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DOMESTIC HOT WATER AND SPACE HEATING RESIDENTIAL SYSTEMS: EUROPEAN UNION SCENARIO AND ENERGETIC AND EXERGETIC ANALYSIS

Dissertação no âmbito do Mestrado Integrado em Engenharia Mecânica, ramo de Energia e Ambiente, orientada pelo Professor Doutor José Manuel Baranda Ribeiro e pelo Mestre João Pedro da Silva Pereira, apresentada ao Departamento de Engenharia Mecânica da Universidade de Coimbra.

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DOMESTIC HOT WATER AND SPACE HEATING RESIDENTIAL SYSTEMS: EUROPEAN UNION SCENARIO AND ENERGETIC AND EXERGETIC ANALYSIS

Submitted in Partial Fulfilment of the Requirements for the Degree of Master's in Mechanical Engineering in the speciality of Energy and Environment.

Equipamentos residenciais para produção de água quente para fins domésticos e para aquecimento de espaços: panorama na União Europeia e análises energética e exergética

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Abstract

Across the European Union, in almost every household, an equipment to provide domestic hot water and/or space heating is installed. These equipments consume a large portion of the energy that is used in the EU. Since the energy sector is one of the biggest contributors to the worsening of the global climate conditions, it is necessary to find more efficient and clean solutions.

The objectives of this master's thesis fall into two distinct, yet intrinsically related areas. The first objective was to determine which technologies are currently used in the EU to provide domestic hot water and/or space heating, compare them, and determine which is best. This comparison considered economic factors, performance indicators, inputs and outputs. The second objective is related to the comparison of technologies, since energetic and exergetic analyses were made to the best technology, the boiler, and its most threatening technology, the ORC micro-CHP systems. Within the ORC micro-CHP systems, several variants of this technology were analyzed to determine which is the most suitable for domestic hot water production.

Content wise, this document is also divided into two parts. The first part consists of a state of the art of the technologies currently installed in the EU to ensure hot water and/or space heating residential needs. This state of the art includes the characterization of the current energetic scenario in terms of heating needs in the EU, the identification of the main technologies installed, a characterization of each of these technologies (including a SWOT analysis of each one), a direct comparison between all the technologies and finally an evaluation of the capacity of each technology to directly replace the most installed technology (non-condensing boiler). In the second part of this master thesis, energetic and exergetic analyses were performed for four configurations. The first configuration is the combination of a condensing boiler, considered the best technology in the previous section, and a heat engine. The remaining configurations are variants of the ORC micro-CHP technology, considered in the previous section as the greatest threat to boilers in the near future. The energetic analyses performed evaluate the thermal and electrical efficiencies of the configurations, determine the required thermal input and electrical output, and quantify the primary energy savings associated with the normal operation of each system. The exergetic analyses boil down to the determination of the exergy destroyed during the normal operation of each configuration and their efficiency according to the second law of thermodynamics.

Finally, it can be concluded that traditional boilers are still the most installed equipment in the EU to suppress domestic hot water and/or space heating needs. However, this technology is threatened by its technological evolution, the condensing boiler, and by two innovative technologies, heat pumps and micro-CHP systems. However, it was found that the most suitable equipments to directly replace a traditional boiler are condensing boilers and ORC micro-CHP systems. On the energetic and exergetic level, it was found that the hybrid variant of the ORC micro-CHP systems obtains the best results in most of the parameters analyzed, thus being a better equipment to supply domestic hot water and space heating needs than the other analyzed variants of ORC micro-CHP systems and then the combination of a condensing boiler and a heat engine for combined heat and power production.

Keywords Domestic Hot Water, Space Heating, Boiler, Micro-CHP, ORC, Exergy.

Resumo

Em toda a União Europeia, na maioria das habitações, encontra-se instalado um equipamento para suprir necessidades de água quente para fins domésticos e/ou de aquecimento de espaço. Estes equipamentos consomem uma grande parte da energia que utilizada atualmente na UE. Dado que o sector energético é um dos que mais contribui para o agravamento das condições climáticas globais, é necessário encontrar soluções mais limpas e eficientes.

Os objetivos desta dissertação de mestrado enquadram-se em duas áreas distintas, mas intrinsecamente relacionadas. O primeiro objetivo foi determinar quais as tecnologias atualmente utilizadas na UE para suprir necessidades de água quente para fins domésticos e/ou de aquecimento de espaços, compará-las, e determinar qual a melhor. Esta comparação considerou fatores económicos, indicadores de desempenho, inputs e outputs. O segundo objetivo está relacionado com a comparação de tecnologias, uma vez que foram feitas análises energéticas e exergéticas à melhor tecnologia, a caldeira, e à tecnologia com maior potencial para a substituir, os sistemas micro-CHP ORC. Dentro dos sistemas micro-CHP ORC, foram analisadas várias variantes desta tecnologia para determinar qual é a mais adequada para a produção de água para fins domésticos e para aquecimento de espaços.

Em termos de conteúdo, este documento está também dividido em duas partes. A primeira parte consiste num estado da arte das tecnologias atualmente instaladas na UE para assegurar as necessidades residenciais de água quente e/ou de aquecimento de espaços. Este estado da arte inclui a caracterização do atual cenário energético em termos de necessidades de aquecimento na UE, a identificação das principais tecnologias instaladas, uma caracterização de cada uma destas tecnologias (incluindo uma análise SWOT de cada uma), uma comparação direta entre todas as tecnologias e finalmente uma avaliação da capacidade de cada tecnologia para substituir diretamente a tecnologia mais instalada (caldeira tradicional). Na segunda parte desta dissertação de mestrado foram realizadas análises energéticas e exergéticas para quatro configurações. A primeira configuração é a combinação de uma caldeira de condensação, considerada a melhor tecnologia na secção anterior, e um motor térmico. As restantes configurações são variantes da tecnologia micro-

CHP ORC, considerada na secção anterior como a maior ameaça às caldeiras num futuro próximo. As análises energéticas realizadas avaliam a eficiência térmica e elétrica das configurações, determinam o input térmico e o output elétrico necessários, e quantificam a poupança de energia primária associada ao funcionamento normal de cada sistema. As análises exergéticas resumem-se à determinação da exergia destruída durante o funcionamento normal de cada configuração e a sua eficiência de acordo com a segunda lei da termodinâmica.

Por fim, pode concluir-se que as caldeiras tradicionais ainda são o equipamento mais instalado na UE para suprir as necessidades de água quente para fins domésticos e/ou para aquecimento de espaços. Contudo, esta tecnologia encontra-se ameaçada pela sua evolução tecnológica direta, a caldeira de condensação, e por duas tecnologias inovadoras, as bombas de calor e os sistemas de micro-CHP. No entanto, verificou-se que os equipamentos mais adequados para substituir diretamente uma caldeira tradicional são as caldeiras de condensação e os sistemas de micro-CHP ORC. A nível energético e exergético, verificou-se que a variante híbrida dos sistemas de micro-CHP ORC obtém os melhores resultados na maioria dos parâmetros analisados, sendo assim um equipamento melhor para fornecer água quente para fins domésticos e para aquecimento de espaços do que as outras variantes dos sistemas de micro-CHP ORC analisados e do que a combinação de uma caldeira de condensação e um motor térmico para produção combinada de calor e energia.

Palavras-chave: Água quente para fins domésticos, Aquecimento de espaços, Caldeira, Micro-CHP, ORC, Exergia.

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List of Symbols

$\dot{E}x$ – Exergy transfer rate; [kW]

$\dot{m}$ – Mass flow rate; [kg/s]

$\dot{Q}$ – Thermal power; [kW]

$\dot{W}$ – Mechanical power; [kW]

Ex – Specific exergy; [kJ/kg]

h – Specific enthalpy; [kJ/kg]

p_{cond} – Pressure of the organic fluid in the condenser of the power circuit (ORC); [kPa]

p_{evap} – Pressure of the organic fluid in the evaporator of the power circuit (ORC); [kPa]

q – Specific value of the energy transferred by heat; [kJ/kg]

r_p – Pressure ratio;

s – Specific entropy; [kJ/(kg·K)]

$T_{CG,in}$ – Inlet temperature of the combustion gases; [°C]

$T_{CG,out}$ – Outlet temperature of the combustion gases; [°C]

$T_{evap,out}$ – Temperature of the organic fluid at the evaporator exit; [°C]

$T_{sat,cond}$ – Condensation temperature of the organic fluid in the condenser; [°C]

$T_{sat,evap}$ – Saturation temperature of the organic fluid in the evaporator; [°C]

$T_{w,in}$ – Inlet temperature of the water; [°C]

$T_{w,out}$ – Outlet temperature of the water; [°C]

w – Specific value of the energy transferred by work; [kJ/kg]

ΔT_{cond} – Difference between the outlet temperatures of the heat exchanging fluids in the condenser of the power circuit; [°C]

ΔT_{evap} – Difference between the outlet temperatures of the heat exchanging fluids in the evaporator of the power circuit; [°C]

ΔT_{SH} – Superheating degree of the organic fluid; [°C]

$\eta_{CHP,E}$ – Thermo-electric efficiency of the micro-CHP system;

$\eta_{CHP,H}$ – Thermo-heat efficiency of the micro-CHP system;

η_{cond} – Global efficiency of the power circuit condenser;

$\eta_{EM,ger}$ – Electromechanical efficiency of the generator;

η_{evap} – Global efficiency of the power circuit evaporator;

η_{Exp} – Isentropic efficiency of the power circuit expander;

η_{II} – Second law efficiency;

η_{P} – Isentropic efficiency of the power circuit pump;

$\eta_{REF,E}$ – Reference thermo-electric efficiency;

$\eta_{REF,H}$ – Reference thermo-heat efficiency;

$\eta_{th,E}$ – Thermo-electric efficiency of the power circuit;

$\eta_{th,H}$ – Thermo-heat efficiency of the power circuit.

Acronyms/Abbreviations

CB – Condensing boiler;

CHP – Combined Heat and Power;

DEM – Departamento de Engenharia Mecânica;

DH – District Heating;

DHW – Domestic Hot Water;

EU – European Union;

FC – Fuel Cell;

FCTUC – Faculdade de Ciências e Tecnologia da Universidade de Coimbra;

GHG – Greenhouse Gases;

HE – Heat engine;

HP – Heat Pump;

I.V. – Intermediate vaporization;

ICE – Internal Combustion Engine;

IEA – Internal Energy Agency;

LPG – Liquified Petroleum gas

NZEB – Near Zero Energy Building;

ORC – Organic Rankine Cycle;

PEMFC – Polymer electrolyte membrane fuel cell;

PES – Primary Energy Savings;

PV – Photovoltaic;

RE – Reciprocating Engine;

RES – Renewable Energy Systems;

SE – Stirling Engine;

SH – Space Heating;

SOFC - Solid oxide fuel cell;

ST – Solar Thermal;

SWOT – Strengths, Weaknesses, Opportunities and Threats;

UK – United Kingdom.

Subscripts

0 – Dead state;

CG – Combustion gases resulting from the burning of natural gas;

cond – Condenser;

CV – Control volume;

destr – Destroyed;

evap – Evaporator;

exp – Expander;

HE – Heat exchanger;

in – Inlet;

int – Intermediate;

out – Outlet;

P – Pump;

r – Real value of a property;

s – Isentropic value of a property;

TO – Thermal oil;

w – Water;

WF – Working fluid.

1. INTRODUCTION

Currently, the energy sector is one of the biggest contributors to the worsening of the global climate situation (IEA - International Energy Agency 2021). Domestic equipments that provide hot water for domestic purposes and/or space heating are present in almost all dwellings in the countries of the European Union (Pezzutto et al. 2019a). Most of them use fossil fuels as their primary energy source, mainly oil byproducts and natural gas (Pezzutto et al. 2019b). The combustion of such fuels emits greenhouse gases that seriously aggravate the problem of the climate changes that is already beginning to manifest itself on a large scale all over the world.

Despite all the efforts invested in the development and massification of the renewable energies, their widespread implementation at the domestic level is still hindered. The main obstacle is the high purchase price of renewable solutions to produce hot water for domestic purposes and/or for space heating, when compared with traditional solutions such as electric water heaters and boilers. Therefore, it is necessary to find intermediate solutions that are within the economic reach of most consumers at the domestic level, and that will help to mitigate the problems associated with the use of traditional systems, such as the burning of fossil fuels. Some of these intermediate solutions, as duly justified in section 2 of this master's thesis, could be the micro-CHP systems and the heat pumps (Chen et al. 2021).

The topic covered in this master's thesis is the domestic systems that are used to provide hot water for domestic purposes and/or space heating. More specifically, the analysis of the types of equipments that are currently used in the European Union for these purposes. This topic fits into the general panorama of the climate changes, more specifically contributing to the numerous investigations that are carried out on this subject and that seek concrete solutions to solve this global problem. It was concluded, after some research, that there was no document that totally characterized this type of equipments in the EU. Therefore, it was defined that the second chapter of this document would be dedicated to making a state of the art of residential systems that provide hot water for domestic purposes and/or space heating.

Considering all that has been said in the previous three paragraphs, the work developed throughout this study is divided into two parts. Firstly, a characterization is made of which technologies are currently used at a domestic level in the EU to provide hot water for domestic purposes and/or space heating. This characterization involves the following steps: i) energetic characterization of the heating sector in the EU; ii) a review of the relevant technologies used at household level to provide hot water for domestic purposes and/or space heating, as well as the number of installed units of each type of technology; iii) further elaboration of step ii) for the five European Union countries with the highest thermal needs; iv) characterization of the relevant technologies used at household level to provide hot water for domestic purposes and/or space heating; v) direct comparison between the relevant technologies used at household level; vi) evaluation of the ability of the various technologies addressed to directly replace a non-condensing boiler (the most widely used solution).

In the second part of the content of this master's thesis (section 3) an energetic and exergetic analysis of four hot water heating solutions for domestic and/or space heating purposes is performed. The energetic analysis was performed to evaluate the performance of the studied solutions. The exergetic analysis was performed to determine the amount of useful work potential that is used in water heating systems for domestic and space heating purposes. This is done to evaluate the use and quality of the energy in the various components of these systems. These solutions were named configurations and organized alphabetically from A to D. The first configuration (A) results from the combination of the most installed solution in the EU to produce hot water for domestic purposes and space heating, the boiler technology, with an electricity production device, a heat engine. Configuration B is a standard ORC micro-CHP system. This second configuration was chosen since, accordingly to section 2.4.2, the ORC micro-CHP technology is the best technology suitable to directly retrofit boilers. The remaining configurations, C and D, are variants of the standard ORC micro-CHP technology. Configuration C is an ORC micro-CHP system with intermediate vaporization of the organic fluid and configuration D is a hybrid ORC micro-CHP system. These configurations are studied to compare with the rest of the configurations their energetic and exergetic performances when operating under the same conditions. The energetic performance of the four configurations is analyzed by calculating their thermal and electrical global efficiencies, thermal and electrical powers and primary energy savings. As for the exergetic performance of the analyzed configurations, it is measured trough parameters such as the second law efficiencies and the exergy destroyed in each configuration.

2. DOMESTIC HOT WATER AND RESIDENTIAL SPACE HEATING TECHNOLOGIES USED IN THE EUROPEAN UNION: A STATE-OF-THE-ART REVIEW

Domestic hot water (DHW) is a key asset regarding the comfort in the lifestyle of European citizens. In residential buildings, DHW can be use direct or indirectly. The direct uses are all those who imply the use of hot water in a tap. These include sanitary purposes, like showering and other hygiene purposes, and domestic activities, such as cooking and cleaning. Within the aim of this thesis, all these direct uses will be encompassed in the designation "hot water for domestic purposes" or simply DHW.

On the other hand, the indirect uses are all those which do not require a constant and direct consumption of water. This type of use applies to space heating (SH), which is normally accomplished using a central heating system. Hydronic central heating systems use water as the heat transfer medium (Martinopoulos, Papakostas, and Papadopoulos 2018). This water, when heated to a specific temperature, is used to warm up the interior spaces of a residential building during the heating season. Usually, central heating systems also produce DHW for direct use by the consumers (Pomianowski *et al.* 2020).

2.1. Energy and the heating sector in the European Union

The energy used for SH and DHW purposes in residential buildings in the European Union (EU), during 2019, accounted for more than 20% of the total energy used in its 28 countries. This percentage of the total energy used translates to 3880 TWh/year, of which 85% was used for space heating purposes (3298 TWh/year) and the remaining 15% for the production of hot water for domestic purposes (582 TWh/year) (Pezzutto *et al.* 2019a).

According to the International Energy Agency (IEA - International Energy Agency 2021), the energy consumption in the EU for the entire heating sector, considering the primary energy used, is represented graphically from 1990 to 2019 in Figure 2.1.

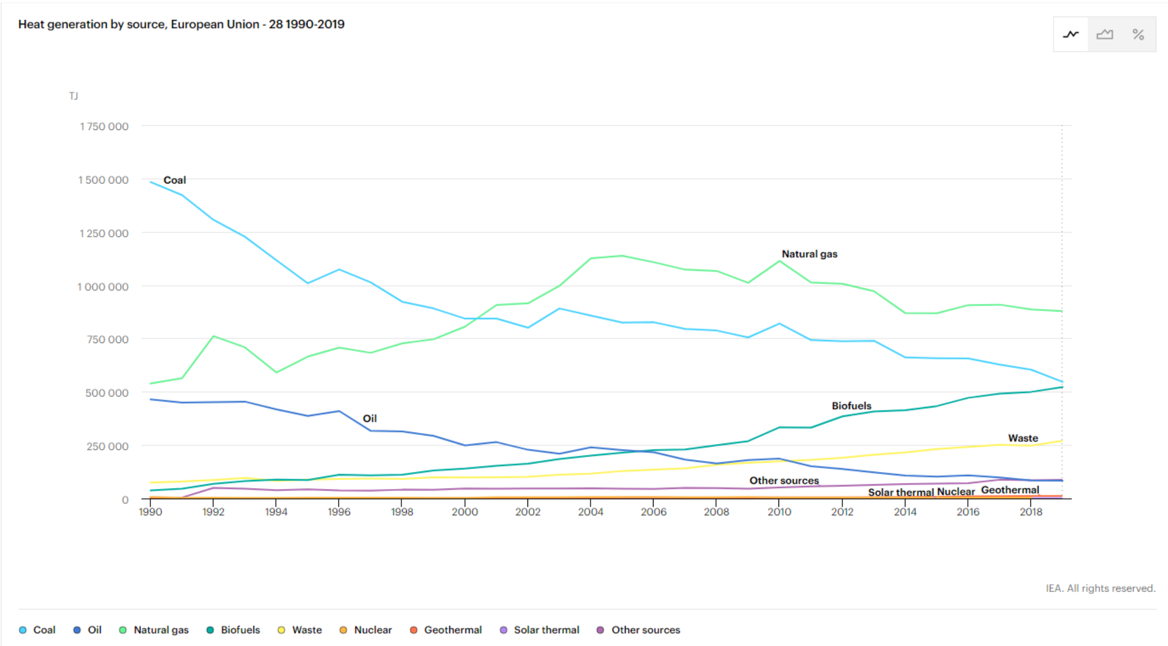

Figure 2.1. Evolution of the primary energy consumption used for heating purposes in the EU in the period between 1990 and 2019 (IEA - International Energy Agency 2021).

From the analysis of Figure 2.1 it is possible to verify that between 1990 and 2019 the fossil fuels (coal, oil, and natural gas) were the main sources of primary energy used for heating purposes in the countries that integrate the EU. However, this is a tendency that has been diminishing over the 29 years analyzed. This is no surprise, due to the many policies and programs launched by the EU to attenuate and solve the problems caused by the intensive use of fossil fuels.

A change in the heating industry is imperative. To fight the climate change problem, before it becomes irreversible, renewable and clean energies must become the norm for heating and other purposes all around the world. However, the proper functioning of renewable technologies, such as solar thermal collectors, is highly dependent of the weather conditions. In periods with low solar exposure, this type of technology may not be able to assure the thermal needs of a residential dwelling. In these situations, it is necessary to use a backup system, typically a traditional natural gas system. These factors may make it unfeasible to invest in renewable energy technologies to assure the thermal needs at the residential level. Thus, an intermediate solution must be found for the transition from fossil fuels to clean and renewable sources of energy. That solution could be the use of heating technologies that use fossil fuels, preferentially natural gas, but that are more efficient than the technologies already installed. This is where micro combined heat and power (micro-CHP) systems step in (Chen *et al.* 2021).

2.2. Assessment of the technologies used for hot water production for space heating and sanitary purposes in European Union countries

As explained in section 2.1, the primary energy sources used for DHW and SH are primarily fossil fuels. This is noticeable in most of the technologies employed for these two purposes, as shown in the Figure 2.2.

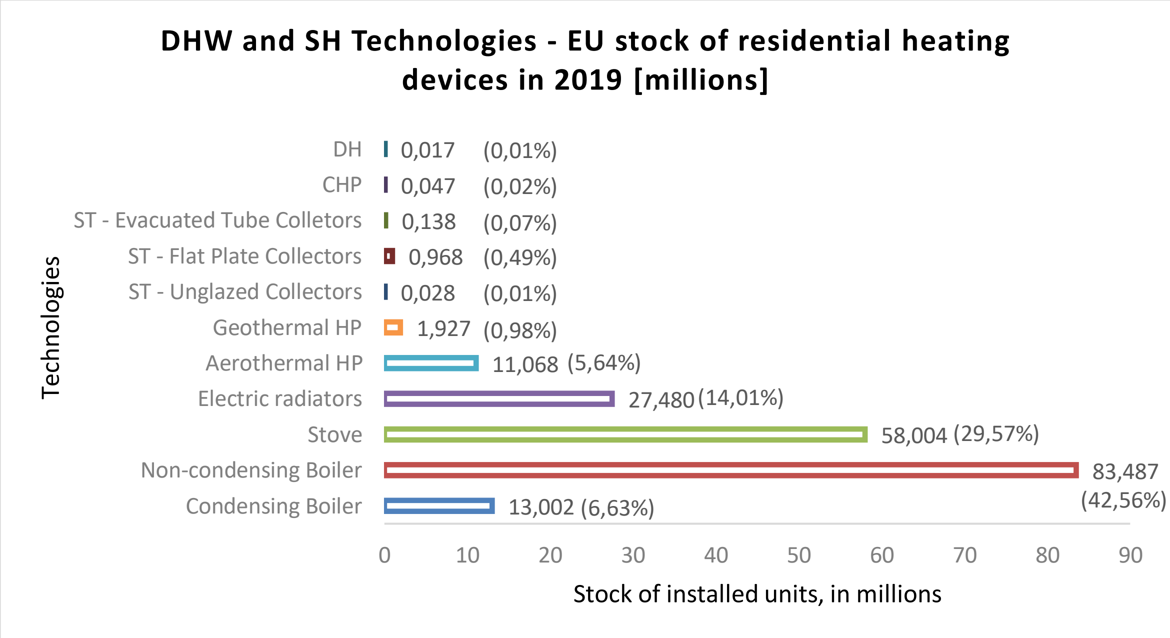

Figure 2.2. Stock of DHW and SH units/devices installed in the EU as of 2019, in millions, and their respective shares (Pezzutto *et al.* 2019b).

Analyzing Figure 2.2 it is possible to understand that boilers, both traditional (non-condensing) and condensing, account for more than 90 million units installed in the EU in 2019. This represents 49,19% of the total installed equipments for DHW and SH purposes. The other two dominant technologies/equipment types used for these purposes are stoves and electric radiators, with a share of 29,57% and 14,01%, respectively.

Accordingly to (Pezzutto et al. 2019a), the majority of the boilers operating in the EU in 2019 use natural gas as fuel. The numbers are: 66,23% for the condensing and 54,30% for the non-condensing boilers. As for stoves, they use wood as fuel, therefore a renewable source of energy. Lastly, electric radiators, as the name implies, make a direct use of electricity to warm up interior spaces or to produce domestic hot water.

It is expected that the largest heat demands occur in the northern European countries. This is justified by their climate conditions, harsher and more extreme that the ones found in southern and western European countries, like Portugal. This is more noticeable during the heating season. The research conducted by Chen and co-workers (Chen et al. 2021) allow him to conclude that the five northern EU countries that had the largest absolute heat demands in 2020 were: Netherlands (Nl), Poland (Pl), France (Fr), United Kingdom (UK) and Germany (Ge). Table 2.1 shows details regarding DHW and SH technologies for those countries. The countries are sorted in ascending order relatively to the total (DHW and SH) heat demands in 2020. The decentralized share accounts for the part of the total heat demand that was produced in decentralized systems. In the final three columns the three most used technologies for DHW and SH purposes are presented. It is also presented the number of units installed, in millions, for each technology in each of the referred countries.

It is important to notice that the data concerning the total heat demand and its respective decentralized share was obtained from (Chen et al. 2021) and concerns to 2020. However, the data regarding the most used technologies was retrieved from (Pezzutto et al. 2019a) and refers to 2019.

Table 2.1. Stock of DHW and SH units/devices installed, as of 2019, in the top five EU countries with the largest heat demands (Pezzutto et al. 2019a) (Chen et al. 2021).

Country	**Total Heat Demand in 2020 [TWh]**	**Decentralized share**	**Stock of DHW and SH units/devices installed as of 2019 (millions of units installed)**		
			First	*Second*	*Third*
Nl	131	86%	Non-condensing boiler (4,95)	Stove (0,98)	Condensing boiler (0,23)
Pl	263	75%	Stove (9,15)	Non-condensing boiler (3,99)	Electric Radiators (0,44)
Fr	438	94%	Non-condensing boiler (11,20)	Electric Radiators (9,50)	Stove (7,36)
UK	540	97%	Non-condensing boiler (16,50)	Electric Radiators (2,00)	Condensing boiler (1,24)
Ge	825	86%	Non-condensing boiler (14,63)	Stove (10,80)	Condensing boiler (5,92)

From the analysis of the Table 2.1 it is possible to verify that for the five analyzed countries, in a similar way to what was found for the entire EU (*c.f.* Figure 2.2), the most installed technology for DHW and SH purposes, as of 2019, was the non-condensing boiler. The exception is Poland, where the most installed technology for these purposes was the stove. The other technologies with large stocks include electric radiators and condensing boilers.

Therefore, the technologies that are going to be analyzed in section 2.3 and compared in section 2.4 are the boilers (traditional and condensing), the stoves and the electric radiators. Besides these technologies, other innovative and highly efficient solutions, such as micro-CHP systems and heat pumps (HP), will be studied in those sections. Although the renewable energy systems (RES) are promising and clean technologies, they will not be analyzed and compared in future sections. This decision was made due to their low number of installed units in the EU as of 2019, as can be seen in Figure 2.2. They represent less than 1% of the equipments installed for DHW and SH purposes in EU as of 2019, accordingly to (Pezzutto et al. 2019b).

Further analysis of Table 2.1 reinforces the importance of the decentralized equipments used for DHW and SH purposes in the residential sector. The decentralized share for the top five EU countries with the largest heat demands in 2020 varies between 75% and 97%. This shows that, although District Heating (DH) facilities can be an important alternative in the heating sector to fight climate change, their implementation is still sparse in the residential scenario, and decentralized systems and equipments still run the show.

2.3. Characterization of the technologies used for space heating and domestic hot water purposes in European Union countries

Residential heating systems produce hot water (or hot air) to provide space heating, hot water for domestic purposes or both. This type of systems that use water as the heat transfer medium, called hydronic systems, must overcome several technical challenges. In terms of DHW, highly variable demands during the day must be suppressed in a very short period, as it is expected by European consumers (Pomianowski et al. 2020). For SH, hot water must be produced at a higher temperature than the ones used for domestic purposes.

This temperature depends on the type of technology implemented (*e.g*: heat transfer devices with natural or forced convection) (Pereira et al. 2018).

In this section, each of the previously mentioned technologies will be analyzed in depth. The parameters that are going to be reviewed are: i) main components; ii) working principles; iii) outputs. Moreover, a SWOT analysis will be carried out for each technology, identifying its strengths, weaknesses, opportunities, and threats.

2.3.1. Non-condensing boiler

Although this technology still has the largest stock of installed units in the EU, for SH and the production of DHW, it is expected that their installed capacity will be replaced by more efficient heating technologies, like its direct technological evolution, the condensing boiler (Martinopoulos, Papakostas, and Papadopoulos 2018).

The non-condensing boiler can integrate a hydronic central heating system. This implies that the heat is produced in a central equipment, in this case a non-condensing boiler. Water is used as the heat transfer medium. Usually, it is required that the central equipment be installed in an appropriate space in the building, usually in the basement and called boiler room (Martinopoulos, Papakostas, and Papadopoulos 2018).

This type of boiler can also be used as a standalone system, providing only hot water for domestic purposes on demand. This type of system usually has a wall hanging design (Martinopoulos, Papakostas, and Papadopoulos 2018).

A central heating system consists in three different subsystems: i) heat production subsystem: (non-condensing) boiler and burner; ii) distribution subsystem: circulation water pump, fuel tank, piping, and radiators; iii) control subsystems: safety devices and control equipment (examples: sensors and actuators) (Martinopoulos, Papakostas, and Papadopoulos 2018).

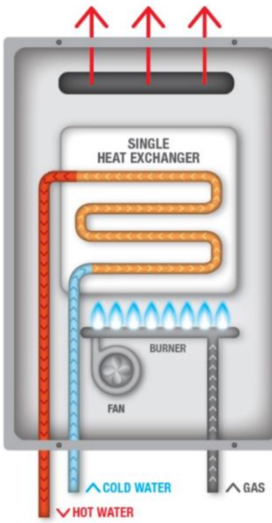

Figure 2.3. Basic schematics of a standalone non-condensing boiler (Hot Water Solutions - How Do Condensing Water Heaters Work? 2021).

The working principle is quite simple and can be easily understood with the help of Figure 2.3. The fuel is supplied to the burner. The combustion of this fuel in the burner, in conjunction with air, produces hot combustion gases. This hot combustion gases exchange heat with the water in the heat exchanger of the boiler. The water is supplied to the heat exchanger and extracted from it via the water pipes. The result of the process is hot water used for domestic purposes and/or for SH. The exhaustion system integrated in the boiler extracts the cooled combustion gases after the heat exchange. The hot water used for domestic purposes is used directly on taps. In case of a boiler that also provides hot water for SH, this water is distributed via water pipes to a system of radiators, for example (Boiler Guide UK 2021).

It is also possible to use different heat emitters than the radiators, such as convectors or underfloor piping (also known as a radiant floor system) (Martinopoulos, Papakostas, and Papadopoulos 2018).

The fuel used can be natural gas, propane or butane gas, oil, or wood. As seen previously in section 2.2, most of the non-condensing boilers installed in the EU as of 2019 used natural gas as fuel.

Another important feature is that, depending on the type of non-condensing boiler used, the water can be heated on demand or it can be stored in a storage tank (Keinath and Garimella 2017).

Table 2.2. SWOT analysis of non-condensing boilers.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Fuel flexibility	Wasted energy trough the combustion gases	Low acquisition cost	EU legislation
Mature technology	Use of fossil fuels	-	Condensing boilers
Good lifespan	Less efficient than other heating technologies	-	Renewable energy technologies

2.3.2. Stove

This type of heating equipment was the second most installed in the EU as of 2019. It uses wood or its subproducts as fuel. The most popular subproduct is pellets. Wood is considered biomass, therefore a renewable source of energy (Pezzutto et al. 2019b).

Wood and pellet stoves only provide heat for SH, unless they have a boiler attached in its back with the purpose of producing hot water for domestic use. This is not a very common solution. Therefore, stoves will only be considered for SH purposes (Martinopoulos, Papakostas, and Papadopoulos 2018).

Usually stoves only provide heat to one room trough forced air ventilation, being considered local heating systems. This air is heated through the combustion of wood or one of its subproducts (Martinopoulos, Papakostas, and Papadopoulos 2018).

In order to assure a good efficiency, stoves must be properly insulated and have a correct, regulated air flow and ventilation system (Martinopoulos, Papakostas, and Papadopoulos 2018). Other important feature is the use of baffles to create a longer heat exchange area between the air and the combustion gases. The employment of a catalytic converter is also beneficial, since it reduces the number of pollutants emitted, which are caused by an incomplete combustion of the fuel (Rise: Wood Stoves - An Efficient and Economical Way to Heat Your Home 2020).

Wood-burning stoves have a maximum efficiency of 75%, whereas pellet-burning stoves efficiency can go as high as 90%. Pellet-burning stoves are more efficient because their design allows for a more efficient and optimized combustion process (Martinopoulos, Papakostas, and Papadopoulos 2018).

Besides efficiency, pellet-burning stoves have more advantages over wood-burning stoves, such as fewer emissions of gaseous pollutants, the fact that pellet ignition and feeding to the stove can be automated, and their maintenance is easier, because less ashes are produced during the combustion process (Martinopoulos, Papakostas, and Papadopoulos 2018).

Although stoves may seem a practical and pleasant solution for heating a room in a residential dwelling, as the living room, they have a few inconveniences, such as the following ones: i) they require electricity to power fans, controls and other optional systems (example: pellet feeder); ii) it is necessary to have a storage room to store the wood or the pellets; iii) the combustion of wood produces smoke and soot, which are a significant source of air pollution; iv) the use and consumption of wood contributes to the deforestation of the planet (Martinopoulos, Papakostas, and Papadopoulos 2018) (Rise: Wood Stoves - An Efficient and Economical Way to Heat Your Home 2020).

Table 2.3. SWOT analysis of wood and pellet stoves.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Cheap to run	Emission of particulate pollutants	Utilization in rural areas (good fuel availability)	Biomass boilers
Easy to install and operate	Manual handling of the fuel	Replacement of more complex and expensive systems	Renewable energy technologies
Good efficiency	Only provide SH for one room	Good aesthetic impact	-

2.3.3. Electric Radiator

Electric radiators were the third most installed device for heating in the EU as of 2019, according to the data from (Pezzutto et al. 2019b). This type of technology can only be used for SH purposes, and it is considered a local heating system.

Their working principle is very simple. They produce heat through the conversion/dissipation of electric current, using electric resistances. This type of electric heaters provide heat to surfaces directly in line with the heater, by emitting infrared radiation. The surfaces heated by this type of device are usually walls, objects, and occupants of a certain room. This implies that they do not provide heat directly to the air inside the space being heated. Electric radiators fall into the more general category of electric heaters. Within these there are also electric devices that do space heating by natural or forced convection (Martinopoulos, Papakostas, and Papadopoulos 2018).

Electric radiators work best when used for short heating periods and for heating spaces that are not permanently occupied, like a guest room in a residential dwelling (Martinopoulos, Papakostas, and Papadopoulos 2018).

Table 2.4. SWOT analysis of electric radiators.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Low acquisition costs	Only provide SH for one room	Occasional room heating	Heat Pumps
Low maintenance	High operation cost	Use of excess electricity produced by renewable sources	Stoves
Long lifecycle	Low thermal comfort	Portable heating	Micro-CHP systems

Another type of electric equipments worth mentioning are electric water heaters. These devices use an electric resistance to heat water. They can be tankless or have a tank, producing hot water on demand or producing and storing it in a cylinder, respectively. Depending on their dimensions and power, they can be used to produce DHW alone or combine it with SH purposes. Electric water heaters used for DHW and SH purposes are generally integrated in a central heating system, like the ones used with boilers (Keinath and Garimella 2017).

Electric water heaters have few moving parts and long lifecycles, making them cheap to buy and maintain, when compared to similar equipments. However, since electricity is the only usable energy source, their operation can be economically unviable if electricity tariffs are pricy, making this equipments more expensive to run then a natural gas boiler, for example (Keinath and Garimella 2017).

Table 2.5. SWOT analysis of electric water heaters.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Low acquisition costs	High operation cost	Integration with PV technology	Heat Pumps
Low maintenance	Dependence of electricity grid	Retrofitting of non-condensing boilers	Condensing boilers
Long lifecycle	Limited heating capacity	Integration with smart grids	Micro-CHP systems

2.3.4. Condensing boiler

A condensing boiler is very similar to a non-condensing one, with the exception that it uses two heat exchangers. This secondary heat exchanger is used to pre-heat the cold water entering the boiler.

The working principle is also very similar to what as explained in section 2.3.1. In this type of boiler, the combustion gases resulting from the combustion process go through two heat exchangers. By increasing the heat exchanging area, the temperature of the combustion gases that exit the boiler is reduced up to the point where a significant condensation of the water vapor present in the mentioned combustion gases is observed. This condensation process occurs in the secondary heat exchanger, allowing the recovery of the latent heat from the combustion gases. Due to this technological difference when compared to a conventional boiler, the condensing boiler has a higher thermal efficiency, resulting in lower running costs (Che, Liu, and Gao 2004).

According to the website of the European Heating Industry (European Heating Industry 2021), the benefits of using a condensing boiler rather than a non-condensing one are the following: i) better energy efficiency; ii) lower emissions of pollutant gases; iii) easily retrofittable; iv) easy installation and maintenance; v) reliable technology.

Table 2.6. SWOT analysis of condensing boilers.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
High efficiency	Use of fossil fuels	Retrofit a non-condensing boiler	Heat pumps
Easy installation and maintenance	Emission of pollutant gases	Well-established market and manufacturers	Decarbonization
Reliable technology	Requirement of an exhaust system	High availability of natural gas	Micro-CHP systems

2.3.5. Heat Pump

The heat pump technology is considered to be highly efficient (Pomianowski et al. 2020). Although its low number of installed units in the EU countries as of 2019 (Pezzutto et al. 2019b), it is expected that this technology will become a major player in the residential heating scenario (Slorach and Stamford 2021).

This technology uses a vapor-compression cycle to transfer thermal energy from a cold heat source to a warmer heat sink. The vapor-compression cycle uses a refrigerant fluid as the working fluid (Pomianowski et al. 2020).

The heat pump is mechanical device that is composed of four main components: an evaporator, a condenser, an expansion valve and a compressor. These four components are connected in a closed hydronic circuit in which circulates the previously mentioned refrigerant fluid (Pomianowski et al. 2020).

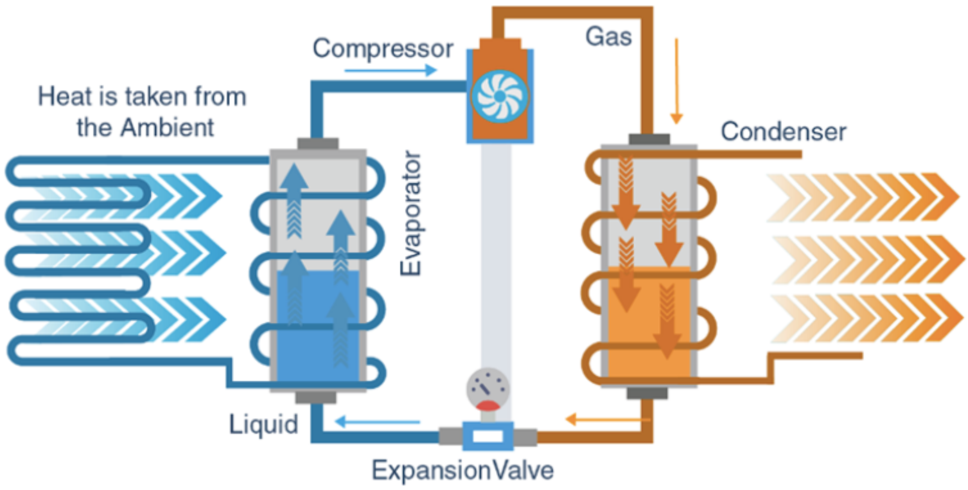

Figure 2.4. Basic schematics of an air source heat pump. Adapted from (Aspiration Energy - Heat Pumps 2020).

As it can be seen in Figure 2.4, the vapor-compression cycle starts with the evaporation of the refrigerant fluid in the evaporator. In the case of Figure 2.4, the heat source used in the evaporation process is the outdoor air. Then the hot and gasified refrigerant fluid is pressurized in the compressor. After that, the pressurized and hot refrigerant fluid rejects heat to a heat sink via the condenser. Finally, the refrigerant fluid, in liquid state, is expanded in the expansion valve. Then the cycle repeats.

In the residential sector, heat pumps can be used for both DHW and SH purposes, and for space cooling as well, during the cooling season. For these purposes, the compressor is usually electric driven. The heat source can be the outdoor air or the ground. Finally, the heat sink is the interior spaces to which it is intended to supply/remove heat or a water storage cylinder integrated in a central heating system (Pomianowski et al. 2020).

The efficiency of heat pumps is measured using the coefficient of performance (COP), that is usually higher than one. The COP of the system varies greatly with the temperatures of the heat source and the heat sink. The higher the temperature of the source and the lower the sink temperature (also referred to as the output temperature), the higher is the COP value (Pomianowski et al. 2020).

As for the sources, air source heat pumps have lower installation costs than the ground sources ones, but their seasonal COP is significantly lower. This is due to the temperature of the air being lower than the temperature of the ground during the heating season (Pomianowski et al. 2020).

In similarity to boilers, heat pumps are part of a hydronic centralized heating system. However, the heat production system is composed only by the heat pump, there is no need for a fuel tank and different solutions are normally used for heat emission rather than radiators (Martinopoulos, Papakostas, and Papadopoulos 2018).

Finally, it is possible to add a hot water storage tank in a heat pump central heating system. This allows to maximize the operation of the compressor at an optimum load ratio, improving the average COP. The addition of this hot water tank can be mandatory in the cases where the power of the heat pump does not allow the domestic hot water demand to be met in a timely manner, since the desired waiting time of the user is reduced (Pomianowski et al. 2020).

Table 2.7. SWOT analysis of heat pumps.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
High efficiency	Dependence of electricity grid	Retrofit of boilers	Condensing boilers
Usable for heating and cooling	Complex system	Integration with renewable energy systems	Micro-CHP systems
Easy access to heat sources	Higher acquisition cost than boilers	Implementation in NZEB's	Renewable energy technologies

2.3.6. Micro-CHP systems

Micro scale cogeneration or micro-CHP (combined heat and power) systems are a miniature version of the regular cogeneration systems. This regular cogeneration systems have been used for more than 100 years to combine the production of heat and electricity. The micro-CHP systems are believed to play an important role in the transformation of the energy sector, especially in the EU. This is due to their lower emissions of CO_2 and primary energy savings, when compared to other technologies currently used for DHW and SH purposes in residential buildings (Murugan and Horák 2016).

The CHP systems scale is classified based on the equipments electrical power output. Micro-CHP systems have an electrical capacity up to 15 kW_e (Murugan and Horák 2016).

It is expected that the implementation of micro-CHP systems will bring various benefits to its users and the environment, such as: i) reduction of the emissions of greenhouse gases (GHG); ii) contribution to the decentralization of energy supply; iii) potential reduction in the energy cost to consumers (Murugan and Horák 2016).

The more attractive implementations for micro-CHP systems are buildings where a significant thermal demand is combined with an electric demand. Some examples are residential buildings, hospitals, and supermarkets (Murugan and Horák 2016).

Besides their thermal and electrical efficiencies, there is another parameter that is important to assess when it comes to micro-CHP systems. This parameter is the power-to-heat ratio. This non dimensional parameter represents the amount of electrical power generated for every unit of heating power that is produced in a micro-CHP system. It is an important parameter in the residential heating scenario because it allows to determine the electrical power produced from the micro-CHP system to satisfy a given thermal demand of the building. The power-to-heat ratio will be assessed for each micro-CHP technology, while their electrical and thermal efficiencies will be presented in Table 2.13.

There are five types of micro-CHP systems that are currently being developed and studied all around the world. Their characterization will be made in the next sections.

2.3.6.1. Reciprocating Engine

This CHP technology uses an internal combustion engine (ICE), working accordingly to same principles as the ones used in the automotive industry, to produce heat and power at various scales. Reciprocating engines are efficient systems and can achieve long-term availability levels between 85% and 92% (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

The main components of a reciprocating engine (RE), as it can be seen in Figure 2.5, are an ICE, an alternator, two cooling systems, one for the lubricant oil and another for the engine, a heat exchanger for the combustion gases, and an exhaust system.

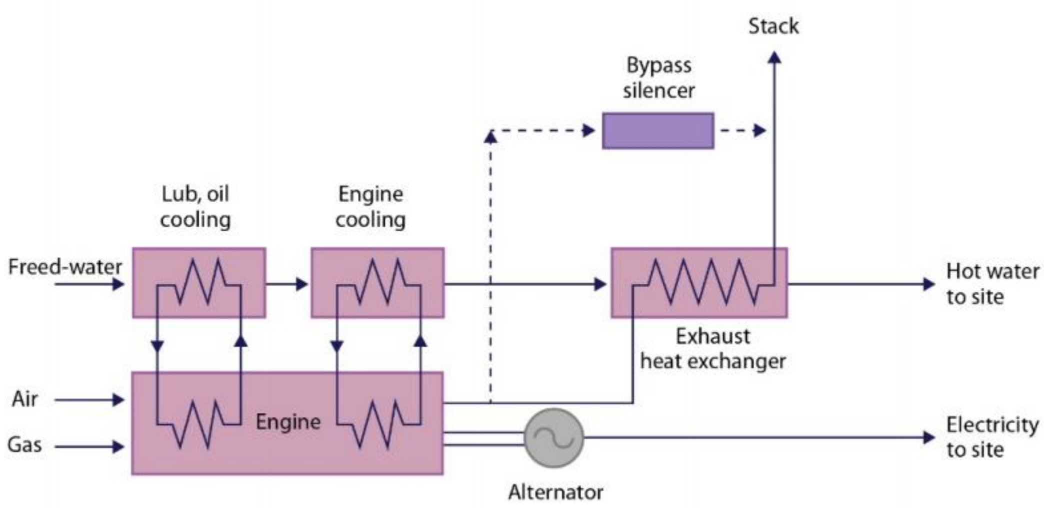

Figure 2.5. Basic schematics of a reciprocating engine (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

Through the interpretation of Figure 2.5, the working principle of a micro-CHP system based on the RE technology can be understood. Firstly, a mixture of air and fuel is injected into the ICE. In the ICE, a combustion process occurs, producing mechanical energy, which is transmitted to the alternator via a shaft. The alternator converts the mechanical energy into electrical energy. Since engines and their lubricating oil require cooling, two cooling systems remove heat from them and use it to heat water. Finally, the hot combustion gases, which are a byproduct of the combustion process, reject heat to water stream via a heat exchanger. Then the cooled combustion gases are extracted trough an exhaust system. The final products of a reciprocating engine operation, when applied in a CHP context, are hot water and electricity.

This type of systems produce heat from two different sources: i) engine exhaust gases, with temperatures up to 400°C; ii) engine and lubricating oil cooling systems, which reject heat to water, warming it up to 120°C. Exhaust gases and cooling systems have an even contribution to the total amount of heat produced in reciprocating engines (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

Accordingly to (Murugan and Horák 2016), the power-to-heat ratio of micro-CHP systems based on the reciprocating engine technology is 0,38.

Some perks of reciprocating engines are better efficiency than MGT (micro gas turbine) based micro-CHP systems and good performance at part-load operation (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

Table 2.8. SWOT analysis of a micro-CHP system using the reciprocating engine technology.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Potential of primary energy savings	Use of fossil fuels	Retrofit of boilers	Other micro-CHP systems
High efficiency	Developing technology	Supply of high thermal needs	Heat pumps
Good performance at part load operation	High acquisition cost	Supply of a share of the electricity consumed in a household	Renewable energy technologies

The classification of reciprocating engines used for CHP applications is made accordingly to their method of ignition. The ignition can be made through spark or compression (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

2.3.6.1.1. Spark-ignition Gas-engines

As the name implies, this type of reciprocating engines ignites the air-fuel mixture using a spark. Also, they only use gaseous fuels, like natural gas (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

Spark-ignition gas-engines are more appropriated for smaller and simpler CHP installations, like residential dwellings. The heat production systems, which remove heat from the lubricant oil, the engine and the exhaust gases, can deliver low/medium temperature hot water on site (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

When compared to compression-ignition engines, spark-ignition ones have some advantages: i) lower capital cost per kW; ii) combustion with less excess air; iii) the exhaust gases transport less heat. However, they have a few downsides, such as: i) lower shaft efficiency, up to 35%; ii) need of spark plugs to operate (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

2.3.6.1.2. Compression-ignition Engines

This type of reciprocating engines, also known as diesel engines, ignite the fuel during the combustion process by compressing it. They can run on “gas-oil” and heavy residual fuel oils. They are more complex systems than a spark-ignition gas-engine because they use turbocharges and intercoolers (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

Compression-ignition engines are typically used for large-scale CHP installations, with an output range from 1MW to 15MW. This means that their use in residential applications is highly unfeasible (Combined Heat and Power Technologies - A detailed guide for CHP developers 2021).

2.3.6.2. Stirling Engine

Stirling engines (SE) are external combustion engines that are suited for micro-CHP applications due to their fuel flexibility and high efficiency (Skorek-Osikowska et al. 2017).

As it can be seen in Figure 2.6, the main components of a SE are three heat exchangers, which function as a heater, a regenerator, and a cooler, and two cylinders, one including the driving piston and another including the sliding/displacement piston (Skorek-Osikowska et al. 2017).

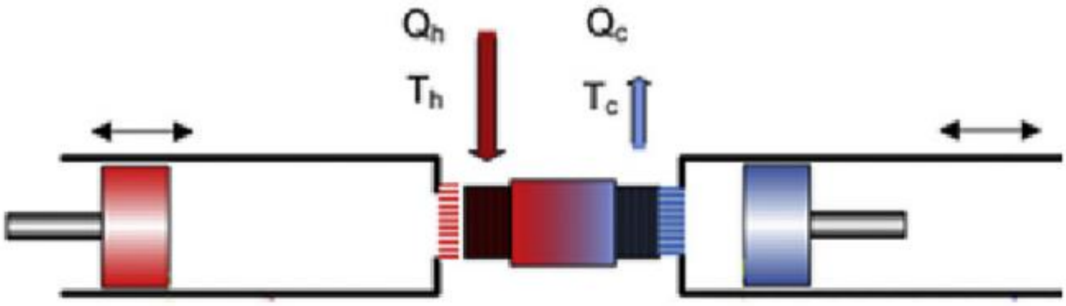

Figure 2.6. Basic schematics of a Stirling Engine with a two cylinder disposition (types α and γ) (Skorek-Osikowska et al. 2017).

This type of engines is classified in three different types: α, β and γ. The α-SE as an angle between the cylinders close to 90°, disposed in a V-arrangement like a classic ICE. These cylinders are connected through the three heat exchangers, that function as a heater, a cooler and a regenerator. The β-SE only uses one cylinder, containing in it both pistons (driving and sliding). The heat exchangers are connected to the top and bottom of the working space of the sliding piston, acting as the heater, the cooler, and the regenerator of the SE. At last, the γ-SE is structurally like the β type. However, it uses a separate cylinder for the working gas displacement, maintaining the disposition of the heat exchangers (Skorek-Osikowska et al. 2017).

The working principle of an SE is complex. It can be better understood with the help of Figure 2.6: i) heat is delivered to the working fluid; ii) the working fluid exchanges heat with the heater; iii) the heater supplies heat to the cylinder containing the driving piston, causing an increase of the pressure over this piston, forcing it to move; iv) the driving piston transmits movement to the sliding piston; v) the movement of the sliding piston causes a change in the position of the displacer, which is inside the cooler; vi) this change in the position of the displacer results in the pumping of the working fluid from the heater, through the regenerator, into the cooler; vii) this leads to heat dissipation in the cooler, which drops the pressure inside the cylinders, forcing them to return to the original position; viii) this final movement of the pistons pumps the working fluid back into the heater (Skorek-Osikowska et al. 2017). The heat rejected in the cooler can be used to produce hot water, while the movement of the pistons produces mechanical energy, that can be converted in electricity by coupling the SE with a generator.

Accordingly to (Murugan and Horák 2016), the power-to-heat ratio of micro-CHP systems based on the Stirling engine technology is 0,2.

Due to its complex working principle, which requires correct balancing of the pistons, proper sealing of the SE and an external heat source, micro-CHP systems based on SE technology are often more expensive than its micro-CHP counterparts. Another problem besides the complexity of SE when applied to micro-CHP, is its weight, which can compromise the use of this systems in retrofitting applications (Murugan and Horák 2016).

Table 2.9. SWOT analysis of a micro-CHP system using the Stirling engine technology.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Heat source flexibility	Complex system	Retrofitting boilers	Other micro-CHP systems
High efficiency	High acquisition cost	Integration with renewable energy systems	Heat pumps
Potential of primary energy savings	Heavy system		Renewable energy technologies

2.3.6.3. Micro-gas turbine

The micro gas turbine (MGT) technology is another alternative within the micro-CHP systems. Their implementation offers a lot of perks, such as: i) ability to work with various fuels; ii) low emission levels; iii) high power density; iv) low maintenance (Murugan and Horák 2016).

A basic configuration of an MGT includes four components, which are a compressor, a combustor, a turbine, and a regenerator (generator). Most configurations use only a shaft, which connects the compressor and the turbine. The more complex configurations use a heat recovery system, called regenerator, to preheat the combustion air using the hot combustion gases. This more complex configuration can be seen in Figure 2.7. The regeneration can increase the electrical efficiency of the system up to the double, but it reduces the amount of heat rejected that is used for the production of hot water (Murugan and Horák 2016).

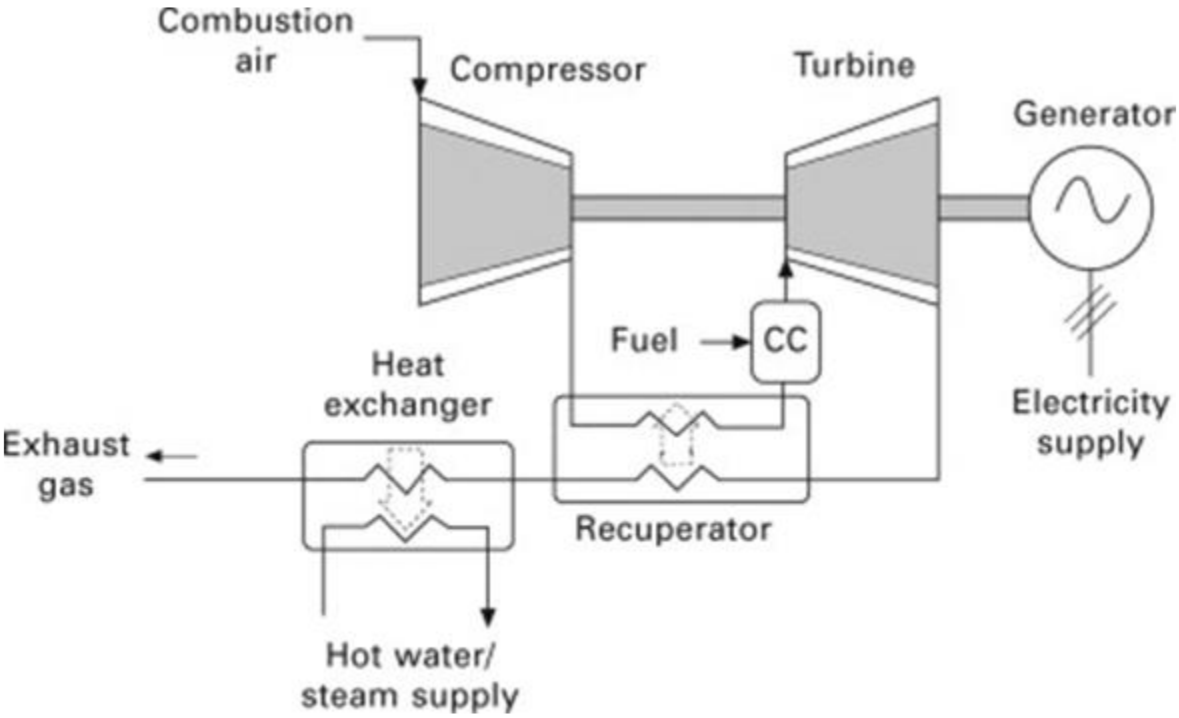

Figure 2.7. Schematics for a micro gas turbine incorporating a heat recovery system (ScienceDirect - Micro Gas Turbine 2019).

Analyzing Figure 2.7 it is possible to deduce the working principle of this type of systems. Firstly, combustion air enters the compressor, where it is pressurized. Then this pressurized combustion air is injected along with fuel in the combustor (CC in Figure 2.7), occurring a combustion process that produces hot combustion gases. After that, the hot combustion gases are expanded in a turbine, producing work. Since the turbine is coupled with a generator the work produced in form of mechanical energy is converted into electrical energy. Later, the hot expanded combustion gases exchange heat in the regenerator with the combustion air, preheating it. Finally, the combustion gases go through another heat exchanger where they rejected heat to water, producing hot water for SH and domestic purposes.

As far as the components are concerned, usually the compressor and the turbine have a radial-flow design, like it is used in the automotive industry for turbochargers. The generator typically used in this micro-CHP systems is a high-speed permanent magnet generator, which produces variable voltage and frequency alternating current (AC) power. Also, the MGT systems only work with gaseous fuels, like natural gas (Murugan and Horák 2016).

Currently designed and produced MGT's work in permanent regime and use a heat recovery system. Currently, the biggest challenges for the MGT technology are adjusting the turbine inlet temperature and maximize the regenerator efficiency (Murugan and Horák 2016).

Accordingly to (Murugan and Horák 2016), the power-to-heat ratio of micro-CHP systems based on the micro gas turbine technology is 0,33.

In conclusion, it is expected that, with the maturing of the MGT technology, this type of equipment can be used to fulfill the thermal demands and cover a part of electrical demands of multifamily residential, commercial, and educational buildings (Murugan and Horák 2016).

Table 2.10. SWOT analysis of a micro-CHP system using the micro gas turbine technology.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Low emission levels	Low thermal efficiency	Retrofitting boilers	Other micro-CHP systems
Low maintenance	Use of fossil fuels	Replacement of small SH and DHW systems	Heat pumps
High power density	High acquisition cost	Supply of a share of the electricity consumed in a household	Renewable energy technologies

2.3.6.4. Organic Rankine Cycle

Within the micro-CHP technologies, some authors believe the organic Rankine cycle (ORC) based one is the most promising of them all, regarding the residential sector. Two major reasons that justify this claim are the simplicity of the ORC technology when compared to other micro-CHP alternatives, and its ability to retrofit currently used heating systems, like boilers. Thus, the ORC based micro-CHP systems are a promising solution to convert low-grade heat into power, typical in residential buildings (Pereira et al. 2018).

An ORC is a classic Rankine cycle where the working fluid is an organic fluid instead of water/steam. The basic configuration of a Rankine cycle involves four components: a pump, an evaporator, a turbine, and a condenser. In a micro-scale ORC the turbine is usually replaced with an expander. This expander is coupled with a generator to produce electricity (Pereira et al. 2018). This basic ORC configuration can be seen in Figure 2.8.

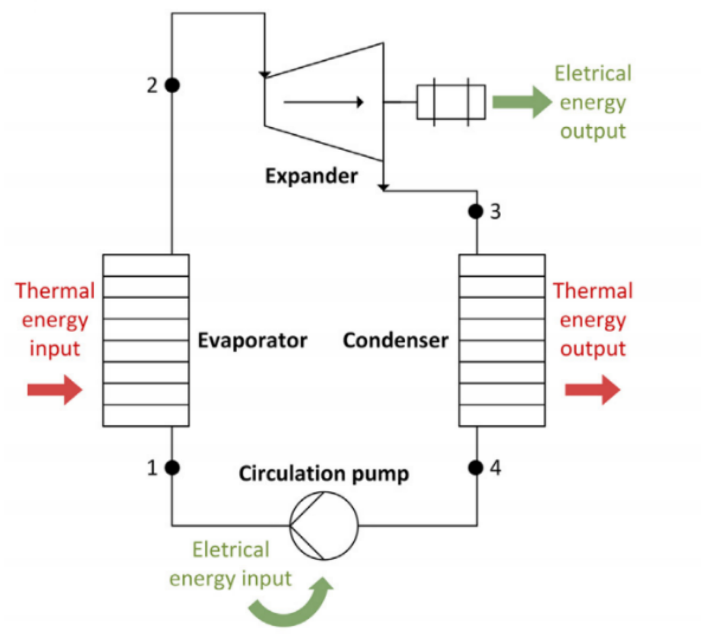

Figure 2.8. Schematics of a basic configuration of an ORC (Pereira et al. 2018).

Through the analysis of Figure 2.8 it is possible to understand the working principle of an ORC based equipment. First, the organic fluid is pressurized in the electric driven pump. Then, the pressurized organic fluid is vaporized in evaporator. In the evaporator, which is essentially a heat exchanger, the heat is supplied by an external source. This external source can be, for example, a natural gas burner. Later, the hot and pressurized organic fluid is expanded in the expander, where mechanical work is produced. The generator coupled to the expander converts this mechanical work into electrical energy. Finally, the low pressure and hot organic fluid, in the form of vapor, goes through the condenser, in which it rejects heat to a low temperature heat sink. This heat sink is usually water. The results of this process are electricity and hot water, two typical needs of a residential building.

Accordingly to (Pereira et al. 2018), the power-to-heat ratio of micro-CHP systems based on the organic Rankine cycle technology is 0,067.

An organic fluid is used instead of water/steam to work with saturated vapor at the expander entrance (point 2 in Figure 2.8). This is advantageous because it allows the cycle to work with lower pressures than the equivalent water/steam Rankine cycle. It also avoids the need of superheating the working fluid. This is possible because organic fluids have a vertical or positive slope of the saturated vapor-line in a T-s diagram. This means that the final stage of expansion in the expander occurs outside the liquid-vapor dome, which implies that only vaporized organic fluid goes through the expander. However, when ORC based micro-CHP systems are not well designed, there is the risk of thermal degradation of the organic fluid if the temperature in the evaporator surpasses a certain limit (Pereira et al. 2018).

According to (Pereira et al. 2018), the number of patents regarding ORC technology has been rapidly increasing since the beginning of this century. Also, there are a significant number of ORC based system manufacturers operating in the market. These two statements reinforce the idea that micro-CHP systems based on ORC technology have an important role to play in the heating industry in the next few decades. However, the residential scale implementation of the ORC technology is still scarce and expensive. Progress must be made in these aspects to spread this technology as a safe, viable and cost-effective alternative to traditional residential heating systems.

Table 2.11. SWOT analysis of a micro-CHP system using the organic Rankine cycle technology.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Simpler system than other micro-CHP technologies	High acquisition cost	Retrofitting boilers	Other micro-CHP systems
Heat source flexibility	Few manufacturers and distributors on the market	Integration with renewable energy systems	Heat pumps
Good electrical and thermal efficiency	Risk of thermal degradation of the organic fluid	Supply of a share of the electricity consumed in a household	Renewable energy technologies

2.3.6.5. Fuel cell

The fuel cell technology has been discussed in the past years as a very innovative and promising technology, especially in the automotive industry. However, it can also be applied to the production of heat and power for residential buildings.

In the fuel cells, an energy conversion process occurs, in which the chemical energy available in hydrogen molecules is converted into electricity. During this process, a fraction of the unused energy becomes available as heat. The use of a fuel processing unit is necessary to convert hydrocarbons into hydrogen (Murugan and Horák 2016).

During the operation of a fuel cell, heat is generated because of the entropic heat of the chemical reactions, the irreversible heat of electrochemical reactions, Joule’s effect, and the condensation of water vapor. This generated heat must be removed in an efficient way to prevent the fuel cell and its components from overheating (Murugan and Horák 2016).

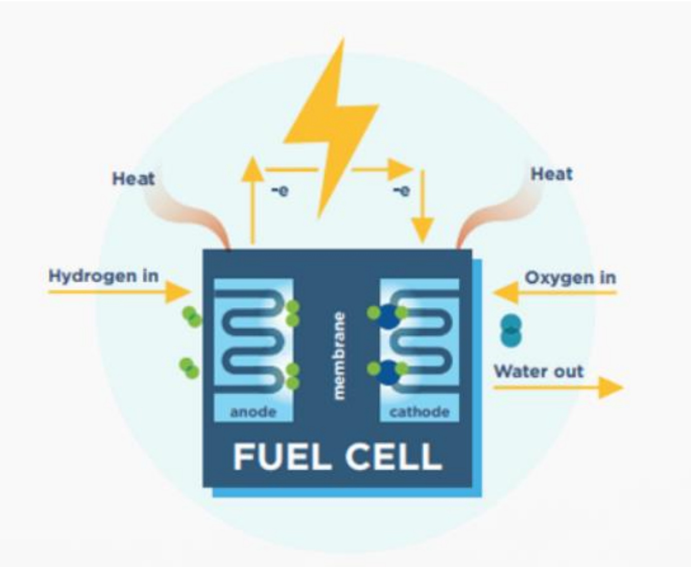

Figure 2.9. Illustrative schematic of a fuel cell (Fuel Cell & Hydrogen Energy Association (FCHEA) - Fuel Cell Basics 2021).

Analyzing Figure 2.9 helps to understand the basic principles of a fuel cell operation. First, the hydrogen atoms enter the anode of the FC. Then, the anode removes electrons from the hydrogen molecules. After that, the protons resultant from the last process pass through the membrane into the cathode. This forces the electrons to follow an external circuit, where electricity is produced. Finally, the protons and the electrons are combined with oxygen in the cathode. This last step produces two byproducts of the fuel cell operation: water and heat (Fuel Cell & Hydrogen Energy Association (FCHEA) - Fuel Cell Basics 2021). These various steps explain the electrochemical reaction of fuel (hydrogen) in the anode side and an oxidant (oxygen) in the cathode side, with the concomitant flow of ions trough the electrolyte (membrane) and the generation of an electric current in the external circuit (Murugan and Horák 2016).

Accordingly to (Murugan and Horák 2016), the power-to-heat ratio of micro-CHP systems based on the fuel cell technology is 1.

Although a promising technology, the fuel cell still requires further overall improvement, in order to maximize its efficiency, lower its production costs, and mitigate the problems associated with the use of hydrogen as fuel (Murugan and Horák 2016).

Table 2.12. SWOT analysis of a micro-CHP system using the fuel cell technology.

SWOT analysis			
Strengths	**Weaknesses**	**Opportunities**	**Threats**
Low emission of pollutants	Hydrogen is a volatile compound	Retrofitting boilers	Other micro-CHP systems
Variety of fuel cell technologies	High acquisition cost	Implementation in NZEB's	Heat pumps
High efficiency	Developing technology	Potential state incentives due to the use of a green fuel (hydrogen)	Renewable energy technologies

There are three main types of fuel cells: i) Low temperature polymer electrolyte membrane fuel cell (PEMFC); ii) High temperature solid oxide fuel cell (SOFC); iii) Molten carbonate fuel cell (MCC) (Murugan and Horák 2016).

2.3.6.5.1. Low temperature polymer electrolyte membrane fuel cell (PEMFC)

This type of fuel cell operates at around 80°C, which can be considered low temperature operation. The cell itself is formed by a proton exchange membrane sandwiched between two electrodes. PEM fuel cells have high power density and the ability to vary their output quickly, to respond to variations in the power demand. The material used for the membrane, an electrolyte polymer, is used due to its ability to conduct hydrogen atoms (Murugan and Horák 2016).

2.3.6.5.2. High temperature solid oxide fuel cell (SOFC)

SOFC's use hydrocarbons as fuel, converting them directly into electricity. Their high operating temperatures, ranging from 800°C to 1000°C, allows them to use various types of hydrocarbon fuels. These high operating temperatures enable various applications for this type of fuel cells, such as cogeneration in a stationary regime (Murugan and Horák 2016).

The strengths of SOFC's are: i) high electrical efficiency; ii) high-grade waste heat; iii) fuel flexibility; iv) low emissions of pollutants; v) power scalability (Murugan and Horák 2016).

In a system based on SOFC technology, four different process streams are necessary: heat, fuel, air, and water. The system also requires an electrical power conditioning unit and a control system to manage the pumps, sensors, and actuators of the system. In this type of system, the heat is removed from the stack using a stoichiometric air supply to the cathode as the cooling medium (Murugan and Horák 2016).

2.3.6.5.3. Molten carbonate fuel cell (MCFC)

The MCFC has a molten carbonate electrolyte. This very specific electrolyte has poor conductivity at low temperatures and at high temperatures there is a risk of its accelerated corrosion and vaporization. These factors limit its operating temperature interval, which is between 580°C and 700°C, that indicates that the MCFC is a high temperature fuel cell (Murugan and Horák 2016).

Systems that use MCFC's can operate at atmospheric pressure or higher pressures. Pressurized MCFC systems are used to improve the power density, which can be beneficial from an economic point of view. However, high pressure usage of MCFC's can cause the dissolution of the cathode (Murugan and Horák 2016).

2.4. Comparison of the technologies used for space heating and domestic hot water purposes in European Union countries

In this comparison section, the most installed technologies for DHW and SH purposes in households in the EU, mentioned in previous sections, will be compared between themselves and to various emergent technologies, like the heat pumps and micro-CHP systems.

Firstly, the technologies will be compared directly accordingly to the following parameters: i) cost per unit of power [€/kW]; ii) temperature of the produced hot water [°C]; iii) global efficiency or COP; iv) energy sources; v) outputs.

Then, the ability of various technologies to retrofit a non-condensing boiler will be evaluated. Accordingly to (Pezzutto et al. 2019b), this was the most installed technology for DHW and SH purposes in residential buildings in the EU as of 2019. However, it is an outdated technology, which is no longer used for new buildings in certain countries like the UK (Boiler Guide UK 2021). Therefore, an assessment of the best technologies to retrofit this type of boilers must be done. The parameters evaluated for each technology are: i) dimensions; ii) complexity; iii) integration in a central heating system; iv) outputs; v) use of existing connections. These parameters are properly explained in Figure 2.10.

2.4.1. Direct technological comparison

In this comparison section the most installed technologies for DHW and SH purposes, mentioned in the previous sections, will be compared between themselves and to various emergent technologies, such as the heat pumps and the micro-CHP systems.

This comparison will be made in a table format. The parameters are the ones explained in the beginning of section 2.4. The presented data is collected from the previous sections and from new sources, properly identified in the following paragraphs.

The cost per unit of power is presented in euros per kilowatt of power [€/kW] of the respective system. For regular technologies, the cost regards the thermal power output of the system. As of micro-CHP systems, the cost concerns the electrical power output of the system. This comparison parameter is an average value of the ratio between the price of acquisition of the equipment and its respective power size. It is an average value because it results from a data gathering across the EU for each specific technology/equipment. No data was found for the stove technology. The sources of the data for this parameter are: regular technologies (Heating and Cooling Technology Assessment Report 2017); micro-CHP systems using RE, SE and FC technology (Mapping and analyses of the current and future (2020 - 2030) heating/cooling fuel deployment (fossil/renewables) 2016); micro-CHP systems using MGT technology (Murugan and Horák 2016); micro-CHP systems using ORC technology (Pereira et al. 2018).

In terms of the temperature of the hot water (T_{HW}), produced by the system analyzed, and measured in degrees Celsius [°C], it regards the maximum temperature of the water output of the system for SH and domestic purposes. In this scenario, the domestic purposes are any direct use of hot water in a tap. It is expected that this parameter is standardized because the use of excessively hot water can cause some problems. For example, the temperature of the hot water used directly in domestic applications must not exceed 50°C, to safeguard the physical integrity of its users. Not all technologies are covered by this parameter. Stoves and electric radiators, as explained in section 2.3, only provide SH and do not use water as the heat transfer medium. No data was found for the micro-CHP technologies. The data for this data parameter was gathered from (Heating and Cooling Technology Assessment Report 2017).

The average performance of each technology, determined for the equipments installed and available in the EU countries, is quantified by their global efficiency, symbolized by ε, which is a non-dimensional parameter with a scale from 0 to 1. The exception is the HP technology, whose performance is measured using the coefficient of performance (COP), as explained in section 2.3.5. It is important to notice that the efficiency of micro-CHP systems concerns their electrical and thermal outputs. Thus, micro-CHP's efficiency will be divided into electrical efficiency (ε_e) and thermal efficiency (ε_t). The data used for this parameter is retrieved from Project Hotmaps (Pezzutto et al. 2019b) for all technologies except micro-CHP systems, whose data was gathered from the review articles from (Murugan and Horák 2016) and (Pereira et al. 2018).

The energy sources are the inputs that allow the systems operation. They include fuels, heat sources and electricity. Heat sources concern the HP technology, which extracts heat from the outdoor air or from the ground, depending on the type of HP. The data for this parameter was retrieved from section 2.3 and from (Heating and Cooling Technology Assessment Report 2017).

Finally, the outputs of the system can be SH, DHW and/or electricity. This information was retrieved from section 2.3.

Table 2.13 – Technological comparison of DHW and SH solutions.

Technology	Cost per unit of power [€/kW]	T_{HW} [°C]	ε/COP	Energy sources	Outputs
Non-condensing gas boiler	38,30 – 54,86	80	0,85	Natural gas, LPG, and biogas	SH and DHW
Non-condensing oil boiler	79,37 – 85,71	80	0,85	Domestic fuel oil (like Diesel)	SH and DHW
Non-condensing wood boiler	321,43 – 562,50	80	0,85	Wood and sub products (logs, pellets, and chips)	SH and DHW
Stove	-	-	0,50	Biomass (wood and subproducts)	SH
Electric radiator	320 – 600	-	0,99	Electricity	SH
Electric water heater	40 – 60	88	0,99	Electricity	SH and DHW
Condensing gas boiler	83,33 – 108,11	82	0,99	Natural gas, LPG, and biogas	SH and DHW
Condensing oil boiler	61,90 – 71,43	82	0,99	Domestic fuel oil (like Diesel)	SH and DHW
Aerothermal Heat pump	500 – 600	55 – 65	3,65	Outdoor air and electricity	SH and DHW
Geothermal Heat Pump	233,33 – 1625	55 – 70	4,58	Ground heat and electricity	SH and DHW
Micro-CHP RE	1451,37	-	ε_e: 0,20 – 0,40 ε_t: 0,64	Gaseous fuels	SH, DHW and electricity
Micro-CHP SE	4201,33	-	ε_e: 0,10 – 0,20 ε_t: 0,75	Any suitable external heat source	SH, DHW and electricity
Micro-CHP MGT	900 – 1200	-	ε_e: 0,15 – 0,30 ε_t: 0,60	Gaseous fuels	SH, DHW and electricity
Micro-CHP ORC	3300	-	ε_e: 0,06 – 0,19 ε_t: 0,77 – 0,90	Any suitable external heat source and electricity	SH, DHW and electricity
Micro-CHP PEMFC	2766,63	-	ε_e: 0,25 – 0,40 ε_t: 0,30	Hydrocarbons	SH, DHW and electricity
Micro-CHP SOFC		-	ε_e: 0,30 – 0,70 ε_t: 0,40		SH, DHW and electricity

By analyzing Table 2.13 it is possible to withdraw some conclusions regarding the various parameters assessed for each technology.

As to the cost per unit of power, it is easily ascertained that the cheapest technologies are the boilers, except for the non-condensing wood boiler. Another technology with low investment cost is the electric water heater, referred to by several authors as the "electric boiler". The more expensive technologies are clearly the micro-CHP systems, whose price is an order of magnitude higher than most of the analyzed technologies.

However, it is important to remember that the prices per unit of power presented for the various micro-CHP technologies are a function of the electrical power output of the system, not the thermal power output. For systems with a low power-to-heat ratio, such as ORC micro-CHP systems, it is not necessary to invest in a high electrical power output to obtain the thermal power output required to meet the thermal needs of a residential dwelling. Heat pumps have a cost per unit of power located between boilers and micro-CHP systems.

Most boilers have a maximum heating temperature of hot water slightly above 80°C, which is more than enough to provide for the DHW and SH needs of a regular residential dwelling. As for the heat pumps, they produce hot water at a maximum temperature inferior to boilers, within the range of 55°C to 70°C. This type of technology is more efficient when the temperature of the heat sink is lower. In this case, it is considered that the water is the heat sink. Therefore, this range of temperatures is acceptable for DHW and SH applications, especially if the central heating system in which the heat pumps are usually integrated uses low temperature heat emitters, like radiant floor. The electric water heater technology presents a similar value to boilers. Finally, no data was found for micro-CHP systems. However, the maximum temperature of the hot water produced by this type of systems should be like the one of the condensing boilers.

The most efficient systems are the heat pumps, regardless of their heat source. They are followed in the most efficient systems by the condensing boilers and the electrical systems. Traditional boilers have a great thermal efficiency, but lower than the previous technologies. Finally, the stoves have a poor thermal efficiency, discouraging its use. The efficiency of the micro-CHP systems must be analyzed separately because it is composed of two separate parameters: electrical efficiency and thermal efficiency. In terms of electrical efficiency, the best micro-CHP technology is the solid oxide fuel cell (SOFC). As to the thermal efficiency, the best technology is the organic Rankine cycle (ORC). The most balanced technology in terms of these two efficiencies, and considering the specificities of residential buildings, is the reciprocating engine.

In terms of energy sources, the most flexible systems are the traditional and condensing boilers. These systems can use solid, liquid, or gaseous fuels. The most eco-friendly systems are the ones that use biomass or renewable energy sources. The systems that use biomass are the stoves and the non-condensing wood boiler. The systems that use a combination of electricity and renewable energy sources are the heat pumps, whose renewable energy sources are the outdoor air or the heat from the ground, depending on the type of heat pump used. Other type of systems that can use a renewable energy source as their heat source are the micro-CHP systems based on the SE and ORC technologies. However, most systems still use fossil fuels as their energy sources. This includes most of the micro-CHP technologies.

Overall, most of the analyzed technologies assure the DHW and SH needs of typical residential buildings. The exceptions are the electric radiators and the stoves, which only provide SH for one room. It is in this parameter that micro-CHP systems have the upper hand, due to the combined production of heat for DHW and SH purposes and electricity to cover part of the electrical needs of a residential dwelling.

Taking stock of all that has been said in this section, boilers remain the best technology to provide DHW and SH to a residential building. This is justified by their low acquisition cost, great thermal efficiency, and fuel flexibility. However, some technologies can threat the boiler's domain in the next decades. These technologies are the heat pumps and the micro-CHP systems. Heat pumps because they are the most efficient technology of the group, and they only require an electrical connection. This means that their non-dependence of fossil fuels makes this technology future proof in the upcoming decades, during the climate transition policies. Micro-CHP systems because they do everything that boilers do and produce electricity, and thus have the potential to provide significant savings for the consumer. Some of these micro-CHP technologies, like the ORC and the SE can be integrated in a renewable energy system, like solar thermal collectors, assuring a clean and sustainable operation.

2.4.2. Assessment of the retrofitting of a non-condensing boiler

In this section it will be evaluated the ability of different technologies, mentioned in previous sections, to retrofit a non-condensing boiler.

This is an important analysis, since it is expected that traditional heating technologies used in the residential scenario will be replaced by more efficient and cleaner ones, like HP's, micro-CHP systems and renewable energy systems (not covered in this analysis) (Chen et al. 2021).

To access the retrofit ability of the various technologies analyzed in section 2.3, five parameters will be analyzed adopting a binary system. This means that the score in each parameter is either one or zero. Example: if the analyzed system can be integrated in a central heating system, similarly to a boiler, then a score of 1 is attributed. If it cannot be integrated, the score is 0.

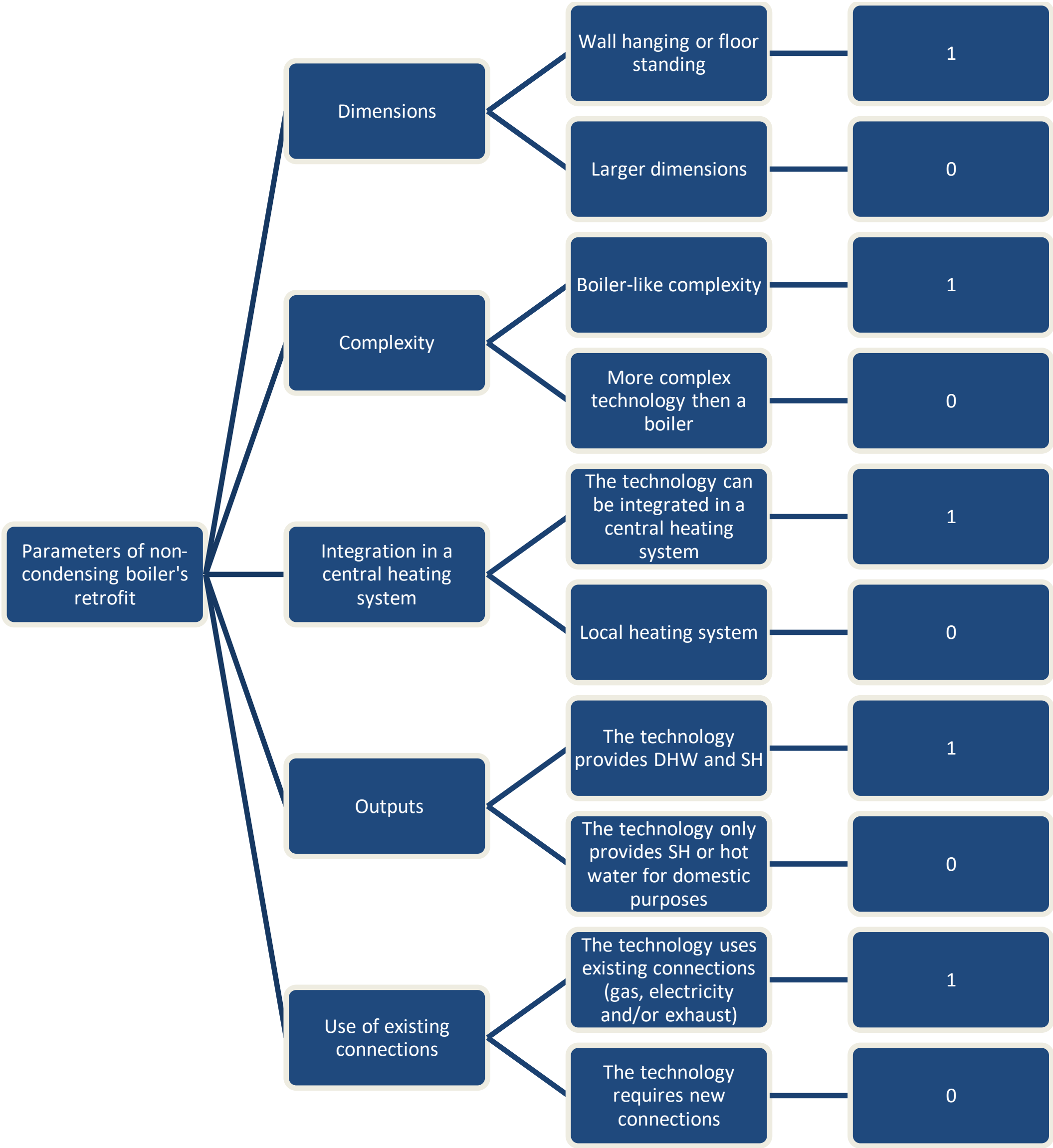

Figure 2.10. Parameters for assessing the ability of a technology to retrofit a non-condensing boiler.

Once the parameters are clarified in Figure 2.10, the assessment of the retrofit ability of the multiple technologies previously mentioned can be done. This assessment will be done in a table format, as shown in Table 2.14. Then, the results of this evaluation will be analyzed and the best technologies to retrofit a non-condensing boiler will be ascertained.

In this indirect comparison between technologies, the stoves and the electric radiators will not be included, as these devices can only heat one room in a dwelling.

Table 2.14. Assessment of the ability of various technologies to retrofit a non-condensing boiler.

Technologies	Parameters of non-condensing boiler's retrofit					
	Dimensions	**Complexity**	**Integration in a central heating system**	**Outputs**	**Use of existing connections**	**Total score**
Electric water heater	0	1	1	1	1	4
Condensing boiler	1	1	1	1	1	5
Heat pump	1	0	1	1	0	3
Micro-CHP RE	0	0	1	1	1	3
Micro-CHP SE	0	0	1	1	1	3
Micro-CHP MGT	1	0	1	1	1	4
Micro-CHP ORC	1	1	1	1	1	5
Micro-CHP FC	1	0	1	1	1	4

By analyzing the outputs of Table 2.14 it can be concluded that the best technologies to directly retrofit a non-condensing boiler in residential buildings that require SH and hot water for domestic purposes are condensing boilers and ORC based micro-CHP systems.

Electric water heaters could also be a good alternative to traditional boilers (score 4/5), but for very high thermal needs it is necessary to install a tank to store the hot water, which would imply that the dimensions of this system would be larger than a traditional boiler, jeopardizing its retrofit ability. Other technologies that obtained a score of 4 out of 5 were the micro-CHP systems that use MGT and FC technologies. However, it is considered that these systems are more complex than a traditional boiler, thus not obtaining the maximum score in this analysis.

The maximum score obtained by the condensing boiler in the assessment made in Table 2.14 does not come as surprise. As previously mentioned in section 2.3.4, the condensing boiler is the technological evolution of the non-condensing one. It delivers the same outputs as a non-condensing boiler, but in a more efficient way. This efficiency increase is achieved through a slight tweak in the working principle, with the use of a second heat exchanger. The condensing boiler still is an efficient and cost-effective technology to provide DHW for SH and domestic needs.

Finally, the third technology to obtain a five out of five score is the ORC based micro-CHP system. This promising technology delivers the same outputs as a non-condensing boiler, with the additional benefit of the production of electricity to suppress part of the electrical energy needs of a household. Although it is still in development, it is expected that this type of technology will be a highly efficient way to provide DHW and SH for residential buildings of various scales.

3. ENERGETIC AND EXERGETIC ANALYSIS

The following introductory part of this section 3 results from an adaptation of the Chapter 8 of the book Thermodynamics: An Engineering Approach by Yunus Çengel and Michael Boles (Çengel and Boles 2004).

Exergy is a thermodynamic property that is commonly unknown. However, it is very important since it quantifies the useful work potential of a given system at some specified state. Exergy can also be named availability or available energy. In other words, exergy is the upper limit on the amount of work a system can deliver without violating any thermodynamic laws. This means that the actual work delivered by a device will always be smaller than the initial exergy of that device.

When applied to a thermodynamic system, this property determines the maximum useful work that can be extracted from that system. To quantify the exergy of a system, it is necessary to specify the initial and the reference or dead states. A thermodynamic system reaches the dead state when it is in thermodynamic equilibrium with the environment that surrounds it. This thermodynamic equilibrium implies that the system's temperature and pressure are equal to the ones of the surroundings. Being at the dead state means that the system has no kinetic or potential energy relative to the environment and does not react with it. Therefore, a system has zero exergy when it reaches the dead state, which means that it cannot produce any work, being dead from a thermodynamic point of view. The properties of a thermodynamic system at the dead state are denoted by the subscript zero.

In an ideal situation, the process path of a thermodynamic system towards the dead state has no irreversibility's. This ideal situation corresponds to the case in which the work produced by the system is equal to its initial exergy. However, no real thermodynamic system has a reversible operation or process path. Every irreversibility during a process in a system is a lost opportunity to do work. Therefore, when present, irreversibility's destroy the exergy of the system along the process path. The fewer the irreversibility's of a process, the bigger is the amount of work that can be obtained from its evolution towards the dead state.

The exergy of a system is a function of both the properties of the system and the conditions of the environment (dead state). This implies that exergy is a property of the system-environment set.

To evaluate the performance of a thermodynamic system, from an exergetic point of view, the second law efficiency is used. This parameter evaluates the approximation of the system real operation to an ideal reversible operation. Its value ranges from zero to one. When the second law efficiency is zero, there is a complete destruction of the initial exergy of the system during its operation. When its value is one, then there is a total use of the initial exergy, meaning that the system does not destroy any exergy during its operation, which can be classified as reversible. The second law efficiency is defined mathematically by equation (3.1).

$$\eta_{II} = \frac{Exergy\ recovered}{Exergy\ supplied} = 1 - \frac{Exergy\ destroyed}{Exergy\ supplied} \tag{3.1}$$

It is important to notice that exergy can be supplied and/or recovered in various amounts and in various forms, such as heat, work, kinetic energy, potential energy, internal energy and enthalpy. For the purposes of this dissertation, it is going to be considered that exergy is supplied and recovered in relevant amounts only in the forms of heat, work, internal energy and enthalpy. Also, the internal energy of a system only regards sensible and latent energies, which means that chemical and nuclear energies are disregarded.

According to the second law of thermodynamics, heat is a disorganized form of energy. It can be converted to work, which is an organized form of energy. However, to do this conversion between energy forms, it is necessary to use specific machines. These machines are commonly referred to as heat engines. Another condition is that the energy in form of heat must be at a higher temperature than its surroundings, otherwise it would be in thermodynamic equilibrium with them. In addition to this requirement, only a portion of the initial energy in form of heat can be converted to energy in form of work. The remaining energy, in form of heat, is rejected to the surroundings of the heat engine. That said, every heat transfer involves a transfer of exergy.

A mass flow is a mechanism to transport energy, entropy and exergy into or out of a system. In the case of the present energetic and exergetic analyses, there are various mass flows that are present in the analyzed configurations. The combustions gases mass flow transports energy, entropy and exergy into the system. The thermal oil (configuration C) and the organic fluids mass flows circulate in the intermediate circuits of the systems, transporting energy, entropy and exergy within the system. Finally, the water mass flow extracts energy, entropy and exergy from the system. The rates of energy, entropy and exergy that are supplied or extracted from a system are proportional to the mass flow rate of the fluid that supplies or extracts them.

Any system and its surroundings can be encompassed within a control volume large enough so that its whole can be considered an isolated system. An isolated system is a system in which there is no transfer of heat, mass or work across its boundaries.

The decrease of exergy principle states that the exergy of an isolated system, during a process, always decreases or remains constant. The exergetic content of an isolated system only remains constant when the process is reversible. During an actual process in an isolated system, exergy never increases, being destroyed during the mentioned process. In this case of an isolated system, the decrease of the system's exergy is equal to the exergy destroyed during the system's process.

The destruction of exergy is always associated with the generation of entropy. Entropy is generated by irreversibility's, such as friction, heat transfer through a finite temperature difference and nonquasi-equilibrium compression and expansion. Since exergy destruction is due to the presence of irreversibility's during the process path of a system, exergy destruction is proportional to entropy generation. The more irreversible a thermodynamic process is, the larger is the amount of exergy that it destroys.

Complementing what has already been said regarding the decrease of exergy principle, it merely states that during a process the exergy destroyed in a system cannot be a negative value. This means that the exergy of a system during a process can either increase or decrease. An example of an increase in exergy is during the operation of a condensing boiler. In this system, energy is transferred from the combustion gases to the water. The exergy of the combustion gases decreases and the exergy of the water increases. However, the exergetic gain of the water is not equal to the exergetic loss of the gases, the latter is bigger than the former. Therefore, the exergy destroyed in the condensing boiler during this heat transfer process is the difference between the exergy of these two fluids.

After all this information, it is necessary to understand why an exergetic analysis is important when analyzing thermodynamic systems that produce hot water, such as boilers and micro-CHP systems. In these systems, a thermal input is converted into a thermal output. This implies that both the input and output of this type of systems is energy in the form of heat, which is a disorganized form of energy. According to the first law of thermodynamics, the energy within the control volume encompassing a system and its immediate surroundings remains constant, regardless of the type of process that the system undergoes. However, given the second law of thermodynamics, during a real process the energy quality decreases. This decrease in the quality of the energy translates to an increase in the entropy and a decrease in the exergy of the set constituted by the system and its immediate surroundings. Going back to the example of the condensing boiler, 98% of the energy in form of heat that is supplied to the system through the combustion gases is transferred to the water as energy in form of heat. However, since the final temperature of the water is lower than the initial temperature of the combustion gases, the quality of the energy in form of heat present in the water is lower than the quality of the energy in form of heat present in the combustion gases. This means that the water leaving the system (output) has a lower potential to do work than the combustion gases entering the system (input).

It is important to analyze domestic water production systems exergetically because it is necessary to quantify how much work potential is wasted during their normal operation, to improve the way that energy is used and/or to reduce the energy consumption. One way to improve the energy usage of this type of systems is to combine the production of heat and electricity using a micro-CHP system. As it will be demonstrated in section 3.4.3, micro-CHP systems provide higher energy savings than the traditional equivalent systems performing under the same conditions.

3.1. Configurations to model and analyze

In this section four different configurations of heating solutions to produce hot water for SH and domestic use will be analyzed. Three of these configurations use ORC based micro-CHP technologies for the purpose while the other is a standard condensing boiler working in conjunction with a heat engine.

The condensing boiler and heat engine configuration is considered to have a fair and complete comparison between all the configurations analyzed. The comparison will be more complete this way, since all configurations have a thermal input in the form of combustion gases and two outputs, a thermal one in the form of hot water and an electrical one in the form of electricity, produced directly by the analyzed systems. The heat engine, in this case, will simulate the conditions at which the electricity is produced in the national electrical production systems. For this comparison all the systems will be set to produce the same amount of thermal energy, being the differences between them evaluated for the electrical output and for the input of energy/exergy.

3.1.1. Configuration A: condensing boiler & heat engine

As previously explained in section 2, the boiler technology is the most used for DHW and SH purposes in the EU. Although the non-condensing boiler is the most used type of boiler, it is an old technology with no room to evolve. Therefore, the more efficient and similar alternative, the condensing boiler, was analyzed, in conjunction with an electricity production device (to simulate the national electric production system). For this purpose, it was considered a heat engine powered by combustion gases in the same conditions as those that feed the condensing boiler (resulting from the burning of natural gas).

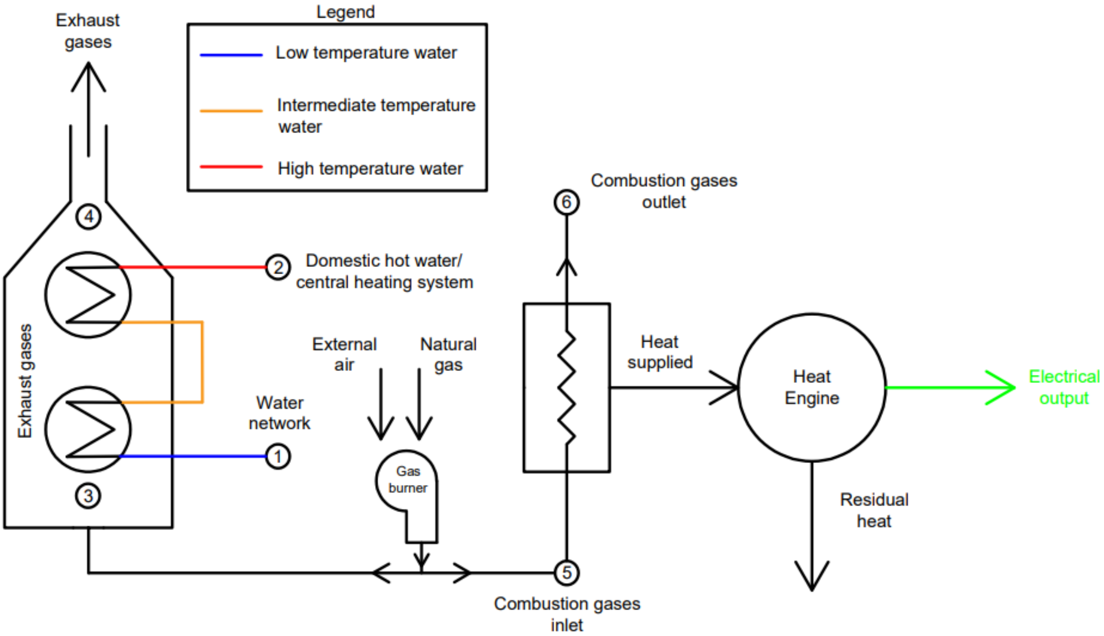

Figure 3.1. Basic schematics of configuration A: gas condensing boiler and heat engine.

The devices that are going to be modeled trough MATLAB are a gas condensing boiler and a heat engine, represented in Figure 3.1. For this analysis, six points are considered. Two points for the water circuit (1 and 2) and four points for the combustion/exhaust gases (3, 4, 5 and 6).

3.1.2. Configuration B: micro-CHP system based on the standard ORC technology

This standard configuration of the ORC technology has been explained with detail in section 2.3.6.4. It can be seen in Figure 3.2.

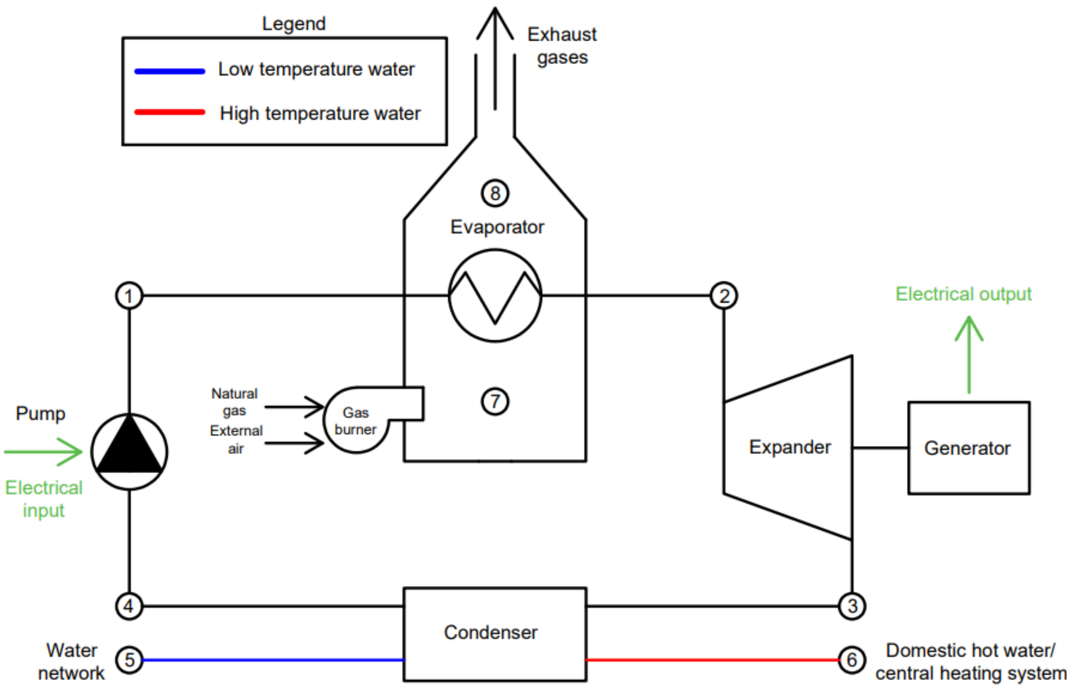

Figure 3.2. Basic schematics of configuration B: micro-CHP system based on the standard ORC technology.

To organize congruently its modeling in MATLAB, the standard ORC based micro-CHP system is divided into three sub-systems.

The first sub-system is the power circuit, in which circulates the organic fluid (r245fa). This sub-system has four notable points: 1.) pump outlet; 2.) expander inlet; 3.) expander outlet; 4.) pump inlet. The power circuit acts as an intermediary in the heat transfer from the combustion gases to the water, with the added advantage of the production of electricity in the expander-generator set.

The water sub-system is integrated in the condenser of the power circuit. It only has two points: 5.) water at the condenser inlet; 6.) water at the condenser outlet for domestic purposes or for a central heating system.

Finally, the combustion gases sub-system is integrated in the evaporator of the power circuit (direct vaporization of the organic fluid). Like the previous system, it also has two points: 7.) high temperature combustion gases at the evaporator inlet, resulting from the combustion of natural gas in the burner; 8.) low temperature combustion gases at the evaporator outlet, which are extracted from the system through the exhaust system.

3.1.3. Configuration C: micro-CHP system based on the ORC technology with intermediate vaporization of the organic fluid

The third configuration that was analyzed is very similar to the standard ORC micro-CHP system. The main difference is that the organic fluid of the power circuit is vaporized indirectly, resorting to an intermediate thermal oil circuit. This more complex configuration is represented in Figure 3.3.

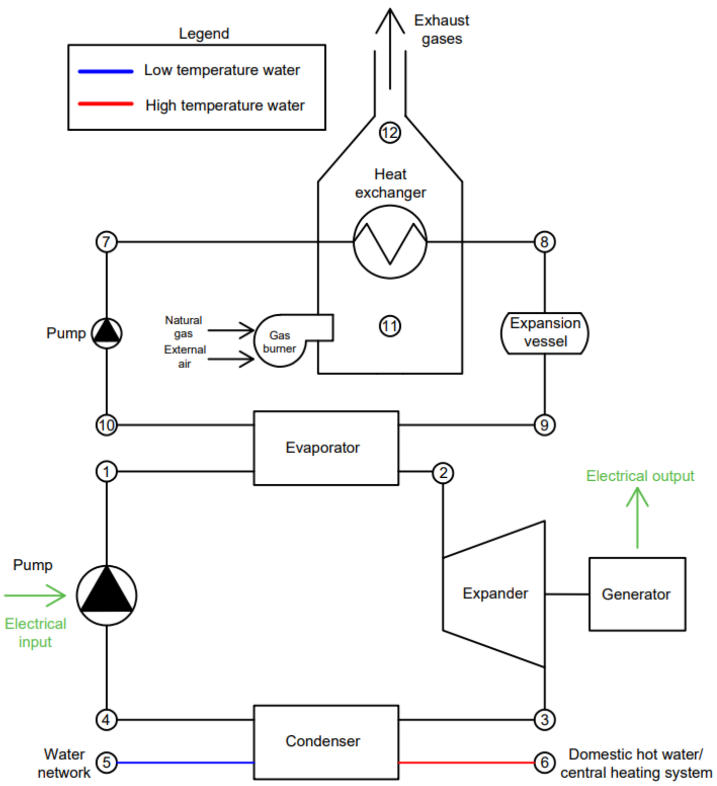

Figure 3.3. Basic schematics of configuration C: micro-CHP system based on the ORC technology with intermediate vaporization of the organic fluid.

To simplify the modeling of this configuration in MATLAB, this system was divided into four separate sub-systems.

The first sub-system is the power circuit, in which circulates the organic fluid (r245fa). This system has the following notable points: 1.) pump outlet; 2.) expander inlet; 3.) expander outlet; 4.) pump inlet. The power circuit acts as one of the intermediaries in the heat transfer from the combustion gases to the water, with the added advantage of the production of electricity in the expander-generator set.

The water sub-system is integrated in the condenser of the power circuit. It only has two notable points: 5.) water at the condenser inlet; 6.) water at the condenser outlet for domestic purposes or for a central heating system.

As for the thermal oil sub-system/circuit, it also operates as an intermediary in the heat transfer from the combustion gases to the water. This additional system has four points: 7.) heat exchanger inlet; 8.) heat exchanger outlet; 9.) power circuit evaporator inlet; 10.) power circuit evaporator outlet. The pump of this system is only used for circulation purposes, as well as the expansion vessel. This means that the thermal oil (Therminol 66) is always in liquid phase and at constant pressure.

Finally, the combustion gases sub-system is integrated in the heat exchanger of the thermal oil circuit (indirect vaporization of the organic fluid). Like the water system, it also has two points: 11.) high temperature combustion gases at the heat exchanger inlet, resulting from the combustion of natural gas in the burner; 12.) low temperature combustion gases at the heat exchanger outlet, which are extracted from the system through the exhaust system.

3.1.4. Configuration D: micro-CHP system based on the hybrid ORC technology

Configuration D corresponds to the hybrid variant of the ORC based micro-CHP technology. The differences to configuration B (standard ORC based micro-CHP system) are the two-step heating of the water and the two-step cooling of the combustion gases. The first phase of the water heating process occurs in the condenser of the power circuit, and the second in the first heat exchanger of the combustion gases chamber. As for the cooling process of the combustion gases, it occurs in the combustion gases chamber, first in the heat exchanger where the combustion gases reject heat to the pre-heated water. Then, the combustion gases are cooled further in the evaporator of the power circuit. This more complex configuration of the ORC technology is schematized in Figure 3.4.

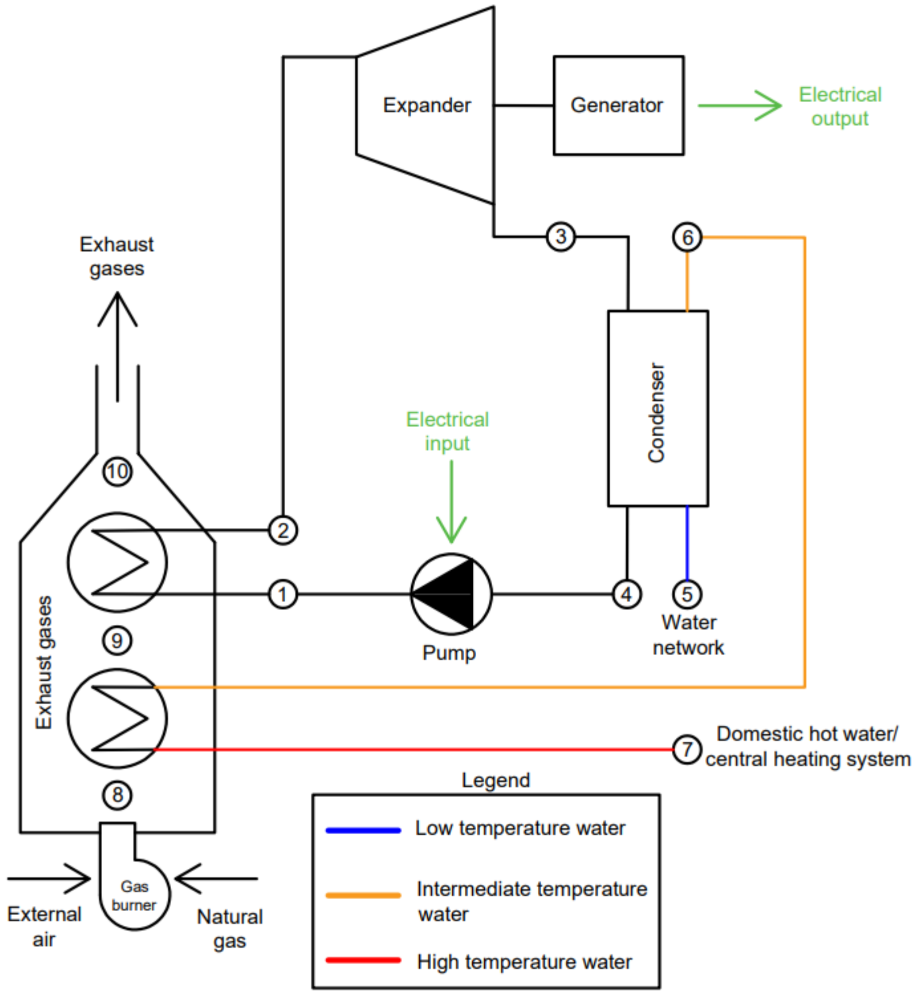

Figure 3.4. Basic schematics of configuration D: micro-CHP system based on the hybrid ORC technology.

The modeling of this hybrid ORC configuration in MATLAB was achieved by dividing the system into three sub-systems.

The first sub-system is the power circuit, in which circulates the organic fluid (r245fa). This sub-system can be characterized through the knowledge of the working fluid thermodynamic properties in four notable points: 1.) pump outlet; 2.) expander inlet; 3.) expander outlet; 4.) pump inlet. The power circuit acts as an intermediary in the heat transfer from the combustion gases to the water, with the added advantage of the production of electricity in the expander-generator set.

The water sub-system is integrated in the condenser of the power circuit and in the first heat exchanger of the combustion gases chamber. It has three points: 5.) water at the condenser inlet; 6.) intermediate temperature water; 7.) water at the heat exchanger outlet for domestic purposes or for a central heating system.

Finally, the combustion gases sub-system is integrated in the heat exchanger that finishes the water heating process and in the evaporator of the power circuit (direct vaporization of the organic fluid). Like the previous system, it also has three points: 8.) high temperature combustion gases at the heat exchanger inlet, resulting from the combustion of natural gas in the burner; 9.) intermediate temperature combustion gases; 10.) low temperature combustion gases at the evaporator outlet, which are extracted from the system through the exhaust system.

3.2. Mathematical model: general balance equations

Although four different configurations for domestic water heating equipments are analyzed in this third section, there are a few equations that are transversal to all these configurations. These equations are going to be designated as general balance equations and will be presented for configuration B. Additional equations will be presented later and the necessary adaptations will be made to the equations already presented, in order to mathematically characterize the remaining configurations (section 3.3.2).

To simplify the modeling via MATLAB of the four configurations previously presented, and to obtain valid and concise results, some assumptions and simplifications are going to be considered to define the general balance equations: i) the condenser of the micro-CHP systems (configurations B, C and D) totally condenses the working fluid, which means that only saturated liquid enters the pump; ii) steady state operation on all the analyzed configurations; iii) kinetic and gravitational potential energy variations are not considered for the calculations; iv) the pressure losses on the connection pipes, the evaporator, the condenser and other heat exchangers are disregarded.

3.2.1. Properties of the working fluids

Before listing the various general balance equations used, it is important to note that the properties of the various working fluids used (r245fa, water, and combustion gases resulting from the burning of natural gas) were taken from the REFPROP database, whenever were known two properties available to use as input and specifying the fluid.

To calculate the properties of the organic fluid in the power circuit (ORC configurations), the condensation temperature of the organic fluid in the condenser ($T_{sat,cond}$) is imposed. This can be done by defining a difference of temperatures between the heat exchanging fluids in the condenser. This means that the condensation temperature of the organic fluid in the condenser is defined through the outlet temperature of the water in the condenser ($T_{w,out}$) and the mentioned temperature difference in the condenser(ΔT_{cond}), as it can be seen in the following equation (3.2).

$$\Delta T_{cond} = T_{sat,cond} - T_{w,out}$$
$$\Leftrightarrow T_{sat,cond} = T_{w,out} + \Delta T_{cond} \ [°C] \quad (3.2)$$

The second relevant equation is the pressure ratio (r_p). It is defined as the ratio between the pressures of the working fluid in the evaporator and in the condenser of the power circuit. This output parameter is calculated trough equation (3.3).

$$r_p = \frac{p_{evap}}{p_{cond}} \quad (3.3)$$

Another property of the organic fluid that needs to be calculated through an additional equation is the temperature of the organic fluid at the evaporator outlet ($T_{evap,out}$). It is defined as a function of the superheating degree of the organic fluid (ΔT_{SH}) and the saturation temperature of the organic fluid in the evaporator ($T_{sat,evap}$).This last temperature is obtained through the REFPROP database, using the value of the pressure of the working fluid in the evaporator (p_{evap}) and assuming a totally vaporized fluid, which corresponds to a quality of one. This temperature is calculated through equation (3.4).

$$\Delta T_{SH} = T_{evap,out} - T_{sat,evap}$$
$$\Leftrightarrow T_{evap,out} = T_{sat,evap} + \Delta T_{SH} \ [°C] \quad (3.4)$$

Through the definition of the isentropic efficiency of the pump (η_P), the real value of the specific enthalpy at the pump outlet ($h_{out,r}$) is obtained. This efficiency is defined as the ratio between the isentropic variation of the specific enthalpy of the organic fluid in the pump and its real variation of specific enthalpy. This property of the organic fluid is calculated through equation (3.5).

$$\eta_P = \frac{h_{out,s} - h_{in}}{h_{out,r} - h_{in}}$$
$$\Leftrightarrow \; h_{out,r} = h_{in} + \frac{\left(h_{out,s} - h_{in}\right)}{\eta_P} \left[\frac{kJ}{kg}\right] \tag{3.5}$$

Like in the pump, the isentropic efficiency of the expander (η_{exp}) can be used to obtain the real value of the specific enthalpy at the expander outlet. It is defined as the ratio between the real variation of the specific enthalpy of the organic fluid in the expander and its isentropic variation of specific enthalpy. This property of the organic fluid is calculated trough equation (3.6).

$$\eta_{Exp} = \frac{h_{in} - h_{out,r}}{h_{in} - h_{out,s}}$$
$$\Leftrightarrow \; h_{out,r} = h_{in} - \eta_{Exp} \times \left(h_{in} - h_{out,s}\right) \left[\frac{kJ}{kg}\right] \tag{3.6}$$

As in the condenser, a temperature difference between the heat-exchanging fluids must be defined for the evaporator of the power circuit. In this case the fluids are the organic fluid and the combustion gases (applicable to configurations B and D). From this temperature difference (ΔT_{evap}) and the temperature of the organic fluid at the evaporator outlet ($T_{evap,out}$) it is possible to calculate the temperature of the combustion gases at the outlet of the analyzed equipment ($T_{CG,out}$), using for this purpose the equation (3.7).

$$\Delta T_{evap} = T_{CG,out} - T_{evap,out}$$
$$\Leftrightarrow T_{CG,out} = T_{evap,out} + \Delta T_{evap} \; [°C] \tag{3.7}$$

To proceed with the calculations, it is necessary to calculate the specific enthalpies and entropies of the combustion gases. These properties are calculated as a function of the mass fractions of each of the combustion gases constituents and their specific enthalpies/entropies. The constituents of the combustion gases are CO_2, H_2O, N_2 and O_2. These properties of the combustion gases are calculated using the general equations (3.8) and (3.9).

$$h_{CG} = \sum_{i=1}^{k} x_i \times h_i(T,p) \ \left[\frac{kJ}{kg}\right] \tag{3.8}$$

$$s_{CG} = \sum_{i=1}^{k} x_i \times s_i(T,p) \ \left[\frac{kJ}{(kg \cdot K)}\right] \tag{3.9}$$

3.2.2. Final outputs

The next equations regard the calculation of the outputs. The following two equations (equations (3.10) and (3.11)) are used to calculate the thermo-electric efficiency ($\eta_{th,E}$) and the thermo-heat efficiency ($\eta_{th,H}$) of the power circuit in the ORC configurations. They are both a function of the specific enthalpies of the organic fluid at certain notable points in the power circuit.

$$\eta_{th,E} = \frac{w_{out} - w_{in}}{q_{in}} = \frac{\left(h_{exp,in} - h_{exp,out}\right) - \left(h_{P,out} - h_{P,in}\right)}{\left(h_{evap,out} - h_{evap,in}\right)} \tag{3.10}$$

$$\eta_{th,H} = \frac{q_{out}}{q_{in}} = \frac{\left(h_{cond,in} - h_{cond,out}\right)}{\left(h_{evap,out} - h_{evap,in}\right)} \tag{3.11}$$

To calculate the mass flow rates of organic fluid ($\dot{m}_{WF}$) and of combustion gases ($\dot{m}_{CG}$), it is necessary to perform energy balances for the evaporator and condenser. These energy balances are performed from the definitions of the global efficiencies of the condenser and evaporator (η_{cond} and η_{evap}, respectively). From these energy balances results, for the condenser and the evaporator, the equations (3.12) and (3.13), respectively.

$$\eta_{cond} \times \dot{Q}_{WF} = \dot{Q}_w$$
$$\Leftrightarrow \ \dot{m}_{WF} = \frac{\dot{m}_w \times \left(h_{w,out} - h_{w,in}\right)}{\eta_{cond} \times \left(h_{cond,in} - h_{cond,out}\right)} \left[\frac{kg}{s}\right] \tag{3.12}$$

$$\eta_{evap} \times \dot{Q}_{CG} = \dot{Q}_{WF}$$
$$\Leftrightarrow \ \dot{m}_{CG} = \frac{\dot{m}_{WF} \times \left(h_{evap,out} - h_{evap,in}\right)}{\eta_{evap} \times \left(h_{CG,in} - h_{CG,out}\right)} \left[\frac{kg}{s}\right] \tag{3.13}$$

Once the efficiencies for the power circuit have been calculated it is necessary to calculate the overall efficiencies of the equipment under analysis. These are a function of the thermal powers of the combustion gases ($\dot{Q}_{CG}$) and the water $\dot{Q}_w$ and of the net mechanical power output extracted from the system ($\dot{W}_{out} - \dot{W}_{in}$). These powers are calculated by the equations (3.14) to (3.17).

$$\dot{Q}_{CG} = \dot{m}_{CG} \times \left(h_{CG,in} - h_{CG,out}\right) \ [kW] \tag{3.14}$$

$$\dot{Q}_w = \dot{m}_w \times \left(h_{w,out} - h_{w,in}\right) \ [kW] \tag{3.15}$$

$$\dot{W}_{in} = \dot{m}_{WF} \times \left(h_{P,out} - h_{P,in}\right) \ [kW] \tag{3.16}$$

$$\dot{W}_{out} = \dot{m}_{WF} \times \left(h_{exp,in} - h_{exp,out}\right) \ [kW] \tag{3.17}$$

Considering the powers calculated by the previous equations, the thermo-electric efficiency ($\eta_{CHP,E}$) and the thermo-heat efficiency ($\eta_{CHP,H}$) of the system are formulated by the following equations (3.18) and (3.19).

$$\eta_{CHP,E} = \frac{\dot{W}_{out} - \dot{W}_{in}}{\dot{Q}_{CG}} \tag{3.18}$$

$$\eta_{CHP,H} = \frac{\dot{Q}_w}{\dot{Q}_{CG}} \tag{3.19}$$

Next, the efficiencies of the system will be used to calculate the primary energy savings (PES), trough equation (3.20). This calculation is done using reference values ($\eta_{REF,E}$ and $\eta_{REF,H}$) from traditional systems that produce heat and electricity separately. These reference values are 0,445 for the referential electrical efficiency (the electrical energy is produced in a natural-gas fueled power plant built before 2012 and an aggregated correction factor that includes the climatic specificities and the grid losses for low-voltage level end-users is considered) and 0,9 for the referential thermal efficiency (the thermal energy is produced in the form of hot water from a natural-gas boiler manufactured before 2016) (Pereira et al. 2019).

$$PES = \left[1 - \left(\frac{1}{\frac{\eta_{CHP,H}}{\eta_{REF,H}} + \frac{\eta_{CHP,E}}{\eta_{REF,E}}}\right)\right] \times 100 \tag{3.20}$$

Finally, the equations that allow the exergetic analysis referred in the title of this chapter will be presented next. As discussed in the opening paragraphs of this section of the dissertation, the exergy in a system is calculated relative to a reference point, called the dead state. If a system reaches this point, it is thermodynamically in equilibrium with its surroundings, and its exergy is zero.

For any open system, the exergy balance equation is written as in the following equation (3.21).

$$\frac{d\dot{E}x_{CV}}{dt} = \dot{m} \times \int_i^f \left(1 - \frac{T}{T_0}\right) \partial q - \left[\dot{m} \times w_{out} - \frac{dV_{CV}}{dt}\right] + \dot{m} \times Ex_{in} - \dot{m} \times Ex_{out} - \dot{m} \times Ex_{destr} \; [kJ] \tag{3.21}$$

Exergy is a combined property of the state of a system and its surroundings. Considering the fluid passing through any point of any configuration as a system, the specific exergy of the fluid at that state is calculated as a function of its specific enthalpy and its specific entropy. The specific exergy at point *i* also depends on the properties of the dead state, namely its temperature [K], its specific enthalpy and its specific entropy. The generic formula for the determination of the specific exergy of a given point *i* is the equation (3.22).

$$Ex_i = (h_i - h_0) - T_0 \times (s_i - s_0) \ \left[\frac{kJ}{kg}\right] \tag{3.22}$$

The calculation of the specific exergy at the various points of a configuration is useful to evaluate the exergy destroyed in each of its components. In the case of the configurations analyzed, exergy is destroyed whenever there is a heat transfer between two different fluids in heat exchangers. Exergy destruction also occurs whenever the organic fluid is pressurized or expanded irreversibly in the power circuit, with these operations occurring in the pump and expander, respectively.

Starting with the pump and the expander, the exergy destroyed by these two components is given by the following equations (3.23) and (3.24).

$$\dot{E}x_{destr,P} = \dot{m}_{WF} \times T_0 \times \left(s_{P,out} - s_{P,in}\right) \ [kW] \tag{3.23}$$

$$\dot{E}x_{destr,Exp} = \dot{m}_{WF} \times T_0 \times \left(s_{exp,out} - s_{exp,in}\right) \ [kW] \tag{3.24}$$

As for the heat exchange between the combustion gases and the organic fluid, which occurs in the evaporator, it is necessary to calculate the specific exergy variation for each of these fluids. These specific exergy variations are calculated resorting to equations (3.25) and (3.26). Only then the exergy destroyed in this component is calculated, based on the specific exergy variations and the mass flow rates of each fluid. The exergy destroyed in the evaporator is calculated trough equation (3.27).

$$\Delta Ex_{CG} = Ex_{CG,out} - \ Ex_{CG,in} = \left(h_{CG,out} - h_{CG,in}\right) - T_0 \times \left(s_{CG,out} - s_{CG,in}\right) \ \left[\frac{kJ}{kg}\right] \tag{3.25}$$

$$\begin{gathered} \Delta Ex_{WF,evap} = Ex_{evap,out} - Ex_{evap,in} \\ \Leftrightarrow \ \Delta Ex_{WF,evap} = \left(h_{evap,out} - h_{evap,in}\right) - T_0 \times \left(s_{evap,out} - s_{evap,in}\right) \left[\frac{kJ}{kg}\right] \end{gathered} \tag{3.26}$$

$$\dot{E}x_{destr,evap} = -\left(\dot{m}_{CG} \times \Delta Ex_{CG} + \dot{m}_{WF} \times \Delta Ex_{WF,evap}\right) \ [kW] \tag{3.27}$$

The exergy destroyed in the condenser is calculated like in the evaporator case. The only difference is that the heat exchange takes place between the organic fluid and the water. The specific exergy variations of the working fluids are calculated resorting to equations (3.28) and (3.29), while the exergy destroyed in the condenser is given by equation (3.30).

$$\Delta Ex_{WF,cond} = Ex_{cond,out} - Ex_{cond,in}$$
$$\Leftrightarrow \ \Delta Ex_{WF,cond} = \left(h_{cond,out} - h_{cond,in}\right) - T_0 \times \left(s_{cond,out} - s_{cond,in}\right) \left[\frac{kJ}{kg}\right] \quad (3.28)$$

$$\Delta Ex_w = Ex_{w,out} - \ Ex_{w,in} = \left(h_{w,out} - h_{w,in}\right) - T_0 \times \left(s_{w,out} - s_{w,in}\right) \ \left[\frac{kJ}{kg}\right] \quad (3.29)$$

$$\dot{E}x_{destr,cond} = -\left(\dot{m}_{WF} \times \Delta Ex_{WF,cond} + \dot{m}_w \times \Delta Ex_w\right) \ [kW] \quad (3.30)$$

The total exergy destroyed in a system can be considered as the sum of the exergy destroyed in each of its components. This form of calculating the total exergy destroyed is going to be referred with the sub-index I. For this mathematical model only four components were considered. However, two of the analyzed configurations (C and D) include an additional heat exchanger. The exergy destroyed in this additional component must also be evaluated and accounted for. This is done in a similar way to the evaporator and condenser. The total exergy destroyed for configuration B (standard ORC micro-CHP) is calculated trough equation (3.31).

$$\dot{E}x_{destr,total(I)} = \dot{E}x_{destr,P} + \dot{E}x_{destr,evap} + \dot{E}x_{destr,Exp} + \dot{E}x_{destr,cond} \ [kW] \quad (3.31)$$

Alternatively, it can also be considered that the exergy destroyed in a system is the difference between its initial exergetic content and the net useful work that is extracted from the system. This form of calculating the total exergy destroyed is going to be referred with the sub-index II. For the configurations analyzed the initial exergy corresponds to the exergy of the combustion gases at the inlet of the system. The net useful work is the difference between the electrical power produced in the expander-generator set and the electrical power consumed by the pump. This parameter is calculated trough equation (3.32).

$$\dot{E}x_{destr,total(II)} = \dot{E}x_{CG,in} - \left(\dot{W}_{out} - \dot{W}_{in}\right)\ [kW] \tag{3.32}$$

The last parameter that was calculated was the second law efficiency (η_{II}). This efficiency is calculated by equation (3.1), present in the first paragraphs of section 3. For the calculation of this efficiency, it was considered that the total exergy destroyed in a system corresponds to the sum of the exergy destroyed in each of its components ($\dot{E}x_{destr,total(I)}$). However, the initial exergy of the system can be considered as the difference between the exergy content of the combustion gases at the inlet and the outlet of the system (form A). Another interpretation is that the initial exergy of the system is only the exergy of the combustion gases at the inlet of the system (form B). Both alternatives were implemented in MATLAB and their results are presented and analyzed in section 3.4.1. These two ways of calculating the second law efficiency are given by the following equations (3.33) and (3.34).

$$\eta_{II,A} = 1 - \frac{\dot{E}x_{destr,total(I)}}{\dot{E}x_{CG,in} - \dot{E}x_{CG,out}} \tag{3.33}$$

$$\eta_{II,B} = 1 - \frac{\dot{E}x_{destr,total(I)}}{\dot{E}x_{CG,in}} \tag{3.34}$$

3.3. Operating conditions and specificities of each configuration

In this section the operating conditions for each configuration are going to be presented and explained. These operating conditions correspond to the inputs used in the MATLAB calculations done for each configuration. The specificities of each configuration not only include certain inputs but also some equations to account for the important details and variations of each configuration.

The first configuration to be modeled in MATLAB was the standard micro-CHP ORC system (configuration B). The remaining three were modeled by changing the code of this first configuration. This means that both the equations presented in section 3.2, and the general inputs that will be presented next, correspond to configuration B. For the remaining configurations, their specific inputs will be presented separately, as well as the equations that allow the modeling of their specificities.

3.3.1. Operating conditions

The general operating conditions will be presented in three separate tables. This will be done so that each table of inputs corresponds to a sub-system in the analyzed configurations. The three sub-systems are the power circuit, the water sub-system and the combustion gases sub-system.

3.3.1.1. Power circuit

The power circuit, which is present in all the micro-CHP configurations (B, C and D), uses r245fa as its working fluid, also designated as organic fluid. The operating conditions or inputs of this system were taken from the article "Analysis of a hybrid (topping/bottoming) ORC based CHP configuration integrating a new evaporator design concept for residential applications" (Pereira et al. 2019).

As for the efficiencies of the four components that make up the power circuit, the following options were considered: i) vane pump; ii) scroll-type volumetric expander; iii) the evaporator and the condenser are plate heat exchangers operating in countercurrent (Pereira et al. 2019).

Table 3.1. General operating conditions for the power circuit of the analyzed configurations.

Symbol	Description	Value	Units
ΔT_{SH}	Superheating degree of the WF	10	°C
p_{evap}	Pressure of the WF in the evaporator	1200	kPa
η_P	Isentropic efficiency of the pump	50%	-
η_{exp}	Isentropic efficiency of the expander	75%	-
η_{evap}	Global efficiency of the evaporator	98%	-
η_{cond}	Global efficiency of the condenser	98%	-
$\eta_{EM,ger}$	Electromechanical efficiency of the generator	95%	-

3.3.1.2. Water sub-system

As for the water sub-system, the operating conditions or inputs of this sub-system where also taken from the article “Analysis of a hybrid (topping/bottoming) ORC based CHP configuration integrating a new evaporator design concept for residential applications” (Pereira et al. 2019).This system is present in all configurations. Initially, the inlet and outlet conditions of the water are considered fixed for all configurations. Later, a parametric analysis is done by changing the values of the inlet and outlet temperatures of the water, to evaluate its influence on the second law efficiency (form B), but only for the hybrid configuration (D). The results of the parametric analysis can be seen in section 3.4.4.4.

Table 3.2. General operating conditions for the water sub-system of the analyzed configurations.

Symbol	Description	Value	Units
$T_{w,in}$	Inlet temperature of the water	10	°C
$T_{w,out}$	Outlet temperature of the water	65	°C
ΔT_{cond}	Temperature difference between the fluids in the condenser	5	°C
p_w	Pressure of the water	300	kPa
$\dot{m}_w$	Mass flow rate of water	0,1	kg/s

3.3.1.3. Combustion gases sub-system

Finally, the operating conditions or inputs of the combustion gases sub-system were also taken from the article “Analysis of a hybrid (topping/bottoming) ORC based CHP configuration integrating a new evaporator design concept for residential applications” (Pereira et al. 2019). These values were obtained considering a complete combustion of natural gas with 30% excess air. The combustion process of natural gas is assumed to take place at atmospheric pressure.

Table 3.3. General operating conditions for the combustion gases sub-system of the analyzed configurations.

Symbol	Description	Value	Units
T_{flame}	Flame temperature	1540	°C
ΔT_{evap}	Temperature difference between the fluids in the evaporator	5	°C
p_{CG}	Pressure of the combustion gases	101,325	kPa
pp_{CO_2}	Partial pressure of the CO_2	7,921	kPa
pp_{H_2O}	Partial pressure of the H_2O	14,630	kPa
pp_{N_2}	Partial pressure of the N_2	74,258	kPa
pp_{O_2}	Partial pressure of the O_2	4,517	kPa
x_{CO_2}	Mass fraction of the CO_2	0,123	-
x_{H_2O}	Mass fraction of the H_2O	0,093	-
x_{N_2}	Mass fraction of the N_2	0,734	-
x_{O_2}	Mass fraction of the O_2	0,051	-

3.3.2. Specificities of each configuration

Each of the following configurations has specificities that change the hot water production process. The most distinctive is configuration A (condensing boiler & heat engine), since the condensing boiler is the type of equipment that micro-CHP systems intend to replace. The remaining two configurations are variations of the standard ORC micro-CHP system. The aim of these variations is to solve current problems derived from the operation of standard ORC micro-CHP systems and to increase its thermal and electrical efficiencies.

3.3.2.1. Specificities of configuration A (condensing boiler & heat engine)

The condensing boiler (CB) and heat engine (HE) set is the simplest configuration analyzed. The heating of the water is done directly, through heat exchange between the combustion gases and the water. The water heating process is done by cooling the combustion gases in two heat exchangers incorporated inside the boiler. For simplification purposes in the modeling in MATLAB, it was considered that all the heat exchange occurs only in one heat exchanger. The production of electricity is done in the heat engine. This device is fed by a mass flow rate of combustion gases like the condensing boiler. The final products of the heat engine operation are cooled combustion gases (which contain a residual thermal power), and electrical power.

Therefore, one of the specific operating conditions of this configuration is the temperature difference between the combustion gases exiting the boiler and the water exiting the boiler (ΔT_{CB} - equation (3.35)). This temperature difference is fixed to ensure that the heat exchange occurs from the combustion gases to the water throughout the boiler. Other specific operating condition is the condensing boiler global efficiency (η_{CB}), for which a typical thermal efficiency value for a natural gas fired condensing boiler is assumed.

The other two inputs regard the heat engine. They are the electrical power that is produced in the heat engine ($\dot{Q}_{HE}$) and the efficiency of the electricity production process that occurs in the heat engine (η_{grid}). For the electric power produced by the heat engine, the value shown in Table 3.4 was assumed, since this is the approximate electrical power produced in the other configurations. As for the heat engine efficiency, a value equal to the efficiency associated with the electricity production process in the Portuguese national grid was assumed, as established in Order nº 17313/2008 issued by the Ministry of the Economy and Innovation (Despacho n.º 17313/2008 2008). They are correlated trough equation (3.37). These specific inputs and their respective values can be seen in Table 3.4.

Table 3.4. Specific operating conditions of configuration A (condensing boiler & heat engine).

Symbol	Description	Value	Units
ΔT_{CB}	Temperature difference between the fluids in the CB	5	°C
η_{CB}	Heat exchange efficiency of the CB	98%	-
$\dot{W}_{HE}$	Produced electrical power in the HE	1,2	kW
η_{grid}	Efficiency of the HE	40%	-

The following equation allows to determine the outlet temperature of the combustion gases:

$$\Delta T_{CB} = T_{CG,out} - T_{w,out}$$
$$\Leftrightarrow\ T_{CG,out} = T_{w,out} + \Delta T_{CB}\ [°C] \qquad (3.35)$$

Through an energy balance of the boiler and using the heat exchange efficiency value defined in Table 3.4 for this device it is possible to calculate the mass flow rate of combustion gases ($\dot{m}_{CG,CB}$) necessary in this equipment, as demonstrated in equation (3.36).

$$\eta_{CB} = \frac{\dot{Q}_w}{\dot{Q}_{CG,CB}}$$
$$\Leftrightarrow \ \dot{m}_{CG,CB} = \frac{\dot{m}_w \times \left(h_{w,out} - h_{w,in}\right)}{\eta_{CB} \times \left(h_{CG,in(CB)} - h_{CG,out(CB)}\right)} \left[\frac{kg}{s}\right] \tag{3.36}$$

To calculate the thermal power input that the heat engine needs to produce the electrical power output specified in Table 3.4, the following equation (3.37) is used.

$$\eta_{grid} = \frac{\dot{W}_{HE}}{\dot{Q}_{HE}}$$
$$\Leftrightarrow \ \dot{Q}_{HE} = \frac{\dot{W}_{HE}}{\eta_{grid}} \ [kW] \tag{3.37}$$

It should also be made explicit that the thermo-electric efficiency of configuration A is calculated as a function of the electrical power produced by the heat engine ($\dot{W}_{HE}$), and not the net mechanical power output resulting from the operation of the system.

As for the exergy destroyed (form I) in this configuration, the calculations boil down to six equations. This exergetic analysis is done using the equations (3.25), (3.26) and (3.27), deduced for the evaporator of configuration B (standard ORC micro-CHP), but considering, in this case of the condensing boiler, the working fluid as water instead of organic fluid. For the heat engine, the exergy destroyed is only dependent of the flow of combustion gases and of the produced mechanical power. This can be seen in the following equations (3.38) to (3.43).

$$\Delta Ex_{CG,CB} = Ex_{CG,out(CB)} - \ Ex_{CG,in(CB)}$$
$$\Leftrightarrow \Delta Ex_{CG,CB} = \left(h_{CG,out(CB)} - h_{CG,in(CB)}\right) - T_0 \times \left(s_{CG,out(CB)} - s_{CG,in(CB)}\right) \ \left[\frac{kJ}{kg}\right] \tag{3.38}$$

$$\Delta Ex_w = Ex_{w,out} - Ex_{w,in} = \left(h_{w,out} - h_{w,in}\right) - T_0 \times \left(s_{w,out} - s_{w,in}\right) \left[\frac{kJ}{kg}\right] \tag{3.39}$$

$$\dot{E}x_{destr,CB} = -\left(\dot{m}_{CG,CB} \times \Delta Ex_{CG,CB} + \dot{m}_w \times \Delta Ex_w\right) \ [kW] \tag{3.40}$$

$$\Delta Ex_{CG,HE} = Ex_{CG,out(HE)} - Ex_{CG,in(HE)}$$
$$\Leftrightarrow \Delta Ex_{CG,HE} = \left(h_{CG,out(HE)} - h_{CG,in(HE)}\right) - T_0 \times \left(s_{CG,out(HE)} - s_{CG,in(HE)}\right) \left[\frac{kJ}{kg}\right] \tag{3.41}$$

$$\dot{E}x_{destr,HE} = -\left(\dot{m}_{CG,HE} \times \Delta Ex_{CG,HE}\right) - \dot{W}_{HE} \ [kW] \tag{3.42}$$

$$\dot{E}x_{destr,total(I)} = \dot{E}x_{destr,CB} + \dot{E}x_{destr,HE} \ [kW] \tag{3.43}$$

Finally, the exergy destroyed in its form II is adapted from equation (3.32), considering the initial exergy of the combustion gases in the condensing boiler and in heat engine and the work produced by the heat engine, as presented in equation (3.44).

$$\dot{E}x_{destr,total(II)} = \dot{m}_{CG,CB} \times Ex_{CG,CB(in)} + \dot{m}_{CG,HE} \times Ex_{CG,HE(in)} - \dot{W}_{HE} \ [kW] \tag{3.44}$$

3.3.2.2. Specificities of configuration C (micro-CHP system based on the ORC technology with intermediate vaporization of the organic fluid)

The main difference between this configuration and the standard ORC micro-CHP system is that the vaporization of the organic fluid is done using an intermediate circuit. That is, the vaporization of the organic fluid is done indirectly, preventing its thermal degradation. In this intermediate circuit the working fluid is thermal oil, for which the Therminol 66 oil was chosen.

The intermediate circuit does an intermediate heat transfer between the combustion gases sub-system and the power circuit. It is composed by a circulation pump, a heat exchanger (combustion gases to thermal oil), an expansion vessel and another heat exchanger (thermal oil to organic fluid). This last heat exchanger is the evaporator of the power circuit. Despite having a pump and an expansion vessel, it is considered that the pressure variation in this intermediate circuit is null, and the pump and expansion vessel act only to ensure the circulation of the thermal oil. This means that the pressure in the thermal oil circuit (p_{TO}) is constant and as the value presented in Table 3.5.

In the following Table 3.5 are the conditions specific to configuration C that will serve as the basis for the equations (3.45) to (3.51).

Table 3.5. Specific operating conditions of configuration C (ORC micro-CHP system with intermediate vaporization of the organic fluid).

Symbol	Description	Value	Units
ΔT_{HE}	Temperature difference between the fluids in the heat exchanger	5	°C
ΔT_{TO}	Temperature variation range of the thermal oil	50	°C
p_{TO}	Pressure of the thermal oil	300	kPa
η_{HE}	Global efficiency of the heat exchanger	98%	-

To ensure that the heat exchange occurs from the combustion gases to the thermal oil in the additional heat exchanger (HE), a fixed temperature difference is set between the combustion gases outlet temperature ($T_{CG,out}$) and the thermal oil temperature at the heat exchanger outlet ($T_{TO,out}$). This temperature difference between the heat exchanging fluids is represented as ΔT_{HE}. The calculation of the combustion gases outlet temperature is done using this temperature variation in equation (3.45).

$$\Delta T_{HE} = T_{CG,out} - T_{TO,out}$$
$$\Leftrightarrow \; T_{CG,out} = T_{TO,out} + \Delta T_{HE} \text{ [°C]} \tag{3.45}$$

A heating degree is defined to calculate the heating temperature of the thermal oil. This heating degree of the thermal oil is designated as ΔT_{TO} and is defined as the difference between the thermal oil temperature at the outlet of the combustion gases heat exchanger ($T_{TO,out}$) and the thermal oil temperature ate the inlet of the same heat exchanger ($T_{TO,in}$). These parameters are correlated trough equation (3.46).

$$\Delta T_{TO} = T_{TO,out} - T_{TO,in}$$
$$\Leftrightarrow \; T_{TO,out} = T_{TO,in} + \Delta T_{TO} \text{ [°C]} \tag{3.46}$$

The combustion gases mass flow rate ($\dot{m}_{CG}$) is calculated using the efficiency of the heat exchanger (η_{HE}) that exchanges heat between the combustion gases and the thermal oil. This calculation is done using equation (3.47), which is basically an energy balance to the heat exchanger.

$$\eta_{HE} = \frac{\dot{Q}_{TO}}{\dot{Q}_{CG}}$$
$$\Leftrightarrow \dot{m}_{CG} = \frac{\dot{m}_{TO} \times (h_{TO,out} - h_{TO,in})}{\eta_{HE} \times (h_{CG,in} - h_{CG,out})} \left[\frac{kg}{s}\right] \tag{3.47}$$

Finally, the exergy destroyed (form I) in this configuration is calculated in a similar way as for configuration B (standard ORC micro-CHP), with the addition of the portion of exergy destroyed in the additional heat exchanger that is used in the intermediate circuit. This additional exergetic calculations are done using equations (3.48) to (3.51). As for the exergy destroyed in form II, it is used the same equation as configuration B (equation (3.32)).

$$\Delta Ex_{CG} = Ex_{CG,out} - Ex_{CG,in} = (h_{CG,out} - h_{CG,in}) - T_0 \times (s_{CG,out} - s_{CG,in}) \left[\frac{kJ}{kg}\right] \tag{3.48}$$

$$\Delta Ex_{TO,HE} = Ex_{TO,out} - Ex_{TO,in}$$
$$\Leftrightarrow \Delta Ex_{TO,HE} = (h_{TO,out} - h_{TO,in}) - T_0 \times (s_{TO,out} - s_{TO,in}) \left[\frac{kJ}{kg}\right] \tag{3.49}$$

$$\dot{E}x_{destr,HE} = -(\dot{m}_{CG} \times \Delta Ex_{CG} + \dot{m}_{TO} \times \Delta Ex_{TO,HE}) \; [kW] \tag{3.50}$$

$$\dot{E}x_{destr,total(I)} = \dot{E}x_{destr,P} + \dot{E}x_{destr,evap} + \dot{E}x_{destr,Exp} + \dot{E}x_{destr,cond} + \dot{E}x_{destr,HE} \; [kW] \tag{3.51}$$

3.3.2.3. Specificities of configuration D (micro-CHP system based on the hybrid ORC technology)

In this last configuration, abbreviated as hybrid, its particularity is that the water heating is done in two phases. This two-stage heating requires the use of two heat exchangers. One of these heat exchangers is the power circuit condenser and the other one is an additional heat exchanger that cools the combustion gases before they reach the power circuit evaporator.

Because of the two-stage heating of the water, the combustion gases are also cooled in two stages. This implies the calculation of an intermediate water heating temperature ($T_{w,int}$) and an intermediate combustion gases cooling temperature ($T_{CG,int}$). This temperature calculations are done using the specifics inputs presented in Table 3.6.

Table 3.6. Specific operating conditions of configuration D (hybrid ORC micro-CHP system).

Symbol	Description	Value	Units
ΔT_{CG}	Initial approach to the intermediate combustion gases cooling	150	°C
η_{HE}	Global efficiency of the water heat exchanger	98%	-
Θ	Water post-heating fraction	0,37	-

The intermediate water temperature ($T_{w,int}$) is calculated using the water post-heating fraction (Θ). This fraction represents the portion of the water heating process that is done in the additional heat exchanger. It is a function of the inlet temperature and the outlet temperature of the water ($T_{w,in}$ and $T_{w,out}$, respectively). It is defined by equation (3.52).

$$\theta = \frac{\left(T_{w,out} - T_{w,int}\right)}{\left(T_{w,out} - T_{w,in}\right)}$$
$$\Leftrightarrow T_{w,int} = T_{w,out} - \theta \times \left(T_{w,out} - T_{w,in}\right) \text{ [°C]} \tag{3.52}$$

The calculation of the intermediate combustion gases temperature ($T_{CG,int}$) is done iteratively. For this, an initial approximation for the degree of the combustion gases cooling (ΔT_{CG}) is used. This degree of cooling is a difference between the initial combustion gases temperature and its intermediate temperature (equation (3.53)). The specific enthalpy of the combustion gases is obtained from the successive values calculated for the intermediate temperature of the combustion gases. This approximate specific enthalpy is then compared to the actual specific enthalpy. The actual specific enthalpy of the combustion gases is calculated by means of three energy balances. The energy balances concern the evaporator, the condenser and the combustion gases-water heat exchanger.

$$\Delta T_{CG} = T_{CG,in} - T_{CG,int}$$
$$\Leftrightarrow T_{CG,int} = T_{CG,in} - \Delta T_{CG} \text{ [°C]} \tag{3.53}$$

Once the properties for the intermediate point of the combustion gases have been calculated, the combustion gases mass flow rate ($\dot{m}_{CG}$) can be calculated. One way to perform this calculation is through an energy balance for the combustion gases-water heat exchanger. For this the heat exchanger efficiency (η_{HE}) is used (equation (3.54)). This efficiency is defined in Table 3.6.

$$\eta_{HE} = \frac{\dot{Q}_{w,HE}}{\dot{Q}_{CG,HE}}$$
$$\Leftrightarrow \ \dot{m}_{CG} = \frac{\dot{m}_w \times \left(h_{w,out} - h_{w,int}\right)}{\eta_{HE} \times \left(h_{CG,in} - h_{CG,int}\right)} \left[\frac{kg}{s}\right] \tag{3.54}$$

Finally, the exergy destroyed (form I) in this configuration is calculated in a similar way as for configuration B (standard ORC micro-CHP), with the addition of the portion of exergy destroyed in the additional heat exchanger that is used in the post-heating process of the water. These changes are demonstrated in the following equations (3.55) to (3.58). As for the exergy destroyed in form II, it is used the same equation as configuration B (equation (3.32)).

$$\Delta Ex_{CG,int} = Ex_{CG,int} - \ Ex_{CG,in}$$
$$\Leftrightarrow \Delta Ex_{CG,int} = \left(h_{CG,int} - h_{CG,in}\right) - T_0 \times \left(s_{CG,int} - s_{CG,in}\right) \ \left[\frac{kJ}{kg}\right] \tag{3.55}$$

$$\Delta Ex_{w,HE} = Ex_{w,out} - Ex_{w,int}$$
$$\Leftrightarrow \Delta Ex_{w,HE} = \left(h_{w,out} - h_{w,int}\right) - T_0 \times \left(s_{w,out} - s_{w,int}\right) \left[\frac{kJ}{kg}\right] \tag{3.56}$$

$$\dot{E}x_{destr,HE} = -\left(\dot{m}_{CG} \times \Delta Ex_{CG,int} + \dot{m}_w \times \Delta Ex_{w,HE}\right) \ [kW] \tag{3.57}$$

$$\dot{E}x_{destr,total} = \dot{E}x_{destr,P} + \dot{E}x_{destr,evap} + \dot{E}x_{destr,Exp} + \dot{E}x_{destr,cond} + \dot{E}x_{destr,HE} \ [kW] \tag{3.58}$$

3.4. Presentation and discussion of the results

In this last section of the energetic and exergetic analysis the results are going to be presented and discussed.

It is important to remember that these results were obtained based on a fixed and equal mass flow rate of water for all configurations. This means that the calculations were made according to the needs of the final consumer of hot water for domestic needs and for space heating.

The results that are going to be presented next include: i) thermal, electric and second law efficiencies; ii) thermal and electrical powers; iii) Primary Energy Savings (PES); iv) exergy destruction.

3.4.1. Efficiencies

As seen in section 3.2.2, the main efficiencies that were calculated and analyzed are the thermo-electric efficiency ($\eta_{CHP,E}$), the thermo-heat efficiency ($\eta_{CHP,H}$) and the second law efficiency (η_{II}) of the various configurations analyzed. These parameters are represented by equations (3.18), (3.19), (3.33) and (3.34).

The second law efficiency is calculated in two different ways. The first one ($\eta_{II,A}$) considers that the exergy supplied to the process that occurs within any configuration is the exergy variation of the combustion gases, between the inlet and the outlet of the sub-system. The other one ($\eta_{II,B}$) assumes that the exergy supplied is only the exergy of the combustion gases at the inlet of the system. However, both alternatives consider that the exergy destroyed during the process is the sum of the exergy destroyed in each of the configuration components (pumps, expanders and heat exchangers), which corresponds to equation (3.31).

The values of the already mentioned efficiencies are graphically represented in Figure 3.5.

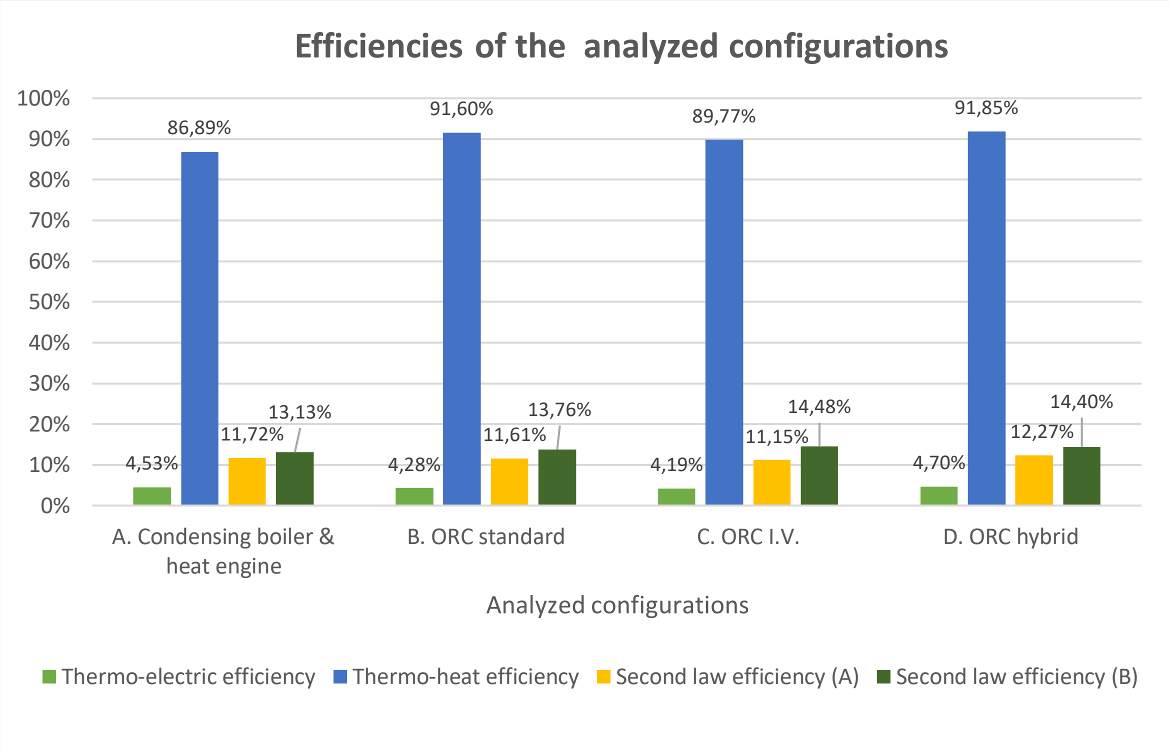

Figure 3.5. Thermo-electric, thermo-heat and second law efficiencies for the four analyzed configurations.

The conclusions presented in the following paragraphs result from the analysis of the graph of Figure 3.5.

Starting with the thermo-electric efficiency, the best configuration in this parameter is the D (hybrid ORC micro-CHP), followed by configurations A (condensing boiler & heat engine), B (standard ORC micro-CHP) and C (I.V. ORC micro-CHP). This result means that the hybrid configuration is the system that best converts a thermal power input into an electrical power output. However, the difference between the efficiencies of the best and the worst configurations is only of 0,51 percentual points.

Continuing with the thermo related efficiencies, the second one presented here is the thermo-heat efficiency. Typical values for this efficiency are an order of magnitude above the thermo-electric efficiency. Once again, the best configuration in this parameter is the D (hybrid ORC micro-CHP). The second-best configuration is the B (standard ORC micro-CHP), followed by the C (I.V. ORC micro-CHP) and the A (condensing boiler & heat engine). These results mean that all the micro-CHP configurations analyzed are better at converting a thermal power input into a thermal power output than the condensing boiler. The difference between the efficiencies of the best and the worst configurations is 4,95 percentual points, which is a relevant value, considering that the main purpose of the analyzed systems is to suppress a thermal need in the form of hot water.

Regarding the second law efficiency in its first form ($\eta_{II,A}$), once again the best configuration is the D (hybrid ORC micro-CHP), followed by configurations A (condensing boiler & heat engine), B (standard ORC micro-CHP) and C (I.V. ORC micro-CHP). This means that the system that wastes the least useful work potential, that is, the system that is exergetically more efficient is the hybrid configuration. However, the difference between the efficiencies of the best and the worst configurations is only of 1,12 percentual points.

Finally, the most efficient system according to second law efficiency in its second form ($\eta_{II,B}$) is configuration C (I.V. ORC micro-CHP), followed by configurations D (hybrid ORC micro-CHP), B (standard ORC micro-CHP) and A (condensing boiler & heat engine). These results mean that (considering that the exergy supplied is only the exergy of the combustion gases at the inlet of the system) the micro-CHP configurations are exergetically more efficient than the condensing boiler and heat engine set. However, the difference between the efficiencies of the best and the worst configurations is only of 1,35 percentual points.

3.4.2. Thermal and electrical powers

The main output resulting from the normal operation of the analyzed configurations is the thermal power output in the form of hot water. This output is equal for all configurations and has a value of 23 kW. To produce this thermal output a thermal input must be supplied to all the systems analyzed. The values of the thermal power input are graphically represented in Figure 3.6. Thermal power input means the thermal power of the combustion gases at the system inlet. Also present in this figure are the thermal powers supplied to each of the configurations analyzed. The thermal power is provided to the system in the form of combustion gases, resulting from the combustion of natural gas, as explained in section 3.3.1.3. The inlet conditions of the combustion gases are the same in all configurations, which include an inlet temperature of 1540°C. However, the outlet temperature and the mass flow rate of the combustion gases depend on the analyzed configuration. The thermal power supplied to each system is correlated with the mass flow rate of combustion gases, whose values for each configuration can be found in Figure 3.7.

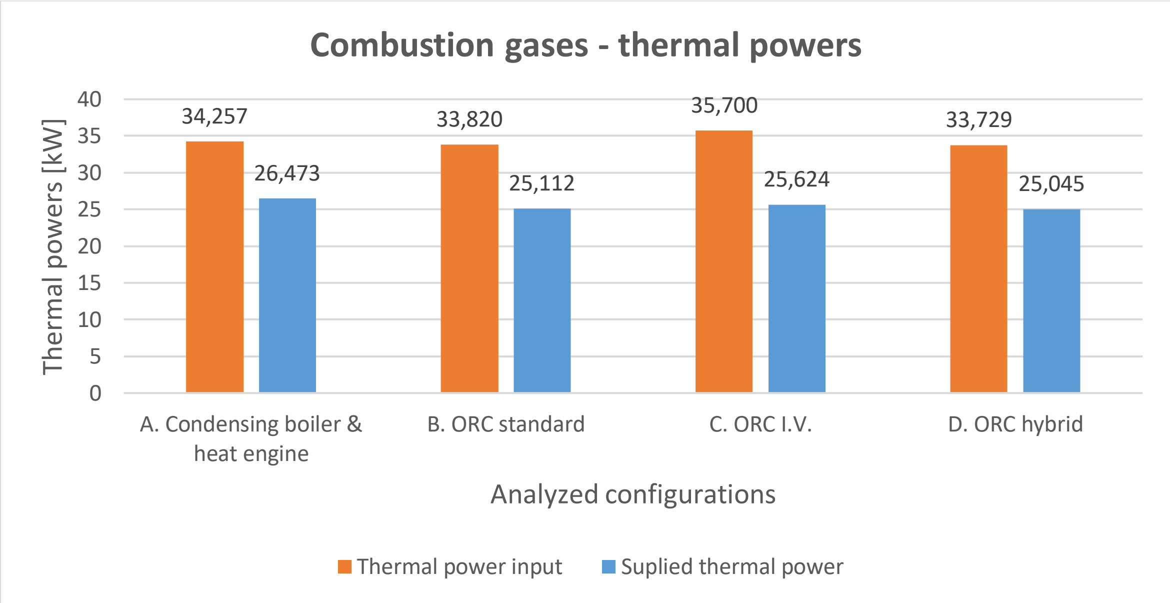

Figure 3.6. Thermal power inputs and supplied thermal powers, in the form of combustion gases, for the four analyzed configurations.

The data concerning the thermal power inputs, represented in the Figure 3.6, leads to the conclusion that the largest thermal power input in the form of hot combustion gases is provided to configuration C (I.V. ORC micro-CHP), followed by configurations A (condensing boiler & heat engine), B (standard ORC micro-CHP) and D (hybrid ORC micro-CHP). This means that it is the hybrid configuration that requires the least thermal power input to produce the same thermal output as the others.

Through the analysis of Figure 3.6 it is also possible to conclude that the configuration that requires a larger supply of thermal power is the A (condensing boiler & heat engine), followed by configurations C (I.V. ORC micro-CHP), B (standard ORC micro-CHP) and D (hybrid ORC micro-CHP). The higher the supply of thermal power required, the higher the combustion gases mass flow rate must be.

Combining the conclusions of the last two paragraphs, the configurations requiring the highest thermal power input (A and C) are also those that harness the greatest absolute amount of thermal power from the input. However, as seen in the section 3.4.1, this does not mean that these configurations are the most efficient. As seen previously, the most efficient configuration, in terms of thermo-electric, thermo-heat and second-law (form A) efficiencies is the hybrid one.

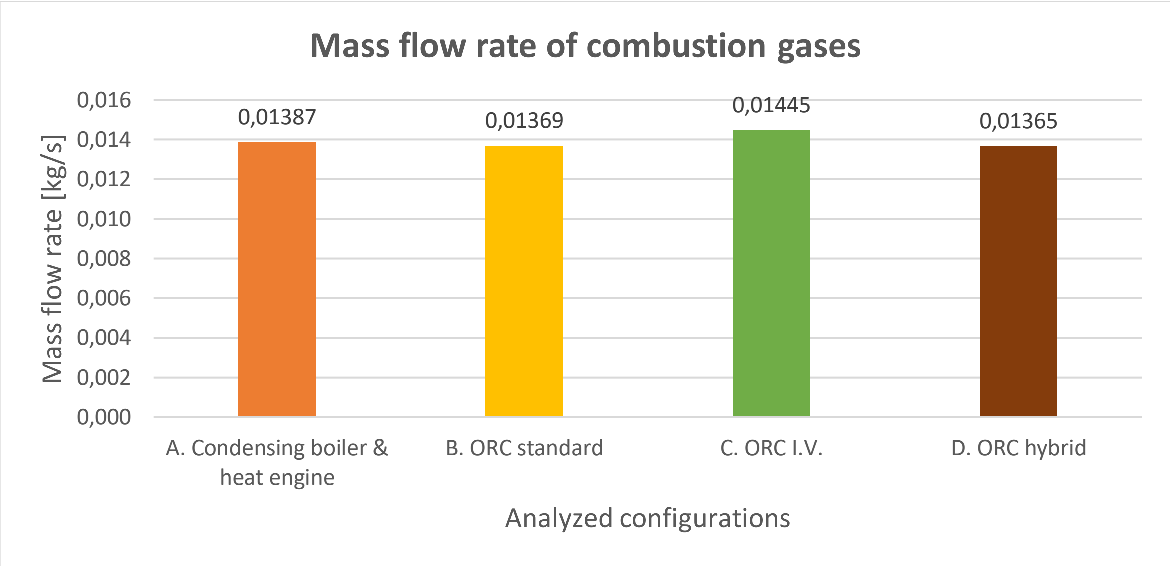

Figure 3.7. Mass flow rate of combustion gases for the four analyzed configurations.

Figure 3.7 shows the data for the mass flow rate of combustion gases in each configuration. Configuration C (I.V. ORC micro-CHP) is the one that requires the highest mass flow rate to produce the same thermal power output as the others. The remaining configurations in descending order are A (condensing boiler & heat engine), B (standard ORC micro-CHP) and D (hybrid ORC micro-CHP). From these results it can be concluded that the configuration that consumes the least combustion gases is the hybrid one.

Once again, the best results are achieved by the hybrid configuration, since it requires the least thermal power input, the lowest supply of thermal power and the smaller mass flow rate of combustion gases. This implies that this configuration is the one that consumes less natural gas to produce the same thermal power output as the other ones.

Besides the thermal power output, the analyzed configurations also produce an electrical power output. For configuration A, this output is produced in the heat engine. In the remaining configurations, since they all use an ORC implemented in a micro-CHP system, the electrical power output is produced in the expander-generator set. The values of this secondary output are graphically represented in Figure 3.8.

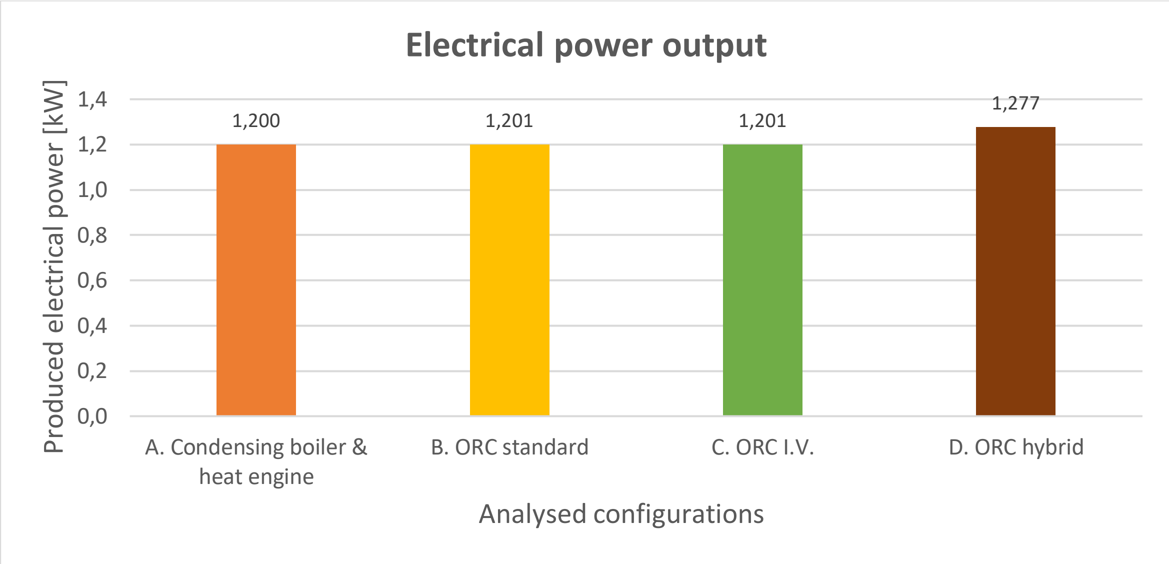

Figure 3.8. Electrical power output for the four analyzed configurations.

Through the analysis of Figure 3.8 it is possible to conclude that the configuration that produces the largest electrical power output is the D, followed by configurations B and C (same output) and A. However, the power values are very similar between all configurations. The difference between the electrical power outputs of the best and the worst configurations is only of 0,077 kW.

Concluding the analysis of the power results, it is possible to state unequivocally that the hybrid configuration is the best of the ones analyzed. It is the configuration that requires the least thermal power input, the lowest supply of thermal power and the smallest mass flow rate of combustion gases to produce the same thermal power input as the other ones. This configuration has the additional benefit of producing the largest electrical power output of the analyzed configurations.

3.4.3. Primary Energy Savings

The Primary Energy Savings (PES) parameter is a performance indicator commonly used to analyze the operation of micro-CHP systems (Pereira et al. 2019). It is calculated as seen in section 3.2.2, more specifically accordingly to equation (3.20). This parameter evaluates the primary energy savings resulting from the normal operation of a micro-CHP system versus traditional systems that produce thermal and electrical energy separately. The values of PES for each configuration are graphically represented in Figure 3.9.

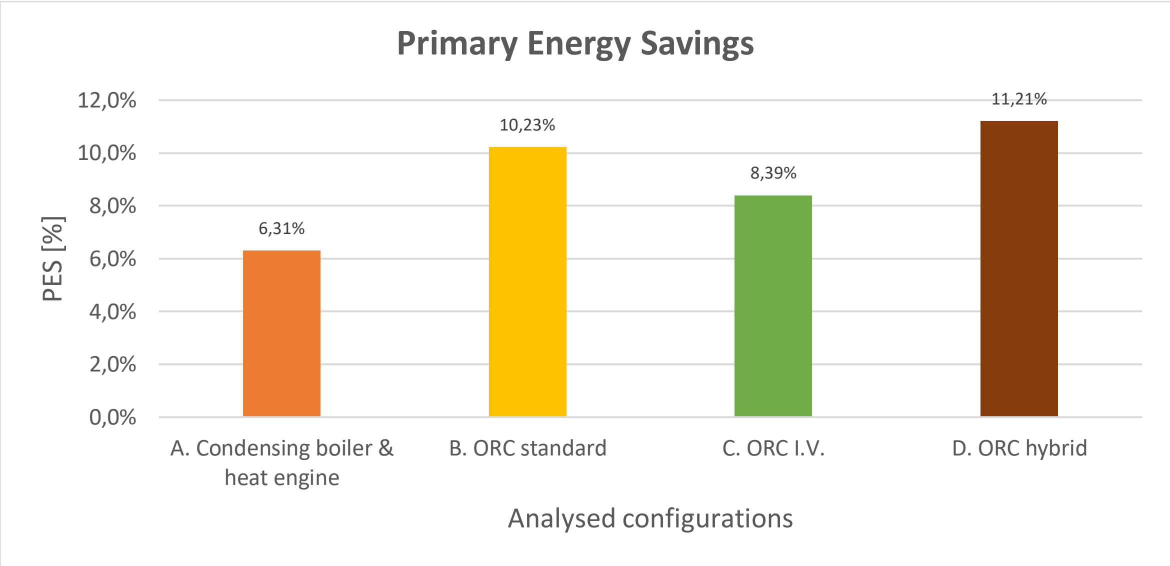

Figure 3.9. Primary Energy Savings for the four analyzed configurations.

From the analysis of Figure 3.9 it is possible to conclude that the hybrid configuration (D) has the largest PES, followed by configurations B (standard ORC micro-CHP), C (I.V. ORC micro-CHP) and A (condensing boiler & heat engine). These results mean that the hybrid configuration is the one whose operation will provide higher savings in primary energy to its user. In the analyzed configurations the primary energy would be natural gas. The difference between the PES of the best and the worst configuration is 4,90%, which is a very significant value since it translates in relevant savings in primary energy.

3.4.4. Exergy

In this last section of the results, data regarding the exergy destruction will be presented. This data results from the normal operation of each of the analyzed configurations.

Firstly, each configuration will be analyzed individually and relatively, presenting the share of total exergy that is destroyed in the components of each configuration. Then, the absolute values will be presented for the total exergy destroyed in each of the configurations. These absolute values will be calculated in two different ways, each of which calculates the exergy destroyed by making different considerations. Afterwards, Sankey diagrams for the exergy destroyed in each configuration will be presented. These diagrams intend to illustrate the exergy path from the starting point (combustion gases) to the products resulting from the normal operation of each one of the configurations (electricity, hot water and cooled combustion gases). Finally, a parametric analysis is presented for the hybrid configuration. The objective of this parametric analysis is to show the influence of the inlet and outlet temperatures of the water in the second law efficiency of configuration D.

3.4.4.1. Exergy destruction: relative values

The first configuration analyzed is the simplest. Configuration A is only comprised of a condensing boiler and a heat engine, who are fed from the same source of combustion gases, a natural gas burner. The share of exergy destroyed in each of these two components can be seen in Figure 3.10.

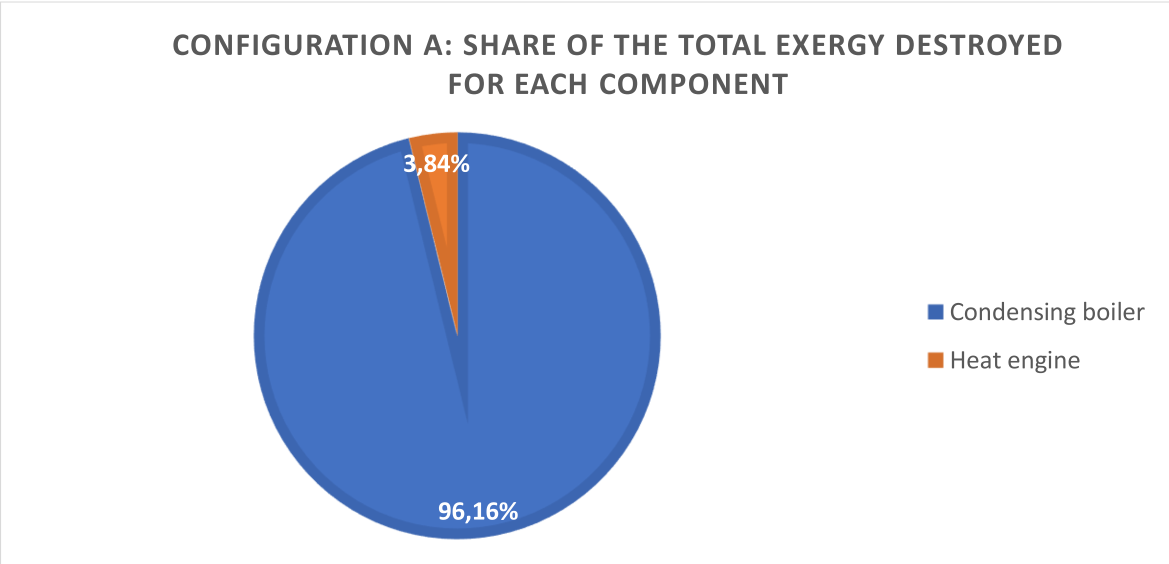

Figure 3.10. Share of the total exergy destroyed for each component of configuration A (condensing boiler & heat engine).

Through the analysis of Figure 3.10 it is easily perceptible that most of the exergy is destroyed in the condensing boiler. This is attested by the fact that the power produced in the condensing boiler (23 kW) is much higher than the power produced in the heat engine (1,2 kW).

The second configuration evaluated is the standard ORC micro-CHP (B). It consists of four components: pump, evaporator, expander and condenser. The share of the total exergy destroyed in each of these components can be seen in Figure 3.11.

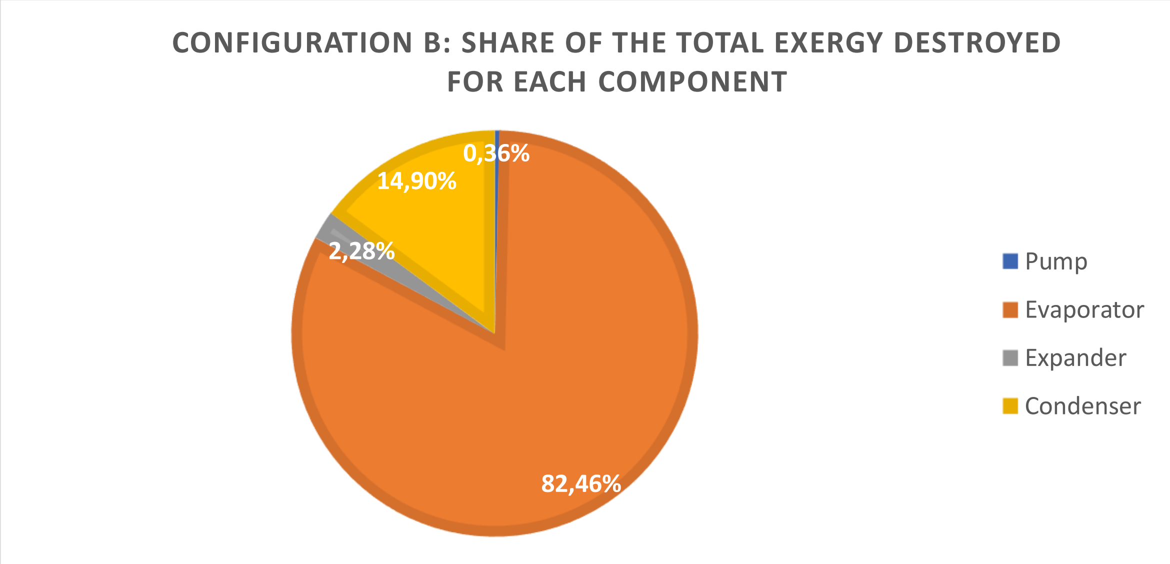

Figure 3.11. Share of the total exergy destroyed for each component of configuration B (standard ORC micro-CHP).

By analyzing Figure 3.11 it is possible to conclude that most of the exergy is destroyed in evaporator and in the condenser. This is justified since these devices are heat exchangers that permute heat between two distinct fluids. The share of exergy destroyed in the evaporator is very large, but it can be justified by the huge temperature difference between the inlet temperatures of the combustion gases and the organic fluid (r245fa). The exergy destroyed in the other two components (pump and expander) is practically insignificant when compared with the remaining values. This is justified since they only provide or extract work from the system, respectively, in small amounts when compared to the thermal power input and output.

The third configuration evaluated is the ORC micro-CHP system with intermediate vaporization of the organic fluid or abbreviated as intermediate vaporization (configuration C). This configuration is the most complex of them all, and has the largest number of components, because it has two intermediate circuits for the heat transfer (thermal oil and organic fluid). However, it is considered that exergy destruction only occurs to a relevant extent in five components: pump, evaporator, expander, condenser and heat exchanger. The share of exergy destroyed in each of these components can be seen in Figure 3.12.

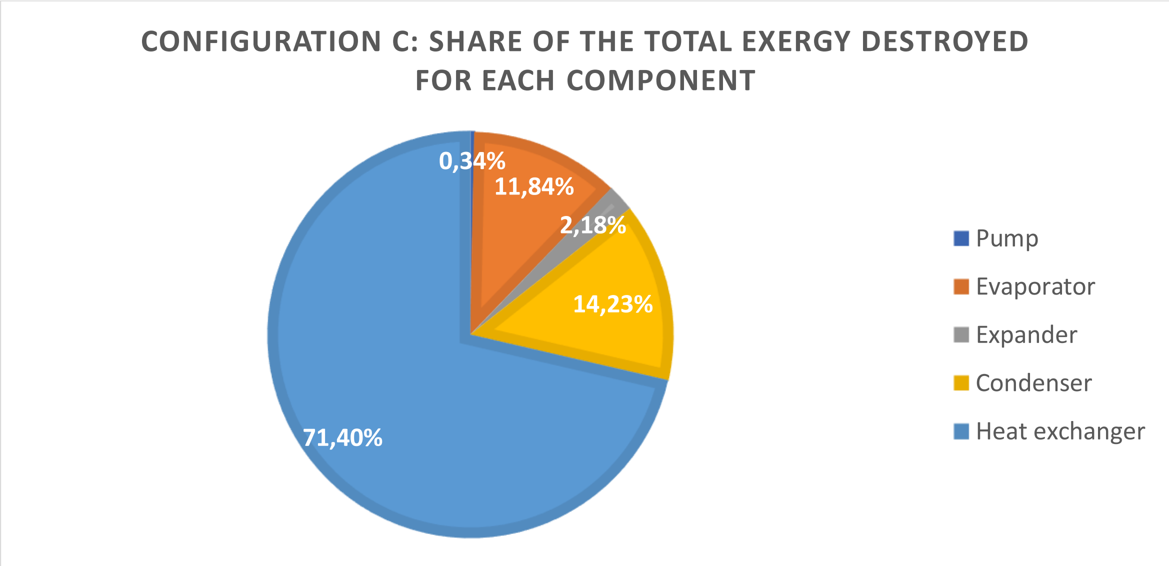

Figure 3.12. Share of the total exergy destroyed for each component of configuration C (IV ORC micro-CHP).

Evaluating the data present in Figure 3.12 it is possible to conclude that most of the exergy is destroyed in the heat exchanger. This is an acceptable conclusion since this heat exchanger ensures the heat exchange between the combustion gases and the thermal oil. The rest of the exergy destruction occurs mostly in the evaporator and the condenser. The exergy destroyed in the pump and in the expander is practically insignificant when compared with the remaining values. Like the results obtained for configuration B, most of the exergy is destroyed through heat exchange between fluids in the heat exchangers of this configuration.

Finally, the last configuration analyzed is the hybrid ORC micro-CHP (D). For this configuration were also considered five components: pump, evaporator, expander, condenser and heat exchanger. In this variant of the ORC micro-CHP technology there are two heat exchangers in contact with the combustion gases, which are the evaporator and the heat exchanger, and also two heat exchanging devices in contact with the water, the condenser and the heat exchanger. The share of exergy destroyed in each component can be seen in Figure 3.13.

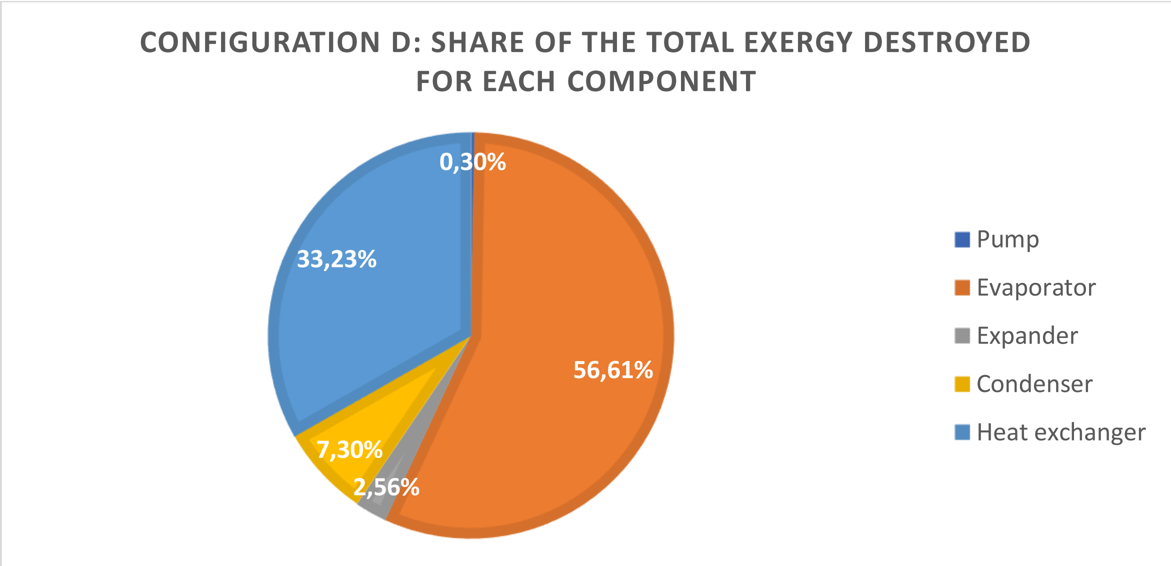

Figure 3.13. Share of the total exergy destroyed for each component of configuration D (hybrid ORC micro-CHP).

By interpreting the data of Figure 3.13 it can be concluded that the component that destroys most of the exergy is the evaporator. The exergy destruction in the heat exchanger and in the condenser is also very relevant. As for the pump and expander, the exergy destroyed in these components is almost residual when compared to the other components. Like the results obtained for configurations B and C, most of the exergy is destroyed through heat exchange between fluids in the heat exchangers of this configuration.

Concluding this relative analysis of exergy destruction in the configurations analyzed three general conclusions can be drawn. The first is that the vast majority of exergy is destroyed in the heat exchangers. The second is that within the heat exchangers, most of the exergy is destroyed in those that do the heat exchange between the combustion gases and another working fluid. Finally, it can also be stated that the exergy destroyed in the components that add or remove work from the system is much lower than the exergy destroyed in the other components.

3.4.4.2. Exergy destruction: absolute values

As previously stated, the total exergy destroyed per configuration was calculated in two different ways, as explained in section 3.2.2, more specifically through equations (3.31) and (3.32).

The first way to calculate the total exergy destroyed ($\dot{E}x_{destr,total(I)}$) considers that this parameter is the sum of the exergy destroyed in every component that integrates the configuration analyzed. This parameter is going to be designated as “Total exergy destroyed I” and its values for each configuration can be seen in Figure 3.14.

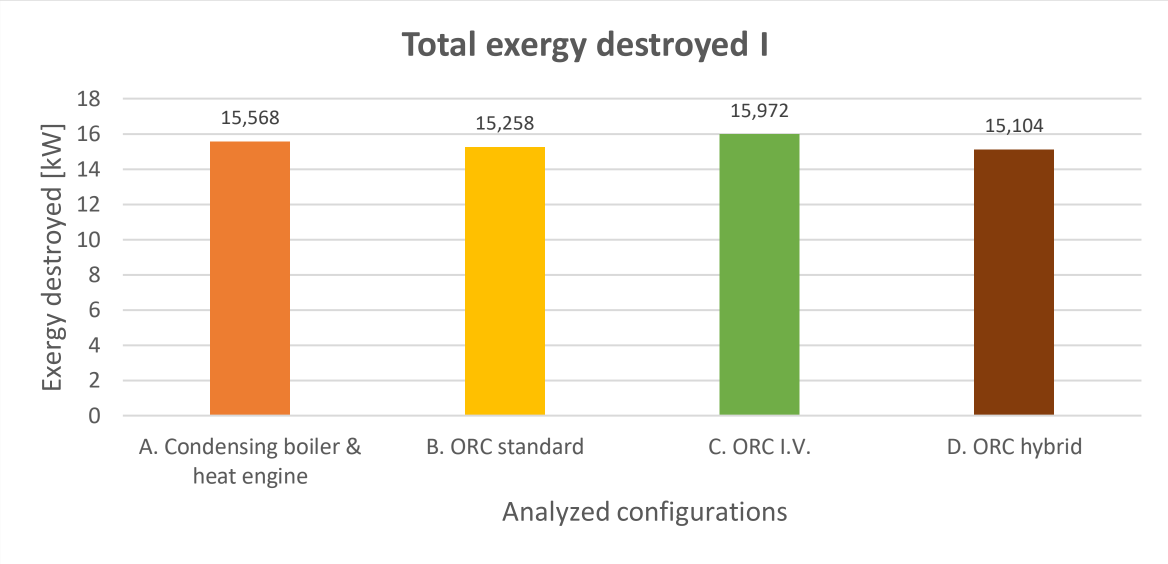

Figure 3.14. Total exergy destroyed I for the four analyzed configurations.

Through the analysis of Figure 3.14 it is possible to conclude that the configuration that destroys more exergy is the C (I.V. ORC micro-CHP), followed by configurations A (condensing boiler & heat engine), B (standard ORC micro-CHP) and D (hybrid ORC micro-CHP). This result implies that configuration D is the one that wastes the least of the system initial useful work potential. Therefore, this configuration, according to the considerations made in this form of calculation, is the best at the exergetic level. However, the difference between the values of the total exergy destroyed for the best and the worst configuration in only of 0,868 kW.

The second way to calculate the total exergy destroyed ($\dot{E}x_{destr,total(II)}$) is the simplest. It considers that this parameter is the difference between the initial exergy of the combustion gases (inlet exergy) and the net power work output of the system. This parameter is going to be designated as “Total exergy destroyed II” and its values for each configuration can be seen in Figure 3.15.

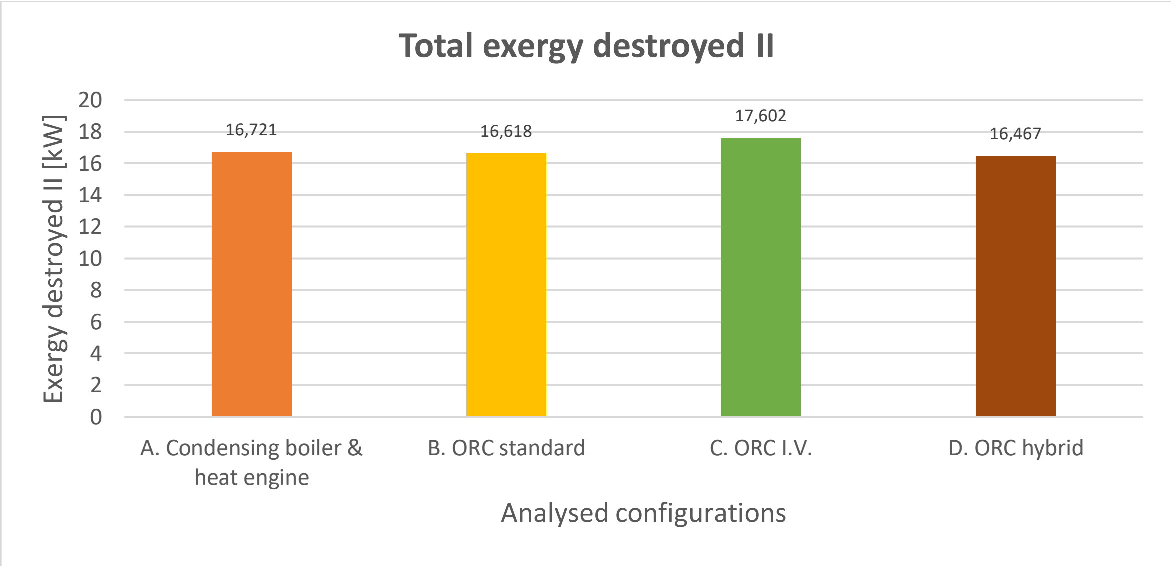

Figure 3.15. Total exergy destroyed II for the four analyzed configurations.

By analyzing Figure 3.15 it is possible to conclude, similarly to the parameter previously analyzed, that it is configuration C that destroys more exergy, followed by configurations A, B and D. This result implies, once again, that configuration D is the one that wastes the least of the system initial useful work potential. Therefore, this configuration, according to the considerations made in this form of calculation, is the best at the exergetic level. However, the difference between the values of the total exergy destroyed for the best and the worst configuration is only of 1,135 kW.

Conjugating the conclusions drawn for the total exergy destroyed in both its forms (I and II) it is possible to state that: i) the order of the configurations on the scale of the total exergy destroyed is the same for both calculation forms; ii) the worst result is obtained by configuration C; iii) the hybrid configuration is the best in an exergetic level; iv) the values obtained for both calculation forms are similar, differing in average by 1.377 kW (comparing the same configurations).

3.4.4.3. Sankey diagrams: exergy path

A Sankey diagram represents in a graphic and intuitive way the path and evolution of a property. In the case of exergy, it represents the entire path from the inlet to the outlet of the system. The initial exergy is supplied to the system through the combustion gases. Then, along the path of the exergy, there are various "losses" of exergy that occur through the destruction of exergy in the various components of a configuration. Finally, the diagram ends with the exergetic outputs of the system. This exergetic outputs are the exergy of the work used to produce electricity, and the exergetic content of the combustion gases and water at the system outlets.

Some conventions were used in the construction of the Sankey diagrams: i) the decrease of exergy of the system along the diagram is proportional to the "losses" occurred through the destruction of exergy; ii) the values displayed for the exergetic content are presented in their relative form (%); iii) the "losses" of exergy are represented with an arrow pointing downwards; iv) the useful exergetic content is represented horizontally, with the final exergetic output being represented with a right-pointing arrow; v) only the exergy that is converted into electricity is considered as useful exergy, which implies that the final exergies of the combustion gases and water are considered as exergetic "losses".

It is considered that the exergy destruction in configuration A occurs in the condensing boiler and in the heat engine. As for the exergy losses, these occur at the system outlets through the combustion gases and water flows. Therefore, the only useful exergy is used in the heat engine to convert work into electricity. The values for these parameters are graphically represented in Figure 3.16.

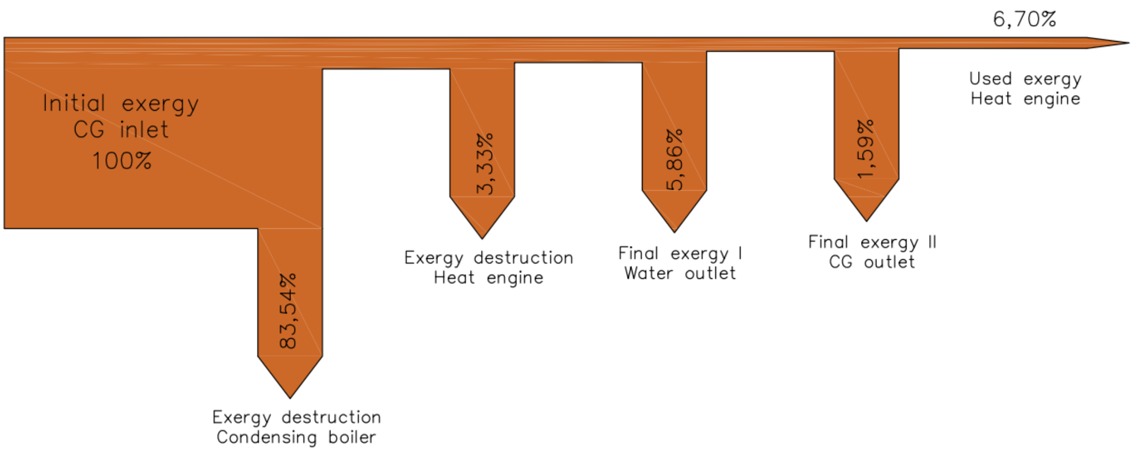

Figure 3.16. Sankey diagram A: exergy path in configuration A (condensing boiler & heat engine).

By analyzing Figure 3.16 it is possible to conclude that most of the exergy is lost through its destruction in the condensing boiler. The remaining losses, in descending order, are the exergy lost through the water outlet, the exergy destroyed in the heat engine and the exergy lost through the combustion gases outlet. The exergetic output of the system in the form of electricity represents only 6,70% of its initial exergetic content.

In configuration B it is considered that the exergy destruction occurs in the four components of the ORC (pump, evaporator, expander and condenser). As for the exergy losses, these occur at the system outlets through the combustion gases and water flows. Therefore, the only useful exergy is used in the expander-generator set to convert work into electricity. The values for these parameters are graphically represented in Figure 3.17.

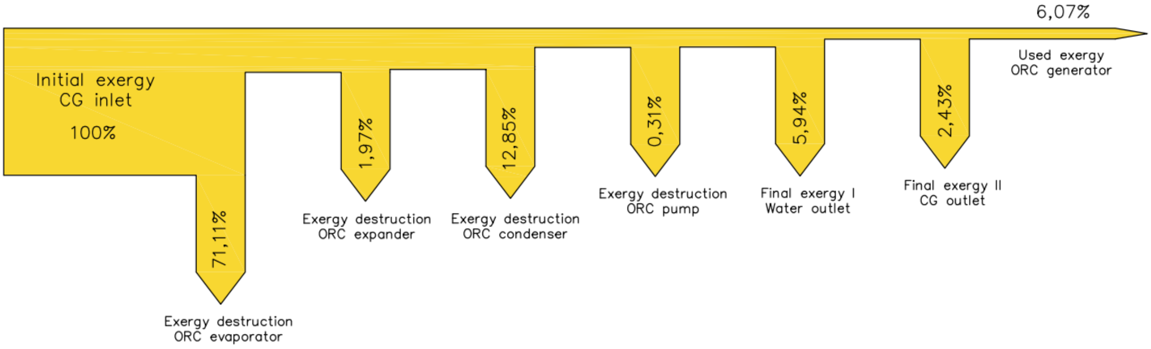

Figure 3.17. Sankey diagram B: exergy path in configuration B (standard ORC micro-CHP).

Through the analysis of Figure 3.17 it can be concluded that most of the exergy is lost through its destruction in the evaporator. The remaining losses, in descending order, are the exergy destroyed in the condenser, the exergy lost through the water outlet, the exergy lost through the combustion gases outlet, the exergy destroyed in the expander and the exergy destroyed in the pump. The exergetic output of the system in form of electricity represents only 6,07% of its initial exergetic content.

The third Sankey diagram regards configuration C, for which it is considered that the exergy destruction occurs in the four components of the ORC (pump, evaporator, expander and condenser) and in the heat exchanger of the thermal oil circuit. As for the exergy losses, these occur at the system outlet through the combustion gases and water flows. Therefore, the only useful exergy is used in the expander-generator set to convert work into electricity. The values for these parameters are graphically represented in Figure 3.18.

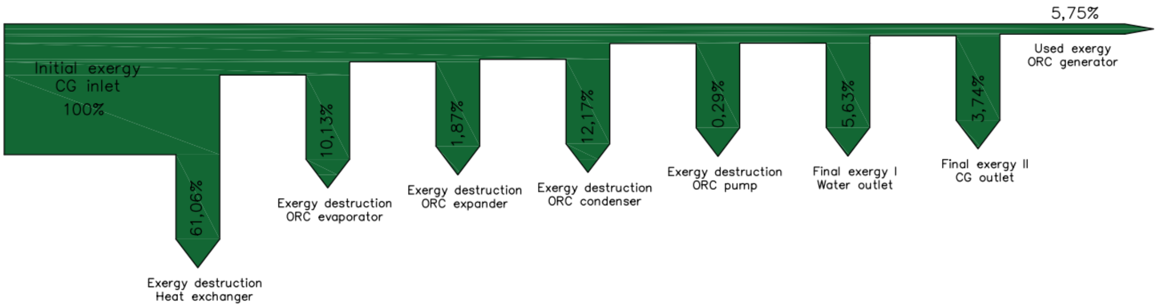

Figure 3.18. Sankey diagram C: exergy path in configuration C (I.V. ORC micro-CHP).

Through the evaluation of the data present in Figure 3.18 it is possible to verify that most of the exergy is lost through its destruction in the heat exchanger of the thermal oil circuit. The remaining losses, in descending order, are the exergy destroyed in the condenser, the exergy destroyed in the evaporator, the exergy lost through the water outlet, the exergy lost through the combustion gases outlet, the exergy destroyed in the expander and the exergy destroyed in the pump. The exergetic output of the system in form of electricity represents only 5,75% of its initial exergetic content.

Finally, the last Sankey diagram represents the exergy path for configuration D, in which the exergy destruction occurs in the four components of the ORC (pump, evaporator, expander and condenser) and in the heat exchanger of the water circuit. As for the exergy losses, these occur at the system outlets through the combustion gases and water flows. Therefore, the only useful exergy is used in the expander-generator set to convert work into electricity. The values for these parameters are graphically represented in Figure 3.19.

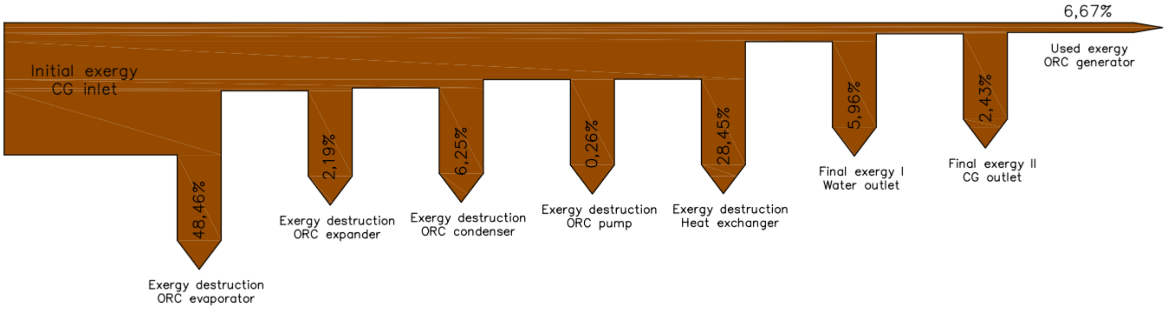

Figure 3.19. Sankey diagram D: exergy path in configuration D (hybrid ORC micro-CHP).

By analyzing the data present in Figure 3.19 it can be stated that most of the exergy is lost through its destruction in the evaporator. The remaining losses, in descending order, are the exergy destroyed in the heat exchanger of the water circuit, the exergy destroyed in the condenser, the exergy lost through the water outlet, the exergy lost through the combustion gases outlet, the exergy destroyed in the expander and the exergy destroyed in the pump. The exergetic output of the system in form of electricity represents only 6,67% of its initial exergetic content.

Once the conclusions have been drawn for each configuration it is possible to make some general conjectures about the exergy pathway: i) regardless of the configuration, most exergy is destroyed in the components that exchange heat between two fluids, namely evaporators, condensers and other additional heat exchangers; ii) the component in which most exergy is destroyed is the one that does the heat exchange between the combustion gases and another working fluid; iii) the exergy lost via destruction in the components that add or remove work from the system (pump and expander) is lower than the other exergy losses; iv) configuration A is the one in which the fraction of exergy used is the highest, closely followed by configuration D; v) the worst exergy utilization result is the one of configuration C; vi) the difference in exergy utilization between the best and worst configuration is only 0.95%; vii) in any of the configurations it can be considered that the use of the initial exergy is low.

The conjectures made in the previous paragraph are in line with what was found in the section 3.4.4.1. It is important to remember that one of the factors that largely contributes to the low usage of exergy in these systems is precisely its main final output. These devices are hot water production systems; therefore, it is acceptable that there is a large exergy destruction during the operation process of any of the configurations. This becomes even more evident considering that the power-to-heat ratio of the analyzed configurations is approximately 0.052. This concept, introduced in the section 2.3.6, means that for every 1 kW of thermal power produced in the form of hot water, only 0,052 kW of electricity are produced.

3.4.4.4. Parametric analysis of configuration D

As it was said in the beginning of section 3.4.4, the objective of the parametric analysis is to observe the influence of the inlet and outlet temperatures of the water in the second law efficiency of configuration D (hybrid ORC micro-CHP). It was considered that the total exergy destroyed is the sum of the exergy destroyed in each component of the hybrid configuration and the exergy supplied to the system is the exergy of the combustion gases at the inlet of the device. These considerations correspond to the B form through which the second law efficiency is calculated ($\eta_{II,B}$).

In the calculations from which the parametric analysis results, it was considered a variation interval of the water inlet temperature from 10°C to 50°C, with a step of 1°C. For the water outlet temperature, a variation range of 55°C to 85°C was considered, also with a step of 1°C. Since at each iteration of the program one of these temperatures' changes, then all the working fluids properties, intermediate variables and second law efficiency must be recalculated.

This parametric analysis was done in MATLAB and is represented in Figure 3.20.

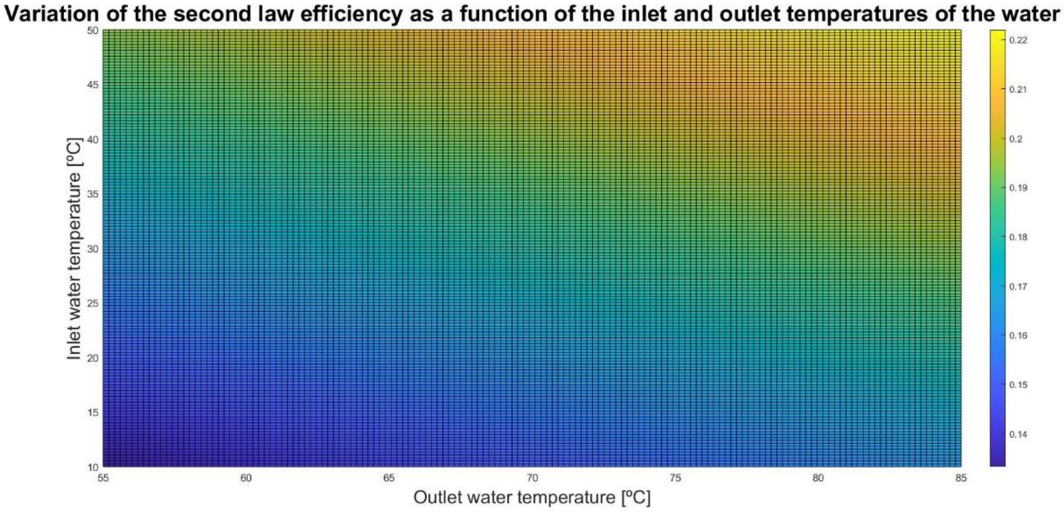

Figure 3.20. Parametric analysis of the second law efficiency of configuration D (hybrid ORC micro-CHP).

Trough the analysis of Figure 3.20 it is possible to concluded that the higher the temperature of the water at the inlet of the system, the better is it's the second law efficiency. The same is true for the water exiting the system, that is, the higher the temperature of the water at the outlet of the system, the better is it's the second law efficiency. That is, the higher the temperatures of the water, both at the inlet and outlet of the system, the higher is its efficiency according to the second law of thermodynamics. The second law efficiency, for the set conditions and the temperatures shown in the Figure 3.20, ranges from 13.5% to 22%.This means that the hybrid configuration takes better advantage of its initial useful work potential when the water sub-system is working at higher temperatures.

4. CONCLUSIONS

Globally, the objectives proposed for this study in the introduction were achieved. It was possible to characterize and compare the various technologies used in the European Union to provide hot water for domestic purposes and/or space heating. Through the energetic and exergetic analyses it was possible to compare the various analyzed configurations and ascertain the best one to produce hot water and electricity.

The main difficulties encountered while writing this master's thesis are mostly related to the search for data to compare technologies. There is little information regarding the parameters analyzed in section 2.4.1, and the information that does exist is scattered among several scientific articles and EU project reports. Therefore, it was necessary to collect, compile, analyze, process, comment and re-present all this openly available information. This was done to understand the current reality of domestic hot water and space heating residential systems in the EU.

From 1990 to 2019, fossil fuels (oil, coal and natural gas) were the main source of primary energy used for heating applications in the European Union. However, it is a trend that has been decreasing due to the environmental policies of the EU and its various countries.

The main technologies installed at residential level in the EU to provide hot water for domestic purposes and/or space heating as of 2019 were the non-condensing boilers (42,56%), the stoves (29,57%) and the electric radiators (14,01%). Moreover, the boiler technology overall represents 49,19% of all the equipments installed in the EU for the purposes mentioned above. Further analysis showed that the three technologies mentioned above are the most widely installed in the five EU countries with the largest heat demands as of 2020.

From the direct comparison between the various technologies installed in the EU to provide hot water for domestic purposes and/or space heating, several conclusions arise. Firstly, the technologies with the lowest investment cost, price [€] per unit of power output [kW], are the boilers and the electric water heaters. The micro-CHP systems are the most expensive and the heat pumps have an intermediate cost per unit of power output. As for the maximum temperature of hot water production, the technologies that present the highest values are the boilers and the electric water heaters, with values around 80°C. Again, heat pumps present intermediate values, within a range between 55°C and 70°C. No data regarding this parameter was found for the micro-CHP systems. However, it is expected that they present maximum water heating temperatures close to those of the condensing boilers. In terms of thermal efficiency, by far the best technology is the heat pump, followed by the condensing boilers and the electric water heaters. The least thermally efficient systems are the stoves. Within micro-CHP systems, the technology with the best thermal efficiency is the ORC. At an electrical level, the most efficient system is the solid oxide fuel cell (SOFC). As for energy sources, most of the technologies installed still rely on fossil fuels. The systems that use biomass are the stoves and the wood-fired boilers. The systems that use electricity are the electric radiators, the electric water heaters, and the heat pumps. There is the possibility of micro-CHP systems which resort to the SE or the ORC technologies to be integrated with renewable energy systems (*e.g*: solar thermal collectors) as their heat source. In general, the most versatile technology in terms of energy sources are the boilers, since there are boilers for solid, liquid and gaseous fuels. Most of the technologies analyzed assure residential needs of domestic hot water and space heating. The systems that obtain the worst result in the outputs parameter are the electric radiators and the stoves, since they can only provide space heating to one division. The systems with the highest number of outputs are the micro-CHP systems, as they combine the production of hot water for domestic purposes and for space heating with the production of electricity. From the direct comparison between technologies, it can be concluded that the boiler is still the best available technology to produce hot water at a residential level, for domestic purposes and for space heating. However, micro-CHP systems and heat pumps are two major threats to the boiler technology. Micro-CHP systems are a threat because they provide the same outputs as a boiler, with the added advantage of producing electricity. Heat pumps threaten the boilers because of their higher efficiency and their ease of operation.

A comparison was performed to evaluate the ability of the technologies to retrofit a non-condensing boiler. The highest scorers were the condensing boilers and the ORC micro-CHP systems. These results mean that these two technologies can directly replace a traditional boiler, with no need to perform changes to the facilities and their connections and guaranteeing the same outputs as the boiler.

At the energetic level configuration D (hybrid ORC micro-CHP system) is the one that obtains the best results. The worst results are obtained, depending on the parameter in question, by configuration A (condensing boiler & heat engine) or by configuration C (I.V. ORC micro-CHP system). Configuration B (standard ORC micro-CHP system) always presents intermediate values.

As for the exergetic analysis, for the second law efficiency in its A form, the most efficient configuration is the hybrid configuration (D). For the results of the B form, the configuration with intermediate vaporization of the working fluid (C) is the most efficient in an exergetic level.

Regarding the absolute exergy destruction, it was concluded that the configuration that, in total, destroys less exergy during its normal operation is the D, and the worst result is obtained by configuration C.

In terms of the relative evaluation of the total exergy destroyed it was concluded that most of the exergy is destroyed in the heat exchangers, especially in those that exchange heat between the combustion gases and another working fluid. Also, the exergy destroyed in the components that add or remove work to or from the system is much lower than the exergy destroyed in the other components of the various configurations analyzed.

Through the Sankey diagrams it can be concluded that the initial exergy of a system is destroyed in all its major components, namely the pump, the heat exchangers (evaporator, condenser or other type of heat exchanger) and the expander. The exergy contained in the water and in the combustion gases flows is wasted. That said, exergy is only used in the devices that convert work into electricity. For configuration A this happens in the heat engine, and in the other configuration's exergy is used through the expander-generator set. The conclusions drawn in the first relative evaluation of the total exergy destroyed are also true in the case of the Sankey Diagrams. Beyond these, it can be concluded that the configuration in which the percentage of exergy used is greatest is the A, with configuration D obtaining a slightly lower result. However, the share values of the exergy used do not vary significantly between configurations. This leads to the conclusion that the exergy used in any of the analyzed configurations is low.

The parametric analysis performed for the hybrid configuration (D) showed that the higher the water inlet temperature, and the higher the water outlet temperature, the better is the second law efficiency of this type of micro-CHP system.

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