Moreover, this analysis aims at investigating the influence of the GT size in a *theoretical* way. That is, due to the extremely large sizes of the plants in the medium, large and extra large scales, they might not be feasible in reality due to space limitation.

The following GT machines are selected to each scale based on the exhaust gases temperature ($\simeq 600\,°C$ as in the reference system) and on the

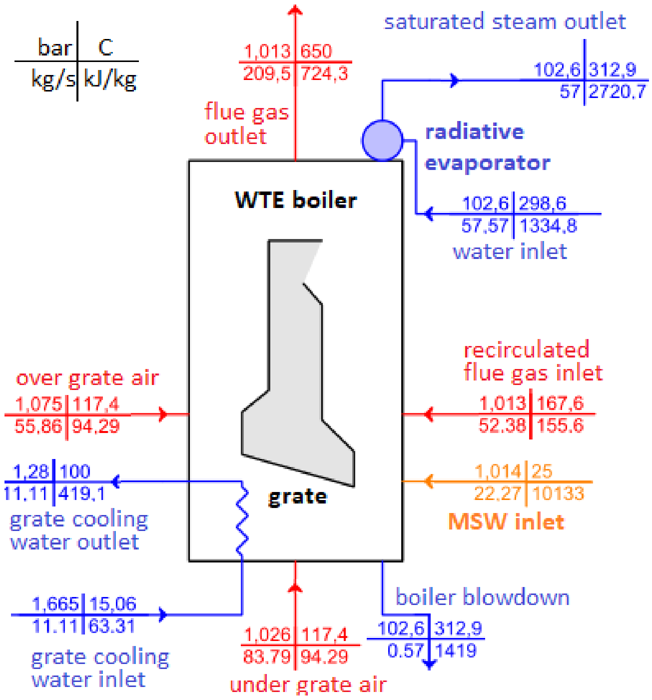


Figure 6.3: Example of WTE boiler component simulated in Thermoflex®

nominal LHV efficiency (η) described by the manufacturers.

- Extra large → GE 9F.06 (η=42%): 388 MW_e.
- Large → MHPS 501 GAC (η=40%): 281 MW_e.
- Medium → MHPS 501F (η=38%): 167 MW_e.
- Small → GE LM6KPGSPT (η=41%): 38 MW_e.

6.2.5 WTE boiler simulation

As observed in the previous chapters, the MSW furnace boiler is one of the lowest performing equipment in WTE plants. Grate-fired furnaces with MSW being fed as received, i.e. without pre-treatment, are better suited for large scale plants. The scheme of the TFX waste boiler component is shown in fig. 6.3, showing an equipment of the type "mass burn reciprocating grate". As observed, in TFX there is a single component that simulates the grate furnace + radiative boiler. It generates flue gases at a pre-defined temperature (radiative zone) and produces saturated steam in the radiative evaporator. The following input variables are required for this component:

- Gage pressure within the furnace (default 0.6227 mbar).
- Excess air (default 60%).

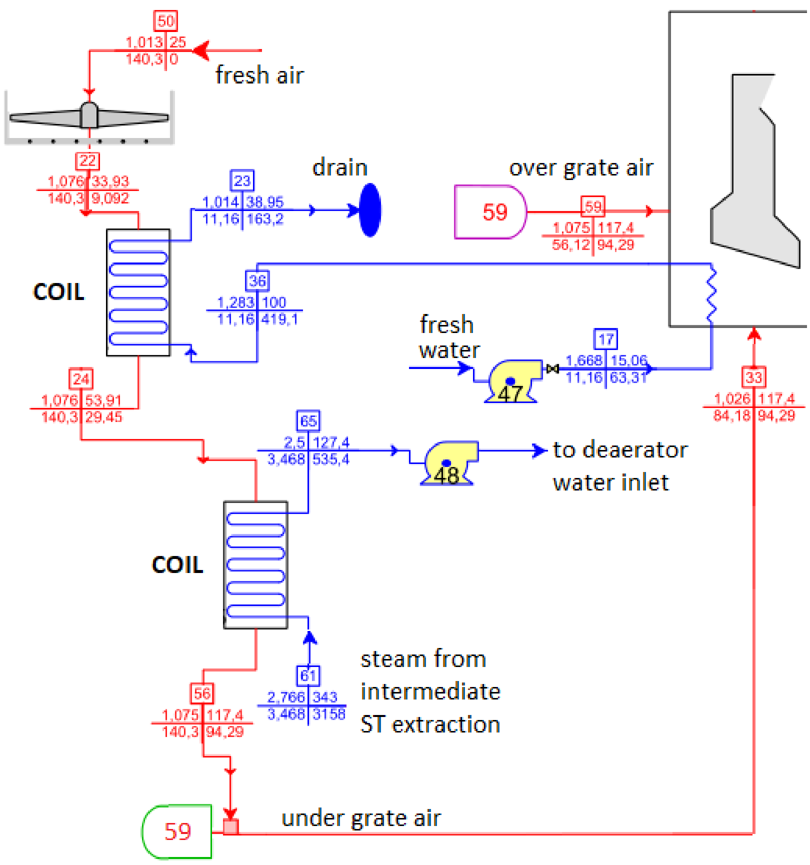


Figure 6.4: Air pre-heating system using water from grate cooling simulated in Thermoflex®

- Boiler type (default natural circulation) governed by (fuel supply or steam supply). Requires indicating the fuel mass flow or the desired steam mass flow produced (kg/s).
- Radiant flux past screen (possible range 0-100%). For instance, 100% means that the radiative evaporator tubes let pass all (100%) the radiation to achieve the superheater or other subsequent heat exchanger in the radiative section. Useful to simulate fouling.
- Furnace gas exit temperature (650 °C, default range 760-871 °C).
- Minor heat loss (2.5%).
- Blowdown (default 1%).
- Under grate air pressure (default 12.45 mbar).
- Ratio between under grate air flow and total air flow (default 60%).
- Overfire air pressure drop (default 62.27 mbar).
- Specific fuel delivery power (default 5.512 kWh/tonne).
- Grate cooling water: water exit temperature (e.g. 100 °C); pressure drop (default 30%) and Q/Q waterwall (default 5%).

It is assumed the grate is cooled with water, as also used in the repowered cycles simulated in chapter 4. Tests have shown that reusing the water from grate cooling to aid in the pre-heating of air is beneficial to the plant's efficiency. In this sense, the adopted air-preheating system, shown in fig. 6.4, uses two coils to heat fresh air before going under and over grate. After cooling the grate, liquid water is addressed to a coil, where the first air pre-heating occurs. Meanwhile, the subsequent coil operates with steam extracted from intermediate-pressure steam turbine, performing the second pre-heat. Both under and over grate air are pre-heated to over 115 °C. In addition, it has also been shown from the simulations that boiler's blowndown reuse also aids increasing the plant's efficiency. As suggested by Thermoflex® instructions, blowdown flows are collected from all evaporators and addressed to a flash tank, where the condensate is discarded and the steam goes back to the cycle entering the feedwater heater stem inlet.

It should be highlighted that the *WTE boiler component* is composed, as observed in fig. 6.3, of the MSW grate combustor + radiative boiler (simulated as a unique component), whereas the MSW furnace or *WTE boiler assembly* is composed of such component + all the heat exchangers in radiative and convective zones (SHs, EVA, ECO). Similarly, the HRSG assembly consists of several heat exchangers placed subsequently. Basically, three layouts of WTE boiler assembly and HRSG assembly are used in the cycles investigated in this chapter: layout A, layout B and layout C, as shown in fig. 6.5. It can be noted that the main differences between layouts A and B is an extra economizer in the WTE boiler assembly, whereas layout C does not use the duct burner in HRSG assembly as the other layouts. In addition, layout A has two and layout B has three feedwater heaters.

It is known that the excess air in grate-fired boilers is relatively high due to heterogeneity of the fuel, which requires more air to achieve complete combustion but also causing the boiler efficiency to be low. The excess air ratio can be reduced, on the primary side, by using a water-cooled grate and on the secondary side by re-circulating a part of the treated flue gas [12]. Moreover, having an excessive high temperature of flue gases at WTE boiler exit represents a loss of energy, that is, a loss of thermal efficiency. Hence, three key aspects that influence the WTE boiler performance are excess air, flue gases temperature and recirculation. Those are discussed in the following subsections.

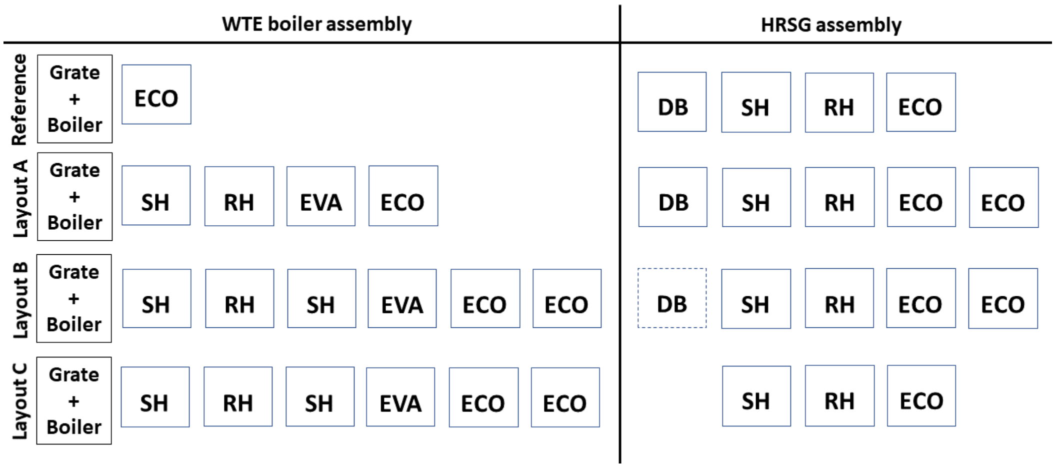


Figure 6.5: Scheme of heat exchangers configuration in both WTE boiler and HRSG assemblies used in each simulated layout.

6.2.5.1
Excess air/ O_2 content

As briefly discussed in section 3.2.2.14, literature reports a broad range that can be assumed for excess air in WTE boilers, 20-220%. The volume (or mols) of oxygen remaining in the exhaust gases of any combustion is due to the presence of excess air. The volumetric oxygen content (O_2%) in dry exhaust gas is defined as the ratio between the volume (or moles) of O_2 and the volume (or mols) of dry exhaust gases (O_2, N_2, CO_2, SO_2, Ar) after complete combustion. It is important to monitor the *O_2 content in the flue gases right after combustion* in order to guarantee a complete combustion and control carbon monoxide formation. From consultation with experts in the field, it is recommended a safe range of 4-8% as the O_2 content in flue gases at WTE boiler outlet. Branchini [12] uses a range of 7-9% and a temperature of 150 °C for the pre-heated primary air, whereas secondary air is not pre-heated or pre-heated to 50 °C. Consonni & Silva [38] assume a value of 5% for the O_2 content in flue gases and define air pre-heating temperature as 115 °C. According to Babcock & Wilcox [62] to aid in the combustion of wet fuels it should be provided undergrate air pre-heated in the range of 149-177 °C. Hence, it is assumed an excess air range of 50-65% and primary/secondary air pre-heated to 115-150°C. O_2 content is monitored to be 4-6% in MSW flue gases in a wet basis, which corresponds to a greater value in a dry basis.

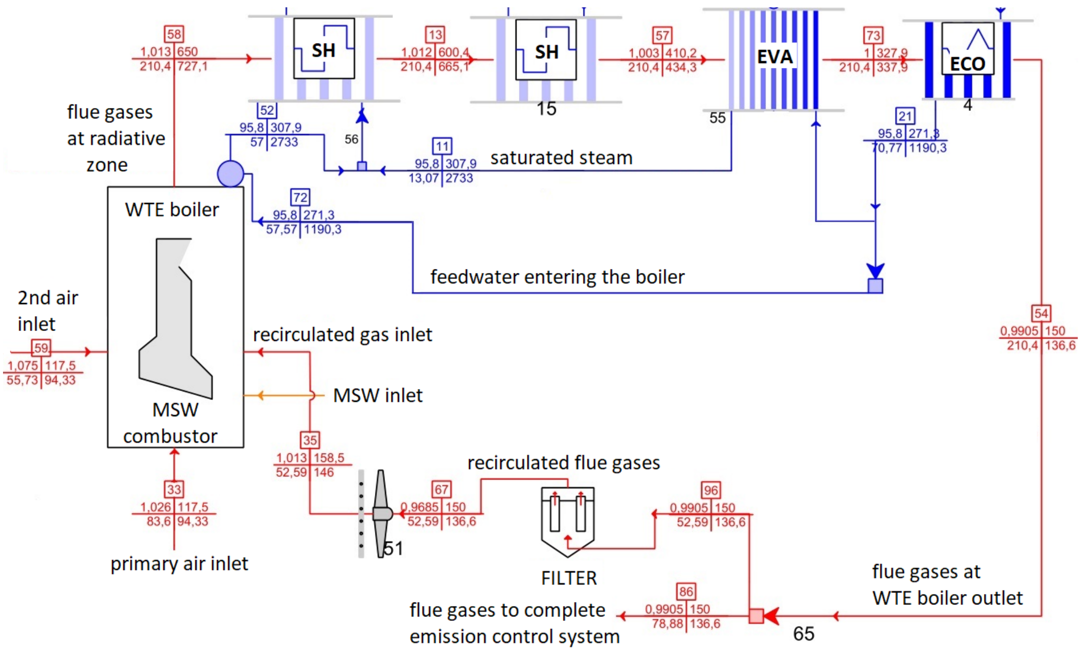


Figure 6.6: Example of WTE boiler assembly with flue gas recirculation simulated in Thermoflex®

6.2.5.2
Recirculation of MSW flue gases

Yin & Li [125] affirm that the flue gas recycling back into the furnace, when properly used, can reduce the excess air and bring other benefits such as "enhancing mixing for homogeneous combustion conditions, better temperature control, suppression of NO_x and dioxins emissions, efficient waste heat recovery, and reduction in slagging tendency and flue gas emissions". Nevertheless, the authors affirm that recycled flue gas is rarely used in grate-fired boilers so far due to various practical difficulties. Leckner [13] describes the possibility of using recirculated flue gas to enhance grate-cooling, as well as mixing it in the freeboard. The author also affirms that in a modern well-designed furnace, such as in the WTE plant of Amsterdam, the excess air ratio is 1.4 and about 25% of the flue gas is recirculated [13]. Lombardi [5] also brings the possibility of blending fresh and recirculated gases, as long as the mixture has an O_2 content not lower than 6%, in order to avoid unburnt material. According to Branchini [12], approximately 10-20% of the usually cleaned flue gases are recirculated, normally after pre-dusting, to replace secondary air feeding into the furnace. This technique is reported to reduce heat losses and primary NO_x emission, as well as to increase the energy efficiency by 0.75-2% [12]. Consonni [14], [38] uses a fraction of 15% of flue gases recirculated. The authors in [37], [12] also assume a recirculated gas rate of 15% at 150 °C. Hence,

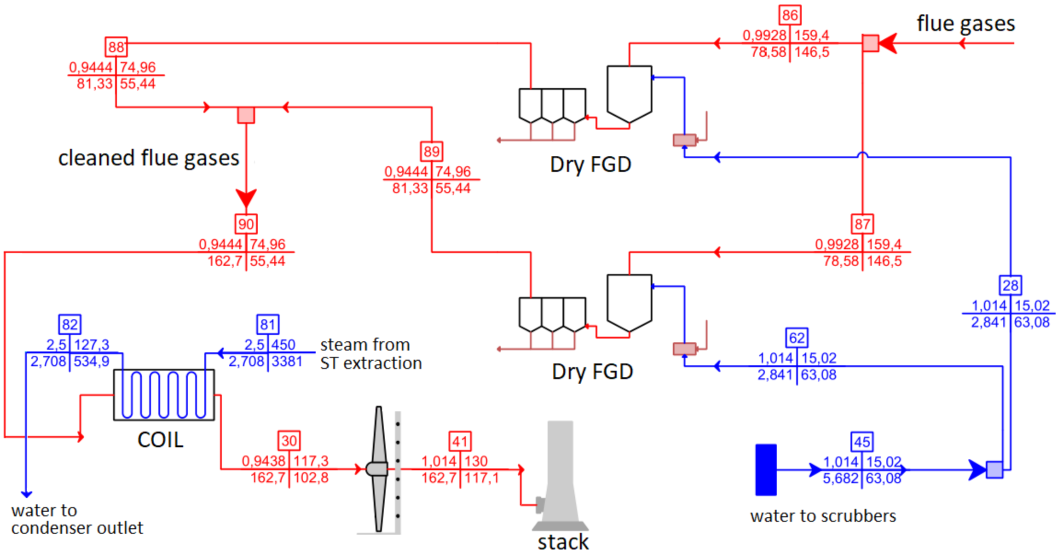


Figure 6.7: Example of flue gas cleaning system simulated in Thermoflex®

it is assumed a range of 10-26% in mass as the fraction of recirculated MSW flue gases. It can be noted in fig. 6.6 that the recirculation includes a pre-dusting system (filter) to partially clean the recycled gases before re-entering the MSW combustor.

6.2.5.3 Flue gases

According to De Greef [126], as the gaseous chloric acids arrive at the superheater section of the MSW furnace corrosion is triggered if the flue gas temperature still exceeds 650 °C and if the superheater surface is at a contact temperature significantly above 400 °C. However, according to Lombardi [5], in the WTE plant of Naples, Italy, the boilers (operating since 2009) are designed for high steam parameters of 500 °C at 90 bar and the superheaters are located in a flue gas temperature zone above 800 °C. Branchini [12] suggests that flue gases outlet temperature should be cooled to a maximum range of 190-160 °C and those from HRSG should be >110 °C. Murer [57] states that the flue gases are cooled down to 180-130 °C. Hence, it is assumed that MSW flue gases leave the radiative zone of WTE boiler at 650 °C and are cooled throughout the convective zone up to 160-150 °C before entering the cleaning system. Observing the reference layout, one can notice that since MSW flue gases are cooled down to 110 °C, at least sulfur condensation probably occurs in the air pre-heating system, which may cause greater costs/time of maintenance. This highlights the importance of respecting those limiting ranges specially for MSW flue gases, in order to avoid corrosion and an excessively high maintenance cost. Another observation is that the reference layout simulates the emission abatement system as Dry FGD (v. section 2.1.10), whose admitted outlet temperature for the cleaned flue gas is 70-75 °C (Thermoflex® default). Because the temperature of MSW flue gases at stack should be preferably superior than 130 °C [57] to avoid condensation of water, it should be included after the FGD a heat exchanger (coil) to increase flue gases temperature from ∼74 °C up to 130 °C at stack. In addition, another practical observation regards the fact that real plants usually have at least two lines for emission abatement placed in parallel, allowing maintenance to be done without shutting down the whole plant. The simulated flue gas cleaning system (MSW furnace outlet) is shown in fig. 6.7.

6.2.6
Exergy, environmental and economic indicators

As mentioned, besides of the 1st Law efficiency, other indicators are calculated to aid in the determination of the best performing system. The ones calculated using the tool 3E EXC described in chapter 2 are: overall exergy efficiency (ϵ), specific carbon emission (mass of CO_2 per MWh of produced electricity) and potential annual profit (A_{sup}, v. eq. 4-6). Other economic indicators are the LCOE (described in 3.2.5.8) and the levelized cost of waste treatment (explained here below). Those are obtained for some designs using the system cost estimated by TFX. Because WTE boiler simulated in all configurations have a pressure that is not within TFX cost functions limit range, the software does not appraise the total equipment cost. Instead it gives a partial estimate of Z_{TPC} missing the WTE boiler cost. Based on tests using cost estimates of the reference layout investigated in ch. 3, it is assumed that $Z_{in} = 3 \times Z_{TFX}$, where Z_{TFX} is the partial equipment cost estimated by TFX.

An additional indicator based on the LCOE formulation is used. Hadidi [127] describes it as "levelised cost of waste" (LCOW), which enables cost comparisons of the treated waste by different conversion technologies in the "basis of a unit cost of waste (ton)", i.e., is given in US$ per ton of waste. It answers the question: "Having fixed the electricity price, how much the Municipality should pay the facility to treat the waste for the investor to break even[2]?" Thus, from the perspective of the society, the LCOW is also an indicator of the type *the lower the better* that is calculated basically the same way as the LCOE (eq. 3-19) as the following:

[2]The incomes equal the charges, i.e. investor has null profit.

Table 6.3: Expenses and incomes considered in the calculation of A_{sup}, $A_{ele,be}$, LCOE and LCOW of hybrid waste-to-energy plants in the Brazilian context.

Expense/income	Value
Electricity sale	141[a]US$/MWh
MSW gate fee	15 US$/ton
NG purchase	10 US$/MBTU
Fly ash disposal	20 US$/ton
Bottom ash disposal	15 US$/ton
O&M variable NG	6 US$/MWh
O&M variable MSW	20 US$/ton
O&M fixed	5% Z_{in}
Operating hours	8000 h/y
Discount rate (r)	10%
Lifespan (n)	20 y
Straight line depreciation	4%
Time to amortize Z_{in}	20 y

[a] Average price for residential Brazilian consumers (0.564 R$/MWh in 2018 [128]). Conversion US$ 1=R$4.00.

$$LCOW = \sum_{t=1}^{n} \frac{\left(I_t + \text{O\&M}_t + P_t + D_t\right)}{(1+r)^t} / \sum_{t=1}^{n} \frac{T_t}{(1+r)^t} \tag{6-1}$$

where LCOW [$/ton] is the levelized cost to treat the waste; T_t [ton/y] is the amount of waste treated in year t; P_t [$/y] is the net profit, i.e., the difference between earnings from electricity sales (positive because is an income) and the fuel cost (C_{NG}, negative) in year t; and the other variables are the same as in eq. 3-19.

Calculation of LCOE and LCOW demands knowing the total initial investment (cost to build & install the plant ready to start-up, i.e. Z_{in}), which is not always available. In this sense, a more simplified economic analysis has been proposed in ch. 4 as an alternative to be used when Z_{in} in unknown and one intends to compare cycles with very similar designs (Z_{in} practically the same for all options). Recapitulating, the strategy assumes the plant is already installed and operating, and consists of a simple cost balance (diminishing the incomes of the charges) giving the annual profit (A_{sup}, v. eq. 4-6). A_{sup} is another indicator of the type *the higher the better*, being useful in the decision between projects with similar fixed costs of amortization, personnel and $Z_{O\&M fixed}$. Another possible use of eq. 4-6 is to find which value of A_{ele} (potential electricity sales price) gives null annual profit (A_{sup}=0). It answers the question: “For a fixed gate fee, for how much should the facility sell the produced electricity for the investor to break even?”. This idealized value of

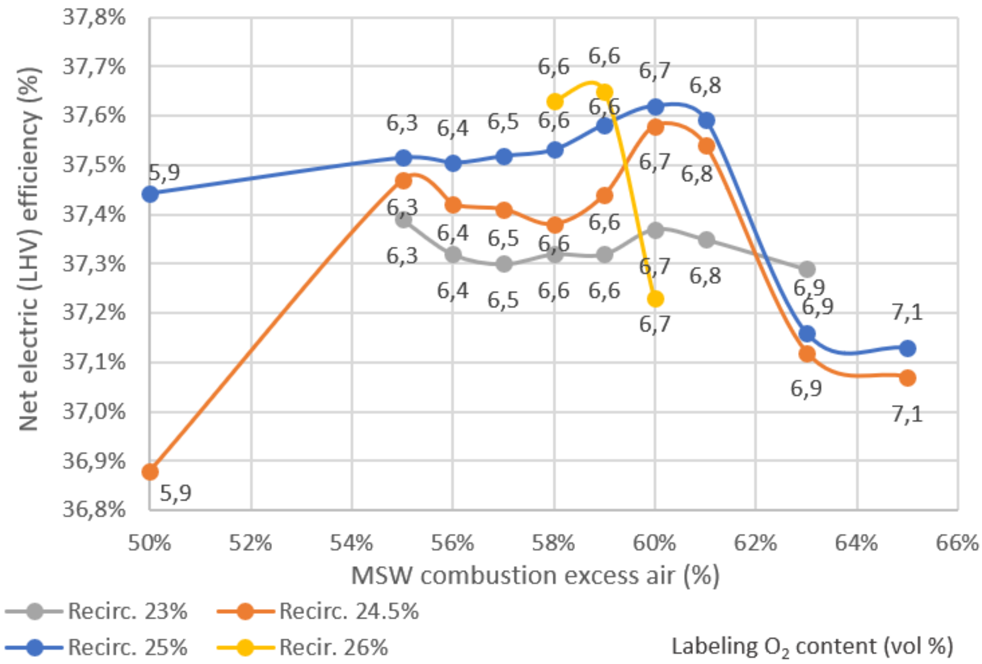


Figure 6.8: Influence of recirculation and excess air in MSW combustion on the energy efficiency.

A_{ele} is called $A_{ele,be}$, which is of the type *the lower the better*. Table 6.3 presents the assumed variables used to calculate the indicators LCOE, LCOW, A_{sup} and $A_{ele,be}$ in the Brazilian context.

6.3 Results and discussion

The results of the energy analysis are first presented aiming to show the influence of five operational and design aspects on the 1st Law efficiency of hybrid WTE cycles, namely the effect of: recirculation of MSW flue gases, deaerator steam pressure, use of a duct burner at HRSG inlet, use of biogas to replace NG in the duct burner, and scale effect determined by the gas turbine size. Then, the energy, exergy, economic and environmental performances of some selected WTE-GT cycles are compared in order to reach a conclusion about the optimal option.

6.3.1 Influence of flue gases recirculation

The influence of excess air in waste combustion and the flue gases recirculation back into the furnace on the energy efficiency of the plant is shown in the fig. 6.8. As it can be observed, there is an optimum value of excess air that allows to achieve a maximum η_{net} for a determined percentage of flue gases recirculation, namely excess air ∼59% and recirculation ∼26% (in mass). Those values are adopted in the simulations here on. It can also be observed that the O_2 content in the flue gases is never lower than 5.9% in wet basis, which means is always greater than ∼6% in dry basis, i.e., within the range of 4-8% in dry basis, hence, assuring complete combustion.

6.3.2 Influence of deaerator pressure

The influence of deaerator pressure on the energy performance is investigated for layout A in two scenarios "A-low" and "A-high", depending on the value of P_{dea}. Scenario A-low has the steam flow feeding the deaerator extracted at ∼2.5 bar (LP-ST), while A-high has it extracted at ∼5 bar (HP-ST). Table 6.4 shows the main energy parameters for the two scenarios, where it can be observed that they present quite similar performances. It can be noticed that increasing the pressure of the deaerator, we increase the temperature of water that enters the HRSG economiser. Since the gas side is not changed and water has a higher temperature, there is less possibility for heat transfer between hot and cold side in the heat exchanger, hence the gas side outlet temperature is increased, which explains the overall efficiency decrease. Hence, the best performing case is "A-low", with the highest efficiencies and highest MSW thermal input percentage (δ). Thus, it can be concluded that changing the extraction point to feed the deaerator maintaining the rest of the cycle fixed is not ideal. That is, to operate with a 5 bar deaerator the configuration of the cycle should be reviewed to obtain a better use of the fuel energy content (increase energy efficiency).

Table 6.4: Energy results showing the influence of deaerator pressure.

Layout	P_{dea}	η_{tot}	η_{net}	$\dot{W}_{tot}$	$\dot{Q}_{NG}$	$\dot{Q}_{MSW}$	δ
A-low	2.6 bar	41.20%	37.16%	145 MW_e	109 MW_t	242 MW_t	68.88%
A-high	4.8 bar	41.04%	36.93%	143 MW_e	109 MW_t	239 MW_t	68.66%

6.3.3 Influence of supplementary firing

Starting from layout "A-high", preserving P_{dea} ∼5 bar, but redesigning the configuration of the system in an attempt to enhance the performance, layout B is obtained. As mentioned, it differs from layout A because it has two extra heat exchangers in the WTE boiler assembly and one more feedwater heater. Layout B is investigated in two scenarios: with and without the presence of a duct burner at the heat recovery boiler inlet, that is, with and without supplementary fired HRSG. The results are shown in tab. 6.5.

Table 6.5: Energy results showing the influence of supplementary firing HRSG.

Layout	Duct burner	η_{tot}	η_{net}	$\dot{W}_{tot}$	$\dot{Q}_{NG}$	$\dot{Q}_{MSW}$	δ
B-wSF	Present	41.73%	37.45%	140 MW_e	109 MW_t	227 MW_t	67.55%
B-noSF	Absent	40.93%	36.68%	119 MW_e	91 MW_t	200 MW_t	68.60%

It can be observed that the energy performance of layout B is better with the supplementary firing to the expense of a smaller δ. That is, with the same GT, the presence of the supplementary firing is able to increase more the superheating temperature of the steam in the HRSG, allowing to produce more power in the ST. This explains the greater power output observed in the case with the duct burner (B-wSF). What can be concluded is that, when removing the DB (case B-noSF), it turns out that the GT exhaust temperature (453 °C) is not sufficient to superheat the steam coming from the WTE boiler much more in the HRSG. Therefore, what happens is a reduction in the efficiency due to an increase of MSW share. Hence, the use of a supplementary fired HRSG is a strategy to increase the plant's efficiency while maintaining the same gas turbine. In particular, the GT machine used in these simulations is the same as in the Bilbao plant (GE LM6000 PD, 43 MW_e), where supplementary firing is used indeed.

6.3.4
Influence of landfill gas use

The results of landfill gas/biogas used to fuel the DB are shown in fig. 6.9. It can be observed that most results are identical, but some differences indicate that operating with landfill gas (as received, i.e., untreated) instead of natural gas to feed the duct burner of supplementary fired HRSG is not a good alternative. The most relevant difference regards the increase in SO_x emission when using landfill gas, due to the fact that it contains a higher sulfur content than natural gas. The energy efficiencies are slightly worse with biogas, which is explained by its smaller LHV (NG LHV ~50 MJ/kg and biogas LHV ~21 MJ/kg). In this sense, there is no energy or environmental advantage in replacing NG by landfill gas in the duct burner of the investigated WTE-GT systems.

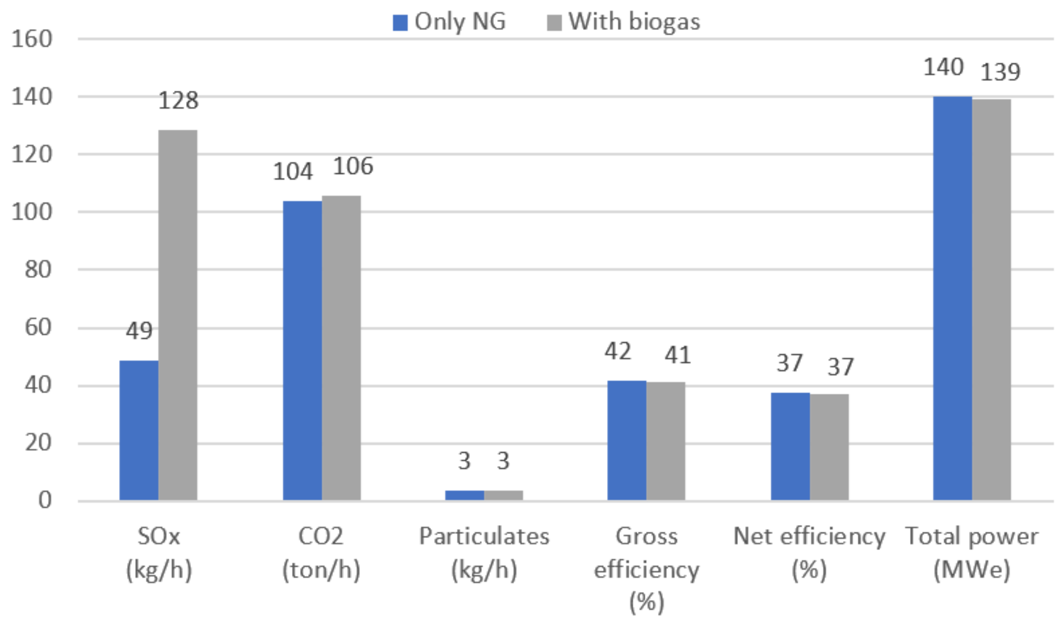


Figure 6.9: Influence of biogas use in the duct burner of HRSG.

6.3.5 Influence of GT size

Both scenarios of layouts A can be confronted with layout B-wSF, because all three cases have in common the presence of supplementary fired HRSG. It can be observed from tabs. 6.4 and 6.5 that case B-wSF presents the highest η but the lowest $\dot{W}$. This makes us realize that using layout B-noSF with a different size of the gas turbine may allow us to enhance the energy performance of the cycle, i.e., increase both η and δ. Thus, the influence of the GT size is investigated within four scenarios of the layout B without supplementary firing, as shown in tab. 6.6.

Table 6.6: Energy results showing the influence of the GT size in layout B without supplementary firing (B-noSF).

GT machine	Scale	MSW capacity	η_{tot}	η_{net}	$\dot{W}_{tot}$	$\dot{Q}_{NG}$	$\dot{Q}_{MSW}$	δ
GE 9F.06	Extra large	11802 t/d	44.54%	40.52%	986 MW_e	829 MW_t	1384 MW_t	62.50%
MHPS-501-GAC	Large	9461 t/d	44.54%	40.54%	785 MW_e	653 MW_t	724 MW_t	62.97%
MHPS-501F	Medium	6048 t/d	43.71%	39.85%	520 MW_e	472 MW_t	718 MW_t	60.28%
GE LM6KPGSPT	Small	1758 t/d	41.81%	37.64%	128 MW_e	99 MW_t	206 MW_t	67.55%

It can be observed that efficiencies increase with scale, that is, the larger the GT the greater η. The extra large and large scales show very similar performances, allowing to achieve the highest η_{net} ~40.5% with a MSW share of about 63%. It is also interesting to notice that the extra-large and large cases have equal values of η_{tot}, but the large scale (machine MHPS-501-GAC) is a better option because achieves a slightly higher η_{net} with a greater δ than the extra-large case. The medium scale has the worst performance because, contrarily to what is expected, presents lower values of η and δ than the small scale. The small scale, by is turn, has the smallest efficiency among all cases justified by a higher MSW share. This shows once again, as concluded in ch. 4, that a poor choice of the GT machine compromises the overall performance of the cycle.

In the next sections are presented the 4E indicators applied only to "small scale" GT layouts. It should be highlighted that those represent large-sized plants (~600,000 ton/y). The reason for not using the other GT scales in the further 4E assessment is because economic results are more reliable in this scale, allowing comparisons with the reference layout and showing the performance evolution up to the optimal arrangement.

6.3.5.1
Feasibility of cases with & without supplementary firing

It is interesting to compare the two small scale cases of layout B: (i) machine GE LM6000 PD with supplementary firing HRSG (case B-wSF in tab. 6.5); and (ii) machine GE LM6KPGSPT without supplementary firing HRSG (small scale in tab. 6.6) because both have the same δ (67.55%).

Table 6.7: Comparison between cases with and without supplementary firing (SF) using energy, exergy, economic and environmental indicators.

Parameters	With SF (i)	Without SF (ii)	Best option
$\dot{Q}_{tot}$	336 MW_t	305 MW_t	ii
$\dot{Q}_{NG}$	109 MW_t	99 MW_t	ii
$\dot{Q}_{MSW}$	227 MW_t	206 MW_t	i
δ	67.5%	67.5%	-
Annual MSW input[a]	644 kton/y	587 kton/y	i
$\dot{W}_{tot}$	140 MW_e	128 MW_e	i
η_{tot}	41.7 %	41.8 %	ii
η_{net}	37.5 %	37.6 %	ii
Overall exergy efficiency (ϵ_{tot})	20.5 %	20.7 %	ii
CO_2 emission	907 kg CO_2/MWh	899 kg CO_2/MWh	ii
LCOE[a,b]	113.4 US$/MWh	112.6 US$/MWh	ii
LCOW[a,b]	335.2 US$/ton	336.2 US$/ton	i
A_{sup}[a,b]	100 Mi US$/y	93 Mi US$/y	i

[a] Availability of 8000 h/y and parameters of tab. 6.3.
[b] American Dollars of 2018.

So far it could be shown that the first (1933 t/d) presents lower η than the second (1760 t/d) and higher power output ($\dot{W}_{tot,i} > \dot{W}_{tot,ii}$), but also higher NG consumption ($\dot{Q}_{NG,i} > \dot{Q}_{NG,ii}$). Hence, choosing the best option requires a deeper investigation of each case including exergy, economic and environmental indicators, as shown in tab. 6.7. The results point out that case ii is the best between the two options because it wins 7 of the 12 evaluated indicators. Even though the difference is very small, it can be concluded that designing a WTE-GT plant with supplementary fired HRSG may not be the best option if there is a good alternative for a GT machine that could be used in the desired scale. That is, designing a HRSG with supplementary firing is only valid if one wants to increase potential profit (or decrease energy production cost) maintaining a specific GT machine.

6.3.5.2
4E indicators of layouts A and B

This subsection presents the comparison between layouts A and B. Three cases, all with supplementary firing, are investigated: (iii) layout A with P_{DEA} ~3 and GT machine GE LM6000PD; (iv) layout A with P_{DEA} ~5 and GT machine GE LM6000PD; and (v) layout B with P_{DEA} ~5 and GT machine MHPS H-25.

Table 6.8: Comparison between layouts A and B using energy, exergy, economic and environmental indicators.

Parameters	**Case iii**	**Case iv**	**Case v**	**Best option**
$\dot{Q}_{tot}$	351 MW_t	349 MW_t	336 MW_t	v
$\dot{Q}_{NG}$	109 MW_t	109 MW_t	110 MW_t	iv
$\dot{Q}_{MSW}$	242 MW_t	240 MW_t	226 MW_t	iii
δ	68.9%	68.7%	67.2%	iii
Annual MSW input[a]	688 kton/y	681 kton/y	641 kton/y	iii
$\dot{W}_{tot}$	145 MW_e	143 MW_e	139 MW_e	iii
η_{tot}	41.2 %	41.1 %	41.3 %	v
η_{net}	37.2 %	36.9 %	37.1 %	iii
ϵ_{tot}	20.2 %	20.1 %	20.4 %	v
CO_2 emission	921 kg CO_2/MWh	926 kg CO_2/MWh	915 kg CO_2/MWh	v
A_{sup}[b]	105.4 Mi US$/y	105.4 Mi US$/y	99.1 Mi US$/y	iii/iv
$A_{ele,be}$[b]	40.1 US$/MWh	38.7 US$/MWh	41.5 US$/MWh	iv

[a] Availability of 8000 h/y and parameters of tab. 6.3.
[b] American Dollars of 2018.

Recapitulating, both layouts A and B have the same HRSG assembly, the difference being in the WTE boiler (v. fig. 6.5) and in the number of feedwater heaters (layout A has two and layout B has three). The results are shown in

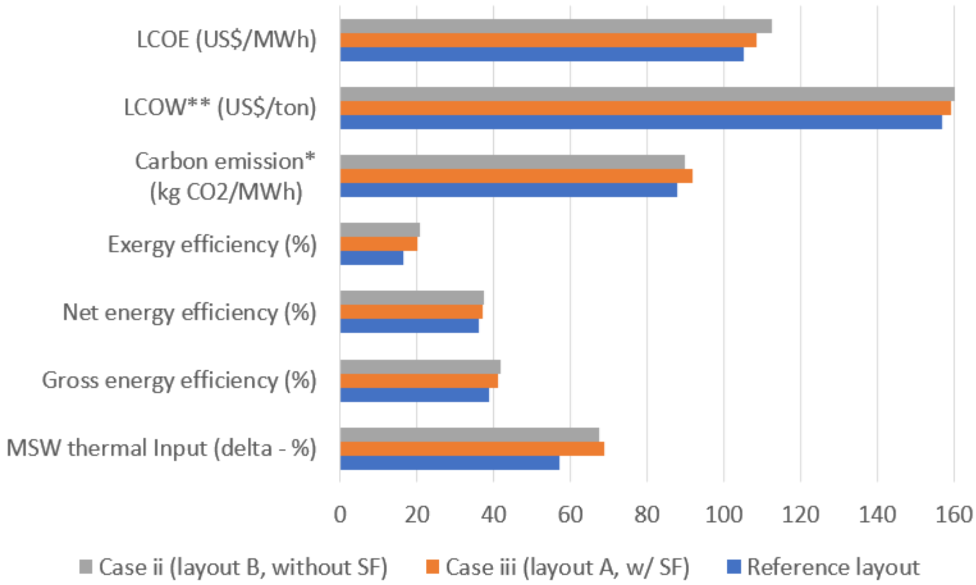


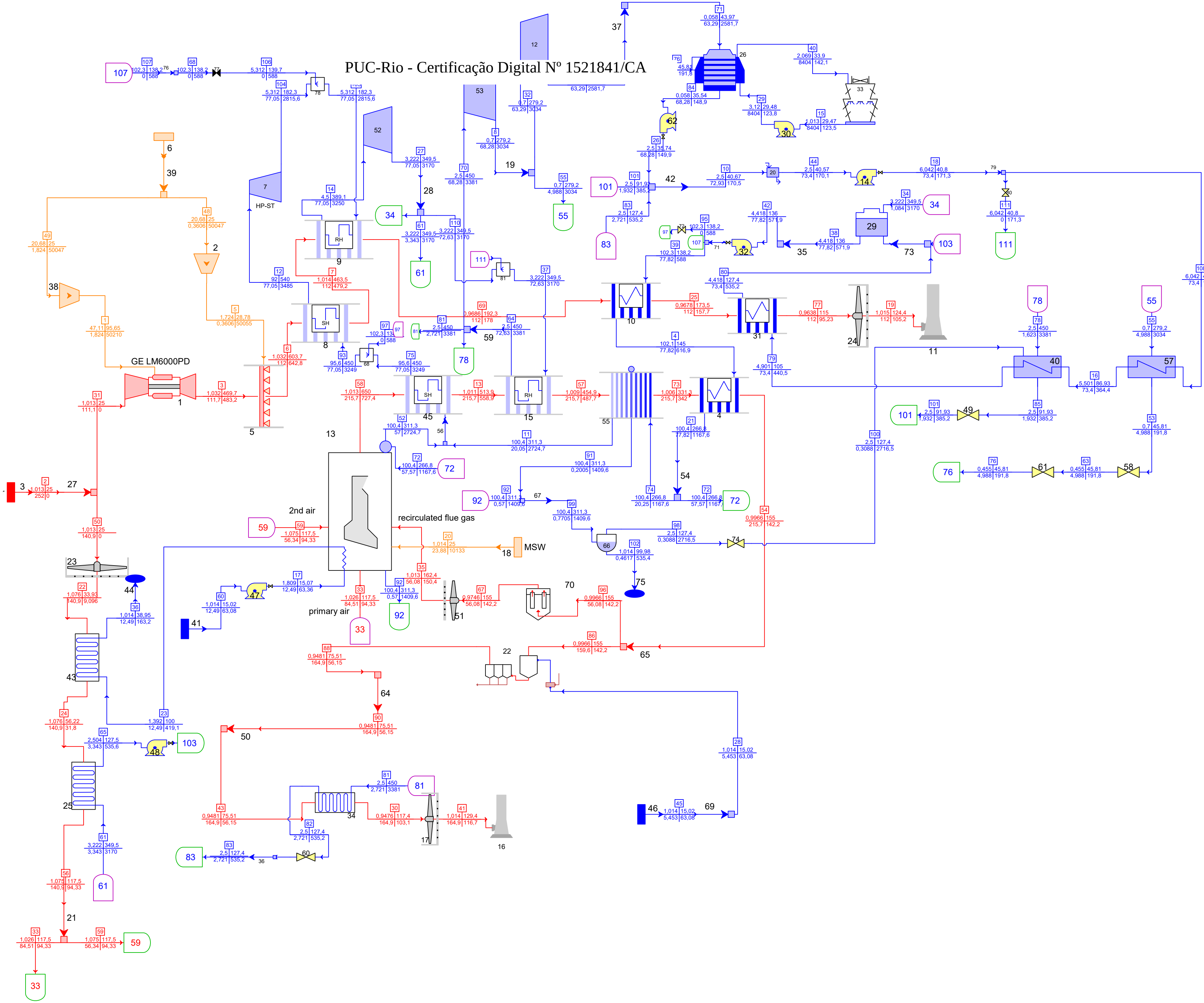
Figure 6.10: Main 4E indicators applied to cases ii, iii and the reference layout. For visualization, *results are divided by 10 and **results are divided by 2.

tab. 6.8, where it can be observed that case iii is the best among the three, wining 6 of the 12 indicators. This is useful to select the case which is worth calculating the LCOE and LCOW in order to compare with case ii.

6.3.5.3
Optimal solution

LCOE and LCOW of case iii are calculated as about US$ 109/MWh and US$ 318/ton. The main 4E indicators of the best options so far (cases ii and iii) and the reference layout are shown in fig 6.10, where LCOW values are divided by 2 and carbon emissions are divided by 10, for visualization purposes. It can be observed the overall evolution of the two improved cycles with respect to the reference layout, where δ has significantly increased as well as η_{tot}, η_{net} and ϵ. The fact that δ has increased explains the higher CO_2 emission of both enhanced cases compared to the reference layout, where the highest carbon emission is observed for the case with the greatest MSW share (highest δ), as expected. The choice of the best option becomes clear when observing the economic indicators, demonstrating that case iii even with the greatest δ has lower LCOE and LCOW than case ii and higher energy and exergy efficiencies than the reference layout, thus being the most favorable case. This is a major finding of this chapter, representing an important conclusion expected in optimization of energy systems: it is worth enhancing an arrangement of a WTE-GT cycle only up to a certain optimal energy/exergy performance, but it is not worth going beyond because it causes excessively high electricity production and waste treatment costs.

The fact that the optimal case has supplementary fired HRSG makes us realize that further improvement could be achieved by removing the duct burner and replacing the GT machine, however, since the gains are quite inexpressive, as observed from results in section 6.3.5.1, this is left as future work. TFX simulation of case iii is shown in the **next page**, where only one line of FGS is depicted even though two lines were considered in the economic analysis.


GE LM6000PD
HP-ST
2nd air
recirculated flue gas
primary air
MSW

6.4 Conclusive remarks and future work

This chapter proposes a realistic optimal design of a hybrid waste-to-energy cycle able to generate electricity with the highest energy/exergy efficiency and the greatest possible amount of energy derived from the waste with the lowest possible cost compared to the reference layout of chapter 3. The system has a gross electric efficiency of 41%, a net electric efficiency of 37% with 69% of MSW share. The overall exergy efficiency is appraised in 20% compared to only 16% of the reference cycle. It represents an increase of 2.5 p.p. in the gross energy efficiency, 1 p.p. in the net energy efficiency, 3.8 p.p. in the exergy efficiency, and 12 p.p. in the MSW thermal input share, hence, achieving the second goal proposed for this chapter. The proposed cycle is a large sized plant of 145 MW_e nominal power output able to treat 688 kton/y of urban waste, which corresponds to about 23% of the waste disposed in landfills of Rio de Janeiro in 2013. The success of the thermodynamic layout is due to the use of an "average-sized" GT integrated to a WTE boiler through a supplementary fired HRSG that allows re-heating the steam twice, first in the HRSG and secondly in the WTE boiler. Even though such steam is reheated at a relatively lower pressure than commonly applied, which may require significantly large tubes and a steam turbine specially designed, those costs are included in the estimates done with Thermoflex® for the modeled cycles. Finally, the optimal solution is compared to the reference layout, which has been shown in ch. 3 to be feasible compared to other WTE and WTE-GT plants in the world. Besides that, important reuse techniques were applied contributing also to the improved energy performance and greater costs of the advanced cycles compared to the reference layout, namely using waste heat from grate cooling water, evaporators blowdown and recycling of MSW flue gases back into the furnace at an optimum percent. Other procedures regarding the practical feasibility are the compliance with the suggested restrictions to diminish corrosion problems: respecting the 450 °C limit for the superheated steam in the WTE boiler, >125 °C for MSW flue gases and >115 °C for HRSG flue gases temperatures at stack. The emissions are controlled by semi-dry flue gas cleaning systems that use less water and are applied in the newest WTE plants worldwide successfully accomplishing international regulations. By the way, this work can be continued by investigating more deeply the emissions, specially NO_x. Further work includes also appraising the effect of using cooling water from the sea in the condenser, supposing the plant is located at sea level, as applied in Amsterdam, which increases efficiency due to lower temperatures expected for sea water.

7
Conclusion

This work presents a comprehensive evaluation of the performance of theoretical hybrid thermopower plants fueled by municipal solid waste and natural gas fired in different combustors. Their structure integrate a topping and a bottoming cycles through a heat recovery boiler aiming to further superheat the steam generated in the MSW furnace boiler with the waste heat from the exhaust gases of a gas turbine. Recent research has shown that such cycles are not as commonly investigated as single-fueled waste-to-energy plants, which are abundantly present in European countries. The fact that MSW can be used to generate partially renewable energy makes it an interesting option to solve both problems of waste treatment and electricity production in big cities. The problem is that, due to urban waste's dirtiness and low calorific value, its combustion requires precise-control techniques and strict restrictions to the cycle's operating conditions to avoid corrosion problems in the WTE boiler, which makes the thermal efficiency of those plants to be small and the electricity production/waste treatment costs to be high compared to other energy conversion and waste disposal technologies. However, another important point is that very few waste disposal technologies are able to efficiently treat extremely large amounts of raw waste in small spaces in a short time. Landfills are being heavily criticized as waste disposal techniques because they are the least efficient, their operation requires a lot of space and their environmental impacts remain long after they close. Only recently the costs of environmental liabilities from closed landfills are being observed in developing countries and their dimensions are yet to be known. Anyway, it is a closed issue that landfills are doomed to extinction, thus what is the solution for our increasing waste generation problem? Waste-to-energy plants are a solution. This is a fact confirmed by their large application in developed countries. However, some questions driving this work are: Should developing nations follow the exact same path as the Europeans? It is known that natural gas is a non-expensive, clean and efficient fuel that can be easily used for electricity production, so why not use such resource, not quite easily accessible in most European region, but available in countries like Brazil? To answer those questions, this works begins with chapter 2 presenting the development of an Excel® code used to automatically calculate exergy balances of complex WTE-GT systems in chapters 4 and 6. Then, chapters 3, 4, and 5 make direct comparisons between single-fueled and dual-fueled systems demonstrating the superior performance of hybrids under the energy, exergy, economic and environmental (4E) perspectives. Finally, chapter 6 proposes a novel advanced WTE-GT layout that allows electricity production at competitive costs, safe environmental emissions with maximum renewable potential. Therefore, two main conclusions of this research are that hybrid waste-to-energy cycles, operating with both urban waste and natural gas, do perform better than single-fueled waste-fired plants and are a feasible alternative to the above-mentioned shortcomings of conventional waste-to-energy plants.

Chapter 3 presents a new strategy to evaluate the feasibility of hybrid waste/gas-fired plants, where a theoretical layout was used to demonstrate the method, which has gross and net energy efficiencies of 39% and 36%, respectively, with 57% of fuel thermal input coming from the waste. The system's ecological efficiency was calculated as 89% which, by its turn, was used to estimate the costs of the pollution abatement devices characterizing one of the main novelties of the proposed method. As unique findings, it was shown that its specific investment costs of the hybrid plants are very attractive compared to existing single-fueled waste-to-energy facilities in Europe and other electricity sources in the Brazilian context. Chapter 5 shows how to appraise the environmental performance of waste-fired plants in an innovative way, also showing the much lower pollution potential of hybrid systems compared to single-fueled ones. It presents a comprehensive technical review of the different theories of the energy-ecologic efficiency and proposes a novel arrangement to solve the main identified problems regarding the existing strategies. As unique findings, MSW fired plants had their energy-ecological efficiency determined through different methodologies and compared to NG and MSW/NG fired systems, also investigating the influence of carbon offset due to biomass regrowth. In conclusion, for the investigated cases, the only way a single-fueled WTE plant can overcome a WTE-GT in terms of EE is if the first has an energy efficiency of about 38% (CHP system) and the second operates with a thermal efficiency 27% smaller than the first.

The most developed European nations decision of not adopting hybrid WTE plants in the past is turning out to be a mistake. In order to avoid increasing even more their dependency on the imported natural gas, already paramount for residential heating, European countries always preferred using conventional WTE plants instead of hybrids, and what happened? The scarcity of waste due to successful prevention policies and increasing of recycling rates force many facilities to import waste or operate under capacity, causing economic losses to the owners, which are now considering repowering the plants to turn them into hybrids. Chapter 4 shows that repowered WTE-GT plants are in all aspects more advantageous than operating underutilized single-fueled waste-fired systems. It investigates the energy, exergy, economic and environmental performance of four repowering alternatives and compare them to the original single-fueled waste-to-energy plant (both under-utilized and fully-utilized). The results show that all repowering options present higher profit potential than the single-fueled system, with a potential extra gain of millions of Euros per year. But what if instead of repowering an existing facility a brand new advanced hybrid plant could be built? How much better would it perform compared to the best repowered WTE-GT layout and the average performing dual fuel cycle of chapter 3 (reference cycle - RC)? To answer this question, chapter 6 investigates several configurations of large-sized WTE-GT and proposes a novel layout with maximized efficiency and MSW share at affordable costs. The optimum design has a gross electric efficiency of 41% (compared to 39% of RC), a net electric efficiency of 37% (compared to 36% of RC) with 69% of thermal input share coming from the waste (compared to 57% of RC), and an overall exergy efficiency of 20% (compared to only 16% of RC). Such system is a large-sized power plant of 145 MW_e with the capacity to process 688 kton of MSW per year, which corresponds to about 23% of the waste sent to final disposal in Rio de Janeiro in 2013. For comparison, the best brownfield project to re-power an existing underutilized plant could achieve a net energy efficiency of 33% with 52% of MSW thermal input share This demonstrates that a much higher energy/exergy performance can be achieved if the plant is well designed as a dual fuel facility *from the beginning*. Not to mention the expressively lower levelized cost of electricity of the greenfield projects; LCOE estimated in $\simeq$109 US$/MWh against 119 US$/MWh of the brownfield project in the Brazilian context. In conclusion, it is clear that investing in well designed hybrid waste-to-energy plants is feasible when considering energy, economical and environmental aspects, allowing significantly higher gains than single-fueled plants, being a much more reasonable choice to design and implement dual-fueled facilities than repowering them later. Hopefully this work can provide valuable information for decision makers and engineers worldwide to help fostering this energy conversion technology in a near future in developing countries which are so proned to copy developed nations, even in their mistakes.

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A
4E Analysis methodology supplementary equations

1st strategy for energy assessment in the 4E analysis method: only analytical equations. Chapter 3, section 3.2.2 describes that the 1st energy analysis method consists of applying the following analytical equations as described in [50]:

$$\dot{m}_{ng1} = LHV_{ng} \cdot HR \cdot \frac{\dot{W}_{net,GT}}{3600} \tag{A-1}$$

where HR is the GT heat rate [kJ/kWh].

$$\dot{m}_{air1} = \dot{m}_{GT} - \dot{m}_{ng1} \tag{A-2}$$

$$\dot{Q}_{cc} = LHV_{ng} \cdot \dot{m}_{ng1} \tag{A-3}$$

$$Perc_{air} = \dot{m}_{air1} / \dot{m}_{ng1} \tag{A-4}$$

$$\dot{Q}_{SF} = \dot{m}_{ng2} \cdot LHV_{ng} \tag{A-5}$$

$$\dot{W}_{GT} = \dot{m}_{GT}\,(h_3 - h_4) \tag{A-6}$$

$$\eta_{GT} = (h_3 - h_4)/(h_3 - h_{4s}) \tag{A-7}$$

$$\dot{W}_{cp} = \dot{m}_{air1}\,(h_2 - h_1) \tag{A-8}$$

$$\dot{W}_{net,GT} = \dot{W}_{GT} - \dot{W}_{cp} \tag{A-9}$$

$$\eta_{cc} = \dot{Q}_{23}/\dot{Q}_{cc} \tag{A-10}$$

$$\eta_{GTS} = (\dot{W}_{GT} - \dot{W}_{cp})/\dot{Q}_{cc} \tag{A-11}$$

$$\dot{V}_{ng} = (\dot{m}_{ng1} + \dot{m}_{ng2}) / \rho_{ng} \tag{A-12}$$

where ρ_{ng} is the NG density in [kg/Nm^3].

$$\dot{m}_5 = \dot{m}_{GT} + \dot{m}_{ng2} + \dot{m}_{air2} \tag{A-13}$$

Where air can be applied to the secondary NG combustor ($\dot{m}_{air2}$), if necessary:

$$\dot{m}_{air2} = Perc_{air2} \cdot \dot{m}_{ng2} \tag{A-14}$$

where $Perc_{air2}$ is the ratio of additional air added to the second combustor.

$$\dot{m}_5 = (\dot{m}_{GT}\, h_4 + \dot{m}_{air2}\, h_1 + \dot{m}_{ng2}\, LHV_{ng})/\, h_5 \tag{A-15}$$

$$h_5 = (\dot{Q}_v/\eta_{HRB} + \dot{m}_5\, h_6)/\dot{m}_5 \tag{A-16}$$

where $\dot{Q}_v = \dot{m}_v\, [(h_{11} - h_{10}) + (h_{13} - h_{12}) + (h_9 - h_8)]$

$$\dot{m}_{gin} = \dot{m}_{air,inc} + \dot{m}_{MSW} \tag{A-17}$$

where $\dot{m}_{gin}$ is the mass flow [kg/s] of flue gases + bottom ashes in the furnace outlet and with $\dot{m}_{air,inc}$ being the air mass flow at MSW combustion at the furnace in [kg/s] given by:

$$\dot{m}_{air,inc} = \dot{m}_{MSW} \cdot F_{air} \tag{A-18}$$

where F_{air} (dimensionless) is the ratio between the mass flows of air and MSW at the furnace.

The calculations of the main parameters of the steam power cycle are described by the following equations:

$$\dot{m}_v = \eta_{HRB}\, [\, \dot{m}_5 \cdot (h_5 - h_6)\,]/(h_9 - h_8 + h_{11} - h_{10} + h_{13} - h_{12}) \tag{A-19}$$

$$\dot{Q}_{MSW} = LHV_{MSW} \cdot \dot{m}_{MSW} \tag{A-20}$$

$$\eta_{inc} = \dot{m}_v\, (h_{10} - h_9)/\dot{Q}_{MSW} \tag{A-21}$$

$$\dot{W}_{STA} = \dot{m}_v\, (h_{11} - h_{12}) \tag{A-22}$$

$$\dot{W}_{STB} = \dot{m}_v\, (h_{13} - h_{14}) \tag{A-23}$$

$$\dot{W}_{pp} = \dot{m}_v\, (h_8 - h_7) \tag{A-24}$$

$$\dot{W}_{net,ST} = \dot{W}_{STA} + \dot{W}_{STB}\, (\dot{W}_{pp} + IC) \tag{A-25}$$

where IC is the internal consumption, which according to [90] for a WTE plant is 150 kWh/ton of the entering waste.

$$\eta_{VC} = \dot{W}_{net,ST}/[\, \dot{m}_5\, (h_5 - h_6) + \dot{Q}_{MSW}\,] \tag{A-26}$$

Total net power generated by the combined cycle is calculated as:

$$\dot{W}_{net} = \dot{W}_{net,ST} + \dot{W}_{net,GT} \tag{A-27}$$

Total gross power generated by the combined cycle is calculated as:

$$\dot{W}_{tot} = \dot{W}_{STA} + \dot{W}_{STB} + \dot{W}_{net,GT} \tag{A-28}$$