

École polytechnique de Louvain

Thermal Integration of Carnot Batteries in energy systems

Providing a framework to evaluate the potential of Carnot Batteries

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Writing a master thesis is like going on a hike in the mountains ; postal cards always show wonderful landscapes and one gets excited to go for a journey which will be unforgettable. Even if this trip is expected from the start not to be easy and to ask for an adequate preparation to be sure to come back in a single piece, a nice view can be obtained from the top of the mountain once up there. Planning the journey on its own can be thrilling for newbies, exploring a whole new world, but it is wise to get advises from experienced mountain guides not to get lost into some dark cliffs in which one might get stuck. Additionally, having friends to discuss with during the hike is never a luxury as the thirst and dizziness that can appear throughout the day are unforgiving for lone travelers.

I would like in the first place to thank my supervisor Prof. Francesco Contino to have allowed me to go on this trip exploring what Carnot Batteries are and how they could be integrated in our energy systems. I would also like to thank him for getting me back on track when losing myself in the limbo of thermodynamics and energy systems by always being pragmatic.

I would also like to thank my mountain guide and PhD Antoine Laterre, who accompanied me during the whole year in my work and helped me to understand how on Earth a Carnot Battery does work. I would furthermore like to thank him for the support he showed me and for the time he devoted to me, as well as for the interesting discussions that we had over the year during our meetings.

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Acronyms

ARC Absorption Refrigeration Cycle. [18](#)

BELPEX M Belgian Index Spot Market. [52](#), [53](#)

Brayton-based CB Brayton-based Carnot Batteries. [10](#), [11](#)

BWR Back Work Ratio. [16](#)

CAPEX Capital Expenditure. [50](#), [52](#)

CB Carnot Battery. [1](#), [6–8](#), [10–12](#), [15](#), [17](#), [25](#), [26](#), [30](#), [32–34](#), [38](#), [39](#), [54](#), [61](#), [63–65](#), [67–70](#), [72](#), [73](#), [76](#)

CHP Combined Heat and Power. [2](#), [42](#), [43](#), [47–49](#), [51–53](#), [55](#), [56](#), [58](#), [59](#), [63](#), [67–69](#), [71–73](#), [78](#), [80](#)

COP Coefficient of Performances. [22](#), [32–35](#), [37](#), [64](#)

DHN District Heating Network. [17](#), [18](#), [35](#), [36](#), [43](#), [44](#), [64](#), [68–70](#)

GHE Greenhouse Emissions. [2–4](#), [49](#), [51](#), [53](#), [55](#), [56](#), [58](#), [59](#), [68](#), [71](#), [72](#)

GWP Global Warming Potential. [14](#)

HEX Heat Exchanger. [16–20](#), [22](#), [23](#), [25](#), [26](#), [30–32](#), [37](#), [63](#), [64](#), [72](#), [73](#)

HP Heat Pump. [15](#), [17](#), [20](#), [22](#), [25](#), [26](#), [32](#), [33](#), [63](#), [64](#)

KPI Key Performance Indicator. [6](#)

LCOE Levelized Cost of Energy. [2](#)

LCOS Levelized Cost of Storage. [13](#), [15](#), [17](#), [18](#)

LTES Latent Thermal Energy Storage. [10–15](#), [17](#)

NREL National Renewable Energy Laboratory. [50](#)

NSGA II Non-dominated Sorted Genetic Algorithm II. [54](#), [55](#)

- O&M** Operation and Maintenance. [50](#), [52](#)
- ODP** Ozone Depletion Potential. [14](#)
- ORC** Organic Rankine Cycle. [11](#), [13](#), [15–20](#), [23–26](#), [28–30](#), [33](#), [35–37](#), [40](#), [41](#), [63](#), [65](#), [67–69](#), [72](#), [73](#)
- P2P ratio** Power-to-Power ratio. [6–18](#), [28](#), [30](#), [32–38](#), [40](#), [41](#), [65](#), [67](#), [68](#), [70–73](#)
- PTES** Pumped Thermal Energy Storage systems. [6](#)
- PV panels** Photovoltaic panels. [3](#), [4](#), [42](#), [43](#), [49–51](#), [53–57](#), [67–69](#), [72](#), [78](#)
- Rankine-based CB** Rankine-based Carnot Batteries. [3](#), [4](#), [11–15](#), [17](#), [19](#), [25](#), [26](#), [28](#), [62](#), [63](#), [67](#)
- RHEIA** Robust design optimization of renewable Hydrogen and dErived energy cArrier systems. [54](#)
- SEC** Specific Equipment Cost. [63–65](#)
- SIC** Specific Cost Investments. [2](#), [63–66](#)
- STES** Sensible Thermal Energy Storage. [10](#), [12–15](#), [17](#), [65](#)
- TES** Thermal Energy Storage. [13](#), [14](#), [19](#), [22](#), [23](#), [25](#), [32](#), [63](#)
- TI** Thermally Integrated. [12](#), [13](#)
- TI-CB** Thermally Integrated Carnot Batteries. [7](#)
- TI-Rankine-based CB** Thermally Integrated Rankine-based Carnot Batteries. [5](#), [13](#), [15](#), [17](#), [18](#), [22](#), [67](#), [68](#), [73](#)
- VCHP** Vapor Compressed Heat Pump. [11](#), [13–15](#), [17](#), [19](#), [20](#), [22](#), [24](#), [33](#), [35](#), [37](#), [40](#), [41](#), [63–65](#), [67](#), [68](#)
- WT** Wind Turbines. [3](#), [4](#), [42](#), [43](#), [53](#), [55–57](#), [67–69](#), [72](#)

Chapter 1

Carnot Batteries : using Thermal Storage Systems to store electricity

“ You look at science (or at least talk of it) as some sort of demoralizing invention of man, something apart from real life, and which must be cautiously guarded and kept separate from everyday existence. ”

Rosalind Elsie Franklin

“ What's the use of doing all this work if we don't get some fun out of this? ”

Rosalind Elsie Franklin

With the increasing penetration of volatile renewable production units in energy system, storing energy and particularly electricity has become a major issue. The interest on screening the potential of newer energy storage system has grown, with the hope to find technologies that could provide an adequate energy storage at a low cost. Asides of energy storage systems as Pumped Hydro Energy Storage or Battery Energy Storage systems which have been used for decades or well-known promising technologies such as Power-to-Fuel, Flow Batteries or Compressed Air Energy Storage systems, a new type of storage system have been investigated more recently.

Carnot Battery (CB) are a type of storage system that stores energy under the form of heat. They consume electricity and heat during their charging cycle and reject heat and produce electricity during their discharging cycle. They are not *per se* a new technology (a patent dating from 1924 was deposited by Marguerre F. for a similar technology) but the interest on this type of storage has grown over the years, as it is usually studied as electricity-storing system.

As the motivation on developing Carnot Batteries has only grown recently, most of the studies were dedicated to determine the theoretical performances of this technology and prototypes have only been developed recently by academics or companies. Studies on the economic potential of such energy storage systems have also been conducted, to determine the economic viability of Carnot Batteries using metrics as the [Levelized Cost of Energy \(LCOE\)](#) or the [Specific Cost Investments \(SIC\)](#). However, there are only few studies which have assessed the real potential of Carnot Batteries, namely the potential of this technology as an energy storage system when integrated within a given energy system. This has actually been highlighted as one of the research gaps in the literature, as most of the studies were conducted on fixed conditions. By doing so, the impacts of operational conditions on the performances are neglected, which might lead to over-estimation of the real potential of Carnot Batteries.

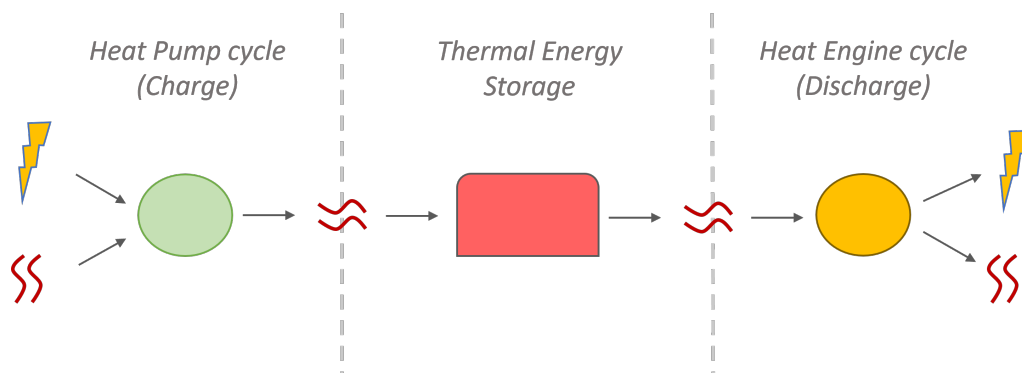


Figure 1.1: Schematic representation of a Carnot Battery working principle

UCLouvain energy system

Like other institutions, UCLouvain aims to reduce its [Greenhouse Emissions \(GHE\)](#) over its different campus all over Belgium and to actually become carbon neutral by 2035. To reach this goal, the university conducted in 2019 an impact analysis of its activities, whose results are shown in Figure 1.2, and it appeared that the energy production on the Louvain-la-Neuve campus is responsible for about 21% of the overall [GHE](#) in 2017. The campus includes academic buildings, student accommodations and shops among other consumers. The 14.3 [ktCO_{2,eq}] linked to the energy production on this campus are split between the generation of 52 [GWh] of heat over the year, which is currently done by boilers and [Combined Heat and Power \(CHP\)](#) units running on natural gas, and the production of 45 [GWh] of electricity by the [CHP](#) units or purchased from the grid.

In order to reach carbon neutrality but also due to the upcoming end of life of the [CHP](#) units, UCLouvain plans to change its energy system towards a fully renewable-based energy system. Green heat would be produced by a biomass [CHP](#) power plant owned by Veolia using biomass from recycling parks in the neighbourhoods of Louvain-la-Neuve ; the latter production unit could also sell green electricity to the UCLouvain. Even though biomass combustion is considered as carbon neutral by the Walloon and European directives [2], the heat and electricity produced by the [CHP](#) unit are actually not carbon neutral due to indirect [GHE](#) such as the transport of the biomass from the recycling

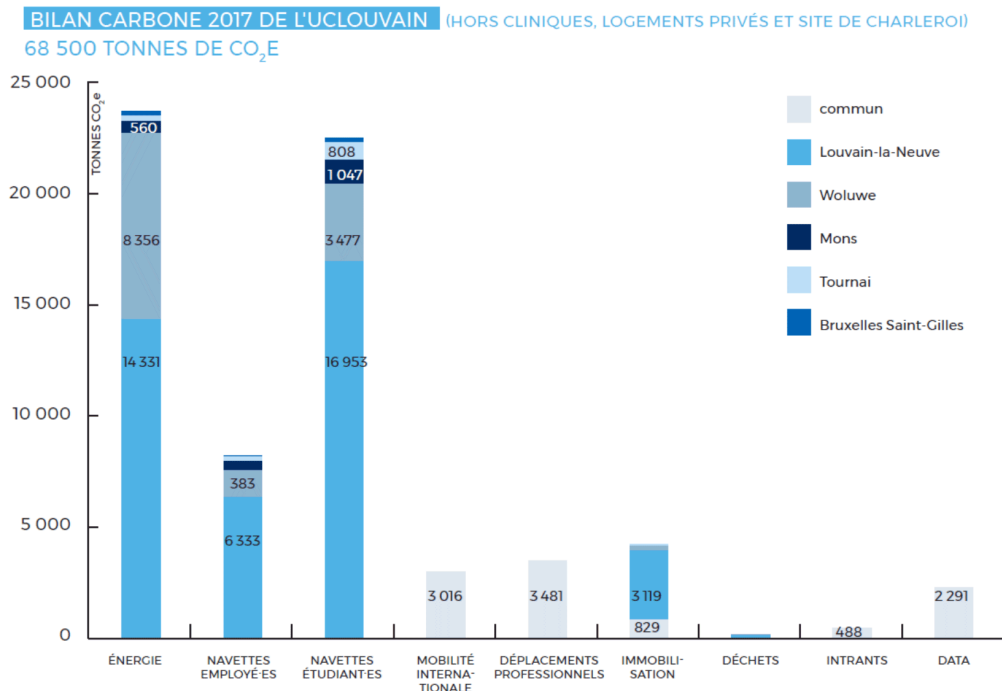


Figure 1.2: Results of the GHE study of the UCLouvain university in 2017 [1] ; it appears that energy production on the Louvain-la-neuve campus represents 21% of the overall carbon footprint.

parks to the powerplant. The installation of Photovoltaic panels (PV panels) and Wind Turbines (WT) to produce green electricity is thus planned to reduce the GHE linked to electricity production and to improve the self-sufficiency of the system.

Improving the electricity self-sufficiency of UCLouvain (hence, consuming more of the electricity produced by the PV panels and the WT) could in this way reduce the GHE linked to electricity production and reduce the system costs ; by considering the use of Carnot Batteries to do so, the potential of Carnot Batteries in operational conditions can be investigated for the UCLouvain energy system as consumption data is available.

Research objective and thesis organisation

The aim of this master's thesis is thus to close the research gap on the potential of Carnot Batteries in energy systems when used under operation conditions, by providing a framework to investigate the potential of integration of a certain type Carnot Battery (namely, Rankine-based Carnot Batteries (Rankine-based CB)) in a certain energy system (the one of the Louvain-la-Neuve campus of the UCLouvain).

As numerous types of Carnot battery architectures can be found in literature, the choice of Rankine-based CB must be motivated ; after explaining more in details what exactly are Carnot Batteries, a literature review on the types of Carnot Batteries and their performances is performed in Chapter 2.

A thermodynamic model for [Rankine-based CB](#) is then proposed in [Chapter 3](#), by presenting in a first time a general thermodynamic model for the overall cycle. In a second time, a model capturing the impact of operational conditions on the performances of the Carnot Battery is set up and then performances of the proposed model.

The potential of a storage system does not only depend on its thermodynamic performances but also on its price for a given design. Such an economic model needed to evaluate its potential is proposed in [Chapter 4](#) after a brief discussion of the existing cost estimate methods.

An in-depth presentation of the energy system in which the Carnot Battery is to be integrated is performed in [Chapter 5](#). The production units that will be considered in this energy system are also presented and a preliminary analysis of the optimal [PV panels](#) and [WT](#) sizing with respect to the annual costs of operation and the annual [GHE](#) is determined through a multi-objective optimization.

The integration scenarios of the Carnot Battery that could be considered for the energy system are presented in [Chapter 6](#) before discussing which parameters might have the most impact on the integration of a Carnot Battery in the energy system considered.

Last, perspectives for future work are discussed in [Section 7](#)

Chapter 2

Literature review

“ *Nothing in life is to be feared, it is only to be understood. Now is the time to understand more, so that we may fear less.* ”

Marie Skłodowska-Curie

The aim of this section is to perform a review of the performances of Thermally Integrated Carnot Batteries based on the literature. In order to understand those results, a clearer and more complete definition of what is a Carnot Battery is given in Section ???. Based on theoretical performances that can be found for different architectures of Carnot Batteries, the motivation to study [Thermally Integrated Rankine-based Carnot Batteries \(TI-Rankine-based CB\)](#) is detailed in Section ??. A review of the performances of [TI-Rankine-based CB](#) is then performed in Section ??.

2.1 Carnot Battery concept

CB, alternatively named **Pumped Thermal Energy Storage systems (PTES)** in the literature, are a family of energy storage systems which store energy under the form of heat [3, 4, 5, 6]. During the charging cycle, electricity is consumed to charge thermal exergy into in a thermal storage charge. The thermal exergy is stored during the storage duration, ideally with as less thermal exergy losses as possible and is converted back into electricity during the discharge cycle [3]. A sketch of this conceptual Storage system is shown in Figure 2.1.

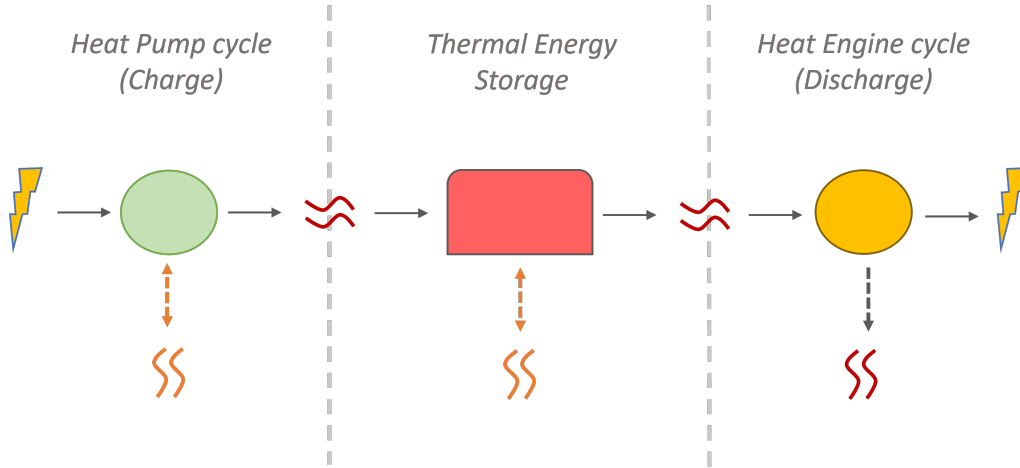


Figure 2.1: General concept of a Carnot Battery

As some authors suggest it, this definition could be extended even more by adding the possibility to exchange heat from or to the storage without considering conversion technologies as for the charge and the discharge cycle, namely by a direct heat exchange [7], even though such configurations have not been studied yet in the literature to the best author's knowledge.

2.1.1 Power-to-Power ratio

As they can absorb and discharge electricity, CB are usually considered as an electric storage system, and are thus characterized by their ability to 'efficiently store electricity' namely by how much electricity can be obtained back from the storage during the discharge cycle for each unit of electricity that was charged in during the charge cycle. This **Key Performance Indicator (KPI)**, named **Power-to-Power ratio (P2P ratio)**, is defined as follows :

$$\epsilon_{P2P} = \epsilon_{ch} \cdot \epsilon_{sto} \cdot \epsilon_{disch} \quad (2.1)$$

where ϵ_{ch} denotes how much heat is charged in the Carnot Battery per unit of electricity consumed, ϵ_{sto} how much energy is usable after a defined storage duration per unit of energy initially charged in, and ϵ_{disch} how much electricity is obtained per unit of heat discharged from the storage. In that sense, if some direct heat injection in the storage is

considered, ϵ_{sto} can be greater than one.

2.1.2 Thermal Integration of Carnot Batteries

The **P2P ratio** of most of storage systems is bounded to unity factor in the ideal case ; this is also the case when considering a **CB** layout with two isolated reservoirs at different temperatures between which heat is transferred during the charging and discharging cycles. The **P2P ratio** is then equal to unity factor if there is no temperature difference ΔT between the hot and cold sides during heat exchanges and asymptotically tends towards one for $\Delta T > 0$, as shown by Mercangöz et al. [8]. Such a layout is sometimes referred to as a **standalone configuration** [3] and is shown in Figure 2.2.

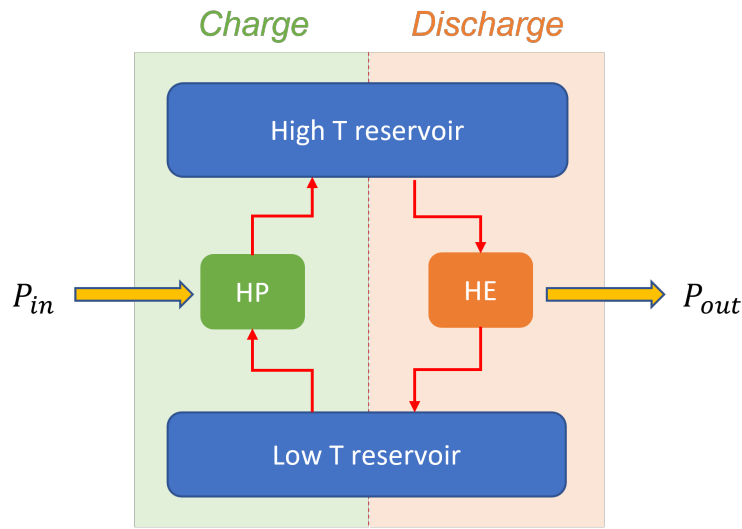


Figure 2.2: Standalone configuration schematic

However, if only one reservoir is used, an external heat source and an external heat sink must be considered ; the additional thermal exergy input coming from the temperature difference between the source and the sink can thus be used to produce electricity, thereby increasing the **P2P ratio** [9]. As the **P2P ratio** does not consider that additional energy input, values higher than unity can therefore be obtained [10, 11], which is the reason why the term 'Power-to-Power ratio' is preferred to the term 'round-trip efficiency'.

Carnot Batteries using waste heat sources at a higher temperature than the heat sink in such a configuration are generally referred to as **Thermally Integrated Carnot Batteries (TI-CB)** [3, 4, 11] and can be categorized in 'hot storage' configurations (using a high-temperature reservoir) and 'cold storage' configurations (using a low temperature reservoir), shown in Figures 2.3a and 2.3b. To the best author's knowledge, only one study explored the impact on the performances of a **CB** using a low temperature heat sink [12].

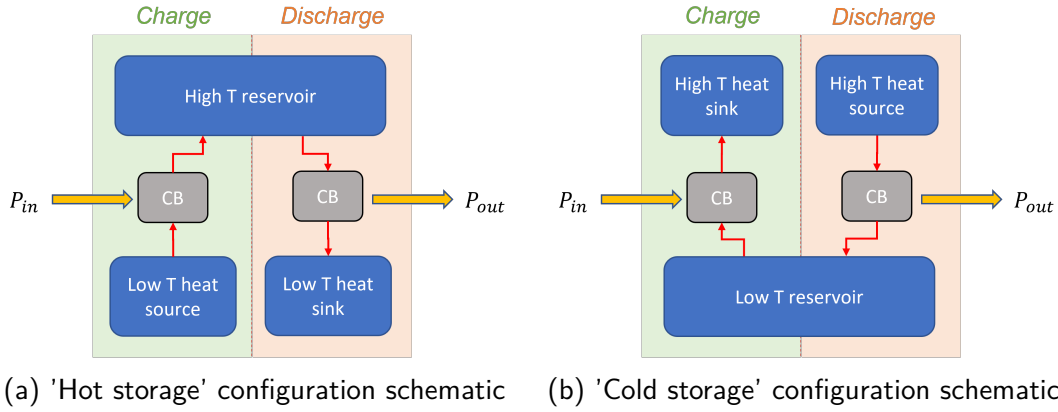


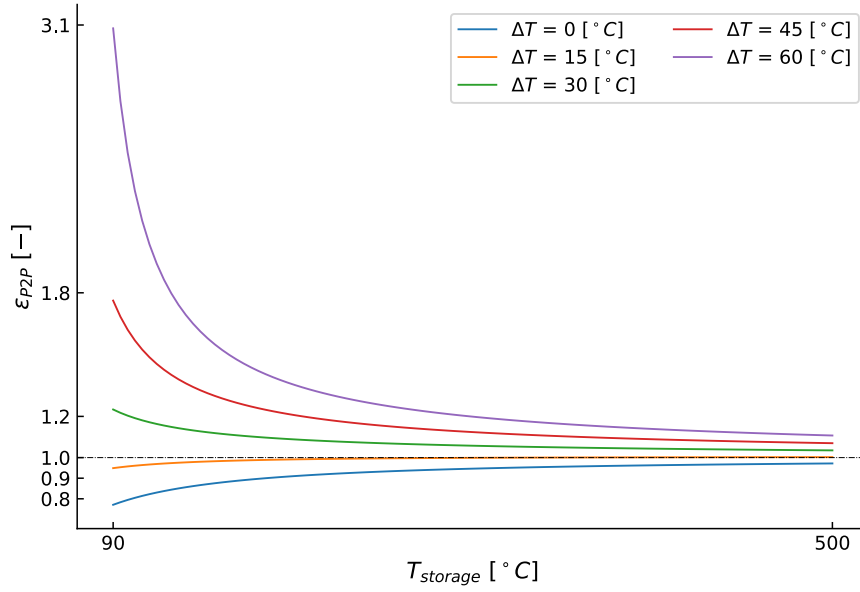
Figure 2.3: Thermally Integrated CB benefits from an additional thermal exergy input from the heat source to improve their P2P ratio, compared to standalone configurations

This boost in the CB performances is particularly remarkable for high temperature differences between the heat source and the heat sink ΔT as showed by Jockenhöfer et al. in [10] for hot storage configurations ; favoring the charging cycle performances by using a storage temperature as close as the heat source is thus more interesting than favoring the discharge cycle, as shown in Figure 2.4a. On the other hand, when using a heat source and a heat sink having a low temperature difference, improving the discharge cycle (and thus increasing the storage temperature) becomes more interesting to improves the cycle performances.

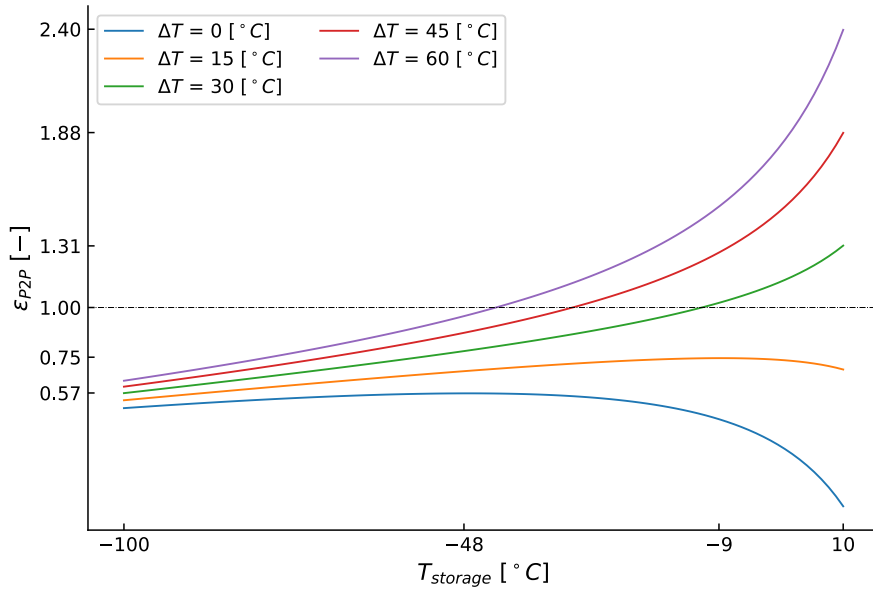
The same conclusions on the cycle to favor depending on the heat source and heat sink temperature difference can be obtained for cold storage configurations as shown in Figure 2.4b by determining the P2P ratio as in Equation (2.2) in the same fashion than in the article of Jockenhöfer et al. :

$$\epsilon_{P2P,cold} = \left(\frac{T_{sink} - \Delta T}{T_{sink} - T_{cold} + 2\Delta T} \right) \cdot \left(\frac{T_{source} - T_{cold} - 2\Delta T}{T_{source} - \Delta T} \right) \quad (2.2)$$

where T_{cold} refers to the temperature of the low-temperature reservoir, and T_{source} and T_{sink} to the temperature of the heat source and heat sink considered.



(a) Theoretical **P2P ratio** for hot storage configuration
 $(T_{sink} = 20 [^{\circ}\text{C}] ; T_{source} = T_{sink} + \Delta T)$



(b) Theoretical **P2P ratio** for cold storage configuration
 $(T_{sink} = 20 [^{\circ}\text{C}] ; T_{source} = T_{sink} + \Delta T)$

Figure 2.4: The **P2P ratio** is improved by the use of waste heat

2.2 Thermal classification

Having presented the general configurations in which a CB interacts with its storage system and eventually with external heat sources and heat sinks, a classification of the CB based on the thermodynamic cycles used for the charge and the discharge cycle can be proposed. The CB are mostly classified in the literature into Brayton-based Carnot Batteries, Rankine-based Carnot Batteries and Hybrid Carnot Batteries.

2.2.1 Brayton-based Carnot Batteries

Brayton-based Carnot Batteries (Brayton-based CB) use Reverse-Brayton and Brayton cycles as the charging and discharging cycles [6, 13]. In such cycles a gas (usually air or argon [14]) is used as the working fluid. Since the gas absorbs heat by increasing its temperature and rejects heat by decreasing its temperature and that the heat capacities of the aforementioned gases are relatively low, the range of temperature for such CB is usually rather broad. A low-temperature reservoir of -70 [°C] and a high temperature reservoir to $1,000$ [°C] is found in [15]. It thus appear that the Brayton-based CB performances difficultly benefit from the use of low-temperature heat sources [4].

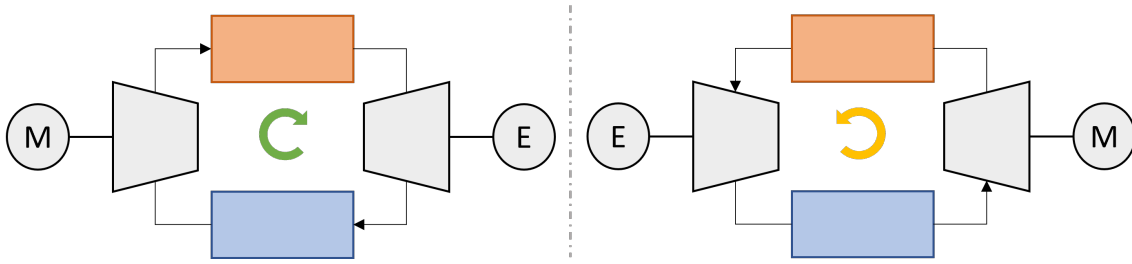


Figure 2.5: Classical Brayton-based CB layout

Classical layouts are composed of a compressor, an expander and two heat exchangers for both the charging and the discharging cycles, while Sensible Thermal Energy Storage (STES) system, storing heat by increasing the temperature of the considered media, are usually considered. Packed-bed STES are pretty common due to their low costs, large heat capacity and high cyclic temperature variations [6]. Latent Thermal Energy Storage (LTES), storing heat by operating a phase change, are practically never considered due to the large temperature ranges involved. Some authors propose to use recuperated Brayton cycles and two tanks systems to improve the P2P ratio [6], considering however an increase in the overall costs due to the additional internal Heat exchangers.

The values of P2P ratio for such CB that can be found in the literature are often in the range of $60 - 70$ [%] ; however, as it has been shown by White in [14] and by Steinmann in [5], Brayton-based CB are highly sensitive to the efficiency of the compressor and expander ; much lower values are obtained for efficiencies lower than 90 [%], down to 10 [%] for classical layouts when considering a compressor and expander efficiency of 80 [%].

2.2.2 Rankine-based Carnot Batteries

Rankine-based CB are a type of CB that uses Vapor Compressed Heat Pump (VCHP) and Rankine cycles as the charging and discharging cycles [3, 16, 17]. The working fluid in those cycles is operated in the neighbourhood of its saturation curve, namely as a vapor. The fluid extracts heat by evaporating at a pressure of extraction p_{extr} and releases heat by condensing at a pressure of rejection p_{rej} , and is compressed or expanded in-between. Rankine-based CB are operated with satisfactory performances at much lower temperatures than Brayton-based CB (usually below 200 [°C] [3, 4], except for some water/ammonia cascaded cycle proposed in [6, 18] with operating temperatures going up to 400 [°C]) and within a much smaller range of temperature. The later characteristic has been identified as particularly interesting, as it allows the use of LTES systems, which show large energy densities ; it also allows the use of pressurized water tanks, which is a cost-effective storage system whose operation is often limited to temperatures under 140 [°C] [3, 19].

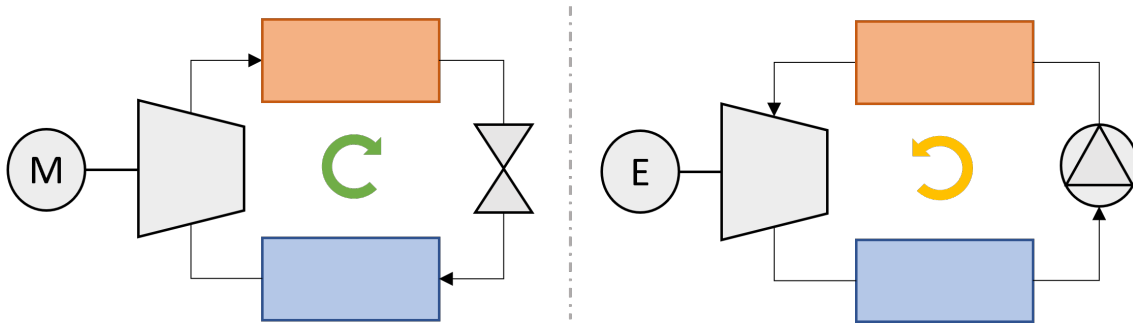


Figure 2.6: Classical Rankine-based CB layout, using a VCHP cycle to charge the CB and an Organic Rankine Cycle (ORC) cycle to discharge the CB

Transcritical cycles

Transcritical Rankine-based CB use working fluids with low-temperature critical points are compressed at high pressures [8, 20]. Due to the high pressure compression and the large variation of the specific heat capacity of the fluid, specific compressors and heat exchangers [6] which are not widespread yet must be used. Good P2P ratio were nonetheless reported for CO_2 transcritical Rankine-based CB by Mercangöz et al. for in [8] (between 51 [%] and 65 [%] using highly efficient machines for compressing/expanding the fluid $\eta_{is} \geq 80$ [%]) and Kim et al. in [21] (up to 63.5 [%] for isothermal transcritical cycles). Because of the low-temperatures of the fluids critical point (equal to 31.1 [°C] for CO_2 for example), waste heat integration on the cold side of the CB is only possible for relatively low-temperature waste heat source [10] ; waste heat integration on the high pressure side, on the other hand, remains challenging.

Subcritical cycles

Subcritical Rankine-based CB mostly use organics fluids, as they can be used under pressure at ambient temperature, avoiding the use of costly and bulky degassing equipment [3] and as their saturation curve usually fit well the heat source and sink temperatures

ranges used. Isentropic fluids (namely, fluids with an almost isentropic saturation curve) have been identified as particularly suited for CB [10, 11] as they limit the superheating of the steam in charging or discharging cycles needed to ensure dry steam to avoid damaging the compressors or the expanders, as shown in Figure 2.7.

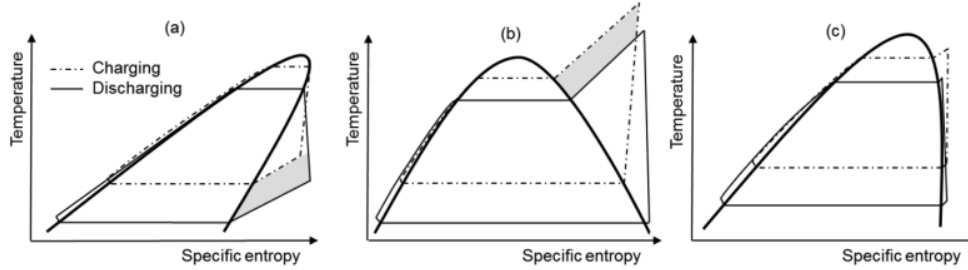


Figure 2.7: Dry (a), Wet (b) and Isentropic (c) fluids, adapted from [10]. Using dry fluids impose to superheat the steam in the charging cycle, which is damageable for the discharging cycle; similarly, using wet fluids impose to superheat the steam in the discharging cycle, degrading the charging cycle performances.

At the exception of the high temperature cascaded subcritical Rankine-based CB proposed in [6, 18] for which P2P ratio values around 70 [%] were reported, subcritical Rankine-based CB show poor performances without thermal integration. P2P ratio values of about 30 [%] were reported by Jockenhöfer et al. in [10] when considering a hot storage configuration using a combination of LTES and STES to store heat and considering an external temperature of $T_{source} = T_{sink} = 40$ [°C]. A pseudo-standalone configuration is proposed by Peterson in [22] considering a -15 [°C] LTES at the cold side and a STES at the hot side as well as a connection to the ambient ; if P2P ratio values of 50 [%] are claimed for isentropic efficiencies of 85 [%] and pinch-point temperature differences of 5 [°C] in the Heat Exchangers, the way the CB interacts with the ambient is not very clear. As a result, a boost in the discharging cycle appears by extracting thermal exergy from the ambient while considering its temperature fixed during both the discharge and the charge.

Thermal Integration

Thermally Integrated (TI) subcritical Rankine-based CB, on the other hand show much better performances when used with low-temperature heat sinks and satisfactory temperature differences between the source and the sink : P2P ratio from 70 [%] up to 130 [%] are often cited in the literature [3, 4, 11, 23, 24, 25, 26, 27] for waste heat sources at temperatures below 120 [°C]. Such an improvement in the performances of the Rankine-based CB leading to practically unmatched P2P ratio is actually the main reason why research on those storage system is favored [28].

Techno-economic studies have been conducted on the potential of CB as storage system, some of them considering fixed prices or fixed operation conditions [16, 17], others

mapping the potential of integration of Carnot Batteries considering dynamic models [26, 29, 30]; varying prices were identified as having a strong impact on the results, and can lead to an increase of 40 [%] in the **Levelized Cost of Storage (LCOS)** in the case as shown in [31]. Simultaneously, the sector coupling of **TI** subcritical **Rankine-based CB** with district heating systems has been highlighted in [10, 32, 27] as very interesting to improve the overall system performances, even though a detailed investigation of such integration is lacking.

2.3 Review on Thermal Integration

Taking into account the research that has been conducted and the scenarios that have been highlighted to be investigated, and due to the presence of a waste heat source stream that could be tapped in the UCLouvain energy system, the aim of this master's thesis is to close the research gap that exists in the domain of **TI-Rankine-based CB** concerning their integration within energy systems.

A review of the performances that have been reported in the literature and more specifically of the parameters that have been identified as highly sensitive is thus developed in this section, keeping in mind that a thermodynamic model able to capture the off-design operation of a **TI-Rankine-based CB** with a reasonable complexity is sought.

2.3.1 Typical charge and discharge cycles layouts

Even if the classical layout of **Rankine-based CB** presented in Figure 2.6 is rather common in the literature [10, 11, 17, 19, 33], many studies consider recuperated (or regenerated) **VCHP** and **ORC** cycles [9, 16, 23], subcooling the fluid exiting the condenser with the fluid exiting the evaporator for the **VCHP** cycle and preheating the fluid exiting the feed-pump with the fluid exiting the expander for the discharging cycle. Internal regeneration for a cycle is expected to improve its performances at a reasonable cost increase and becomes more common for both high temperature **VCHP** and **ORC** [9]. Some authors suggest other layouts, using for example an Organic Flash cycle instead of an **ORC** which considers the addition of a flash tank before the expander eventually expanding the fluid into the tank with a screw expander [24].

2.3.2 Thermal Energy Storage layouts

Considering the **Thermal Energy Storage (TES)**, it is of interest to reduce the exergy losses during the heat exchange with the storage [3]; for this reason, a combination of **STES** and **LTES** is considered in the studies conducted within the CHESTER H2020 European project framework [34]. The **LTES** is a 133 [°C] eutectic mixture used to exchange heat during the isothermal phase change while a pressurized water storage is used as the **STES** to match the **VCHP** subcooling and the **ORC** preheating. Considering such a storage configuration, **P2P ratio** values ranging between 40 [%] and 125 [%] are obtained for heat source temperatures ranging between 40 [°C] up to 100 [°C], a 15 [°C] heat sink and a pinch-point temperature difference $\Delta T_{pp} = 5$ [°C]. An additional heat exchanger is however needed at low temperatures as the heat ratio between the heat

exchange with the STES and the LTES must remain the same for the charge and the discharge.

There is however no significant difference in the performances obtained by Rankine-based CB with STES-only or LTES-only layouts. Eppinger et al. report in [23] overall similar P2P ratio ranging between 30 and 90 [%] and generally similar trends in sensitivity analyses for both types of TES for heat source temperatures, isentropic efficiencies considered for the compressor and the pinch-point temperature differences. LTES P2P ratio performances nonetheless show to be much more sensitive to the choice of the storage temperature (namely the maximum storage temperature for the STES and the phase change temperature for the LTES), than on the STES ; a decrease of the P2P ratio from 65 [%] at a storage temperature $T_{sto} = 110$ [°C] down to 35 [%] at a storage temperature $T_{sto} = 160$ [°C] is reported when using R1233zd(E) in a LTES layout, whereas variations in the P2P ratio smaller than 10 [%] around 52.2 [%] are obtained for the studied fluids in a STES layout.

It is noteworthy that, if a continuous spectrum of temperatures can be considered for STES, LTES can only operate at the state change temperature of the material considered, imposing design constraints on the storage temperature. Frate et al. does however not recommend to increase the spectrum of usable temperatures by considering less common Phase Change Materials in [3], as their technical feasibility, long-term stability and reliability might not have already been checked as for the reference PCMs.

2.3.3 Working fluid selection

Independently of the cycles or storage layouts, the vast majority of the studies uses organic fluids, and especially R1233zd(E) which has been frequently identified as a high performing fluid. Dumont et Lemort conducted in [11] a study over 16 different organic fluids using a to determine the P2P ratio ratio-optimal fluid for a classic Rankine-based CB layout used under hot and cold configurations. Different temperature lifts ΔT_{lift} (namely the difference of temperature between the evaporator and the condenser) in the VCHP and different heat source and heat sink temperatures were investigated and R1233zd(E), R1234Y, R236EA, R11 and R245FA reported the best performances. However, when considering environmental hazards such as Global Warming Potential (GWP) or Ozone Depletion Potential (ODP) and criteria such as flammability or toxicity as Pilai et al. in [35], R1233zd(E) and R1234Y are the only 'safe' fluids left from Dumont et al.'s selection.

The impact of the fluid choice for a heat source temperature was also studied by Frate et al. in [9] for 17 fluids meeting the European environmental legislation; they noted that the overall system globally increased with the heat source temperature T_{source} but that a maximum was reached for values of T_{source} close to the critical temperature. The best performances were once again obtained for R1233zd(E), reaching P2P ratio values of 130 [%] for an isothermal heat source at a temperature $T_{source} = 110$ [°C].

Considering two different fluids for the charging and discharging cycle has also been investigated in a few articles. R1233zd(E) was again identified as the best performing

fluid for a large range of source and storage temperatures by Hassan et al. in [36] ; they concluded that using Butene in the ORC when considering R1233zd(E) for the VCHP led to even better performances, a fluid pair that was used by Ökten et al. in [12] for a TI-Rankine-based CB. A multi-objective analysis was also conducted by Frate et al. [7] for different fluid pairs regarding the P2P ratio, exergetic efficiency¹ and energy density for a regenerated layout. They concluded that the use of Pentane for the VCHP and of R245fa for the ORC allowed to reach good performances for each of the three objectives.

2.3.4 Compressor and expander isentropic efficiencies

Compressor and expanders are often characterized by their ability to produce or extract work compared to an isentropic transformation, namely their isentropic efficiency. Ideally, this efficiency should be as close as possible to one, a lower value denoting the presence of irreversibilities. If most of the studies analyzed consider a constant efficiency for their Rankine-based CB model [4, 7, 17, 23, 24, 26], the high dependency of the P2P ratio on this parameter is well known [7, 19, 23]. Eppinger et al. reported a quasi-linear dependency of the P2P ratio on the isentropic efficiency of the compressor and expander $\eta_{is,comp/exp}$; for STES or LTES layouts, a variation of $\eta_{is,comp/exp}$ from 65 [%] to 85 [%] led to an increase of the P2P ratio value of about 25 [%] points. Ökten et al highlighted in [12] that the P2P ratio and the LCOS were actually more sensitive on the turbine efficiency than on the compressor efficiency, a conclusion that was shared by Hu et al. [17].

As the isentropic efficiency of a machine actually depends on operational conditions such as the pressure, the temperature but also the shaft speed, using a constant efficiency may lead to an over-estimation of the CB performances when used for a broad range of operational conditions. The experimental investigation of a small-scale Rankine-based CB conducted by Dumont et al. in [25] using a scroll expander shows indeed that the isentropic efficiency varies between 42 [%] and 75 [%] for the VCHP compressor and between 35 [%] and 62 [%] for the ORC expander ; the reported P2P ratio reached 72.5 [%] for a classic Rankine-based CB using R1233zd(E), a 75 [°C] heat source and a 15 [°C] heat sink, considering a temperature lift in the VCHP $\Delta T_{lift,HP} = 8$ [°C] and a temperature lift in the ORC $\Delta T_{lift,HP} = 49$ [°C]. Large variations in the isentropic efficiency of the compressor was also obtained for high-temperature Heat Pump (HP) in [37].

Dumont et al. use in [38] a simplified version of the semi-empirical model for scroll compressors and expanders initially proposed by Lemort in [39] and which was simplified and generalized for other volumetric machines in [33]. Volumetric machines offer satisfactory performances for small scale applications, in the range of the [kW_{el}] for scroll and piston expanders, up to 200 [kW_{el}] for screw expanders [3] ; a 260 [kW_{el}] semi-hermetic screw compressor was also mentioned in the study of Kosmadakis et al. [37]. Even though the use of radial inflow turbines is recommended for operation in

¹It is noteworthy that the definition used for exergetic efficiencies vary rather broadly over the literature so that the evaluation of performances of a fluid for this objective depends on the definition used, as the one of Frate et al. (taking into account the intermittency of the use of the waste heat source) differs from the one proposed by Jockenhöfer et al. in [10] or from the one considered by Fan et al. in [16].

the range of 50 to 500 [kW_{el}] [3], the use of parallel expanders could be thought of to increase the power capacity.

2.3.5 Feed-pump isentropic efficiency

The notion of **Back Work Ratio (BWR)** can be found in the literature to express the ratio between the work of the feed-pump and the work provided by the expander in Rankine cycles. For steam cycles, the **BWR** is usually small (a few percents) so that the feed-pump thermodynamic performance has negligible impact on the overall thermal efficiency ; it becomes however much more important in **ORC** for low critical pressures and low heat source temperature [40].

The definition of the feed-pump efficiency is also not always clear, some authors using the isentropic efficiency $\eta_{is,pump}$ while other rather use an internal efficiency definition $\eta_{i,pump}$ as defined in Equation (2.3), where w_{pump} is the actual work of the pump, $w_{is,pump}$ the work required for an isentropic compression and $\int_1^2 v dp$ the work related to the pressure increase across the pump. Other authors even consider the electro-mechanical losses through an electro-mechanical efficiency η_{mech} , representing the efficiency of the electrical conversion occurring on top of the feed-pump thermodynamic performance.

$$\eta_{is,pump} \triangleq \frac{w_{is,pump}}{w_{pump}} \quad \eta_{i,pump} \triangleq \frac{\int_1^2 v dp}{w_{pump}} \quad \eta_{mech} \triangleq \frac{w_{pump}}{w_{el,pump}} \quad (2.3)$$

Cavitation of the fluid in the pump is also much more common for organic fluids than for water, as their latent heat of vaporization and their evaporation temperature are much lower than the one of water [41], so that subcooling the fluid is critical to avoid damaging the feed-pump. On the other hand, the value of the subcooling has shown to have a determinant impact on the **BWR** and the thermal efficiency of **ORC** [40], and must ideally be minimized.

The impact of those parameters are however practically never considered in sensitivity analyses in the literature and fixed values are systematically used for the feed-pump efficiency (except in [38] where the model proposed by Landelle et al. in [40] is used) while subcooling are fixed without discussing the choice of such value, when they are considered.

2.3.6 Heat Exchangers performances

Heat Exchanger (HEX) are a key component of Carnot Batteries as the pinch-point temperature difference ΔT_{pp} (also referred to as the pinch) across the **HEX** should be minimized as much as possible [10, 23, 42]. Dumont et al. claim that a 1 [°C] increase in the pinch leads to a 3 [%] decrease in the **P2P ratio** ; minimizing the pinch can hence be done by oversizing the **HEX** compared to other applications. Designing the **HEX** for a given pinch can be done as in [43] iteratively, as the area and the pinch are calculated based on the inlet temperatures and mass-flow rates of the two fluids in the **HEX** and on the heat transfer coefficients.

The calculation of the heat transfer coefficients is actually particularly hard as it depends on many parameters such as the geometry, material properties and state properties as highlighted by Steger et al., who proposed in [44] a model to fit experimental data of a 100 [kW] brazed HEX used for reversible Rankine-based CB using R1233zd(E). Pressure drops of 0.13 [bar] were observed in the HP mode of the HEX whereas pressure drops of 0.33 [bar] were obtained when using the HEX in the ORC mode.

The results obtained and the underlying analytical model must however be validated with other experimental data on the word of the authors, which can be understood by the fact that correlations to find the heat transfer coefficient are usually valid for a specified range of validity and as they depend on the type of HEX used, extrapolating those results to other types of HEX is complicated.

As the type of HEX that would be used in the CB has not been defined yet, it is actually difficult to size those components without choosing the type of HEX used or without determining an optimal design. The choice of technology for HEX is usually done based on the thermal capacity considered [3] while the determination of an optimal design would typically require the use of Computational Fluid Dynamics tools, which are rather computational intensive.

For this reason, most of the studies consider a fixed pressure drop across the HEX and a fixed pinch-point temperature difference to model their HEX ; even if this method highly simplifies the HEX behavior by not reflects the part-load degradation of the HEX performances, its provides enough information to easily assess the thermodynamic performances of a CB in steady-state. The accuracy of dynamic studies might benefit from a more thorough model if the type of HEX used can be determined beforehand.

2.3.7 Techno-economic assessment

Only a few studies have assessed the techno-economic potential of TI-Rankine-based CB [17, 12, 30], all of them considering steady-state thermodynamic models for the TI-Rankine-based CB and none of them considering LTES-only layouts to the author's best knowledge.

Hu et al. studied in [17] a classical VCHP and ORC layout with a STES using R1233zd(E). They performed a multi-objective optimization on both the P2P ratio and the LCOS. The LCOS was calculated by considering a fixed amount of discharging hours per year and a fixed electricity price (namely 2,190 [$h/year$] and 42.27 [$€/MWh$]) while determining the CB price by decomposing it into its elementary components (i.e. heat exchangers, compressor, turbines, etc).

Their results suggest that Rankine-based CB could compete with other storage system, reporting values of LCOS as low as 180 [$€/MWh$] when considering free waste heat sources and as high as 400 [$€/MWh$] when considering that heat is purchased from a District Heating Network (DHN) at a fixed price of 25.4 [$€/MWh_{th}$], for P2P ratio close to 80 [%].

Another important conclusion from their study is that minimizing the **LCOS** and maximizing the **P2P ratio** are competitive objectives ; the lowest **LCOS** are obtained mostly obtained for the lowest **P2P ratio**. This can be explained by the fact that high **P2P ratio** restrict the discharged power and thus amount of electricity that can be discharged and that is considered in the **LCOS** – which do not counterbalance the reduction in the costs linked to the purchase of electricity compared to lower **P2P ratio**. They also highlight the fact that a larger use of the waste heat (through a higher mass-flow rate at the heat source) leads to a lower **LCOS** as the charging power (and thus the energy stored) is greater for a same **P2P ratio**. Last, when considering different applications scenarios, for which a cycle heat source temperature $T_{source} = 85$ [°C] is considered in each case, the purchase of heat in a **DHN** heat source scenario (at a price of 25.4 [€/MWh_{th}]) results in a **LCOS** about 1.5 times higher than when using industrial waste heat, even when considering additional investment costs linked to the purchase of a specific intermediate **HEX** and storage tanks to tap the heat source.

Ökten et al. considered in [12] the addition of an **Absorption Refrigeration Cycle (ARC)** to a **TI-Rankine-based CB** to improve its performances, using industrial waste heat. This source was also tapped by using intermediary storage tanks and an additional **HEX**, providing heat at a temperature of 90 [°C] with a glide of 25 [°C]. Their results showed that the introduction of the bottoming refrigeration cycle significantly improved the **P2P ratio** by reducing the condensation temperature of the **ORC**, with **P2P ratio** values of 66 [%] and 91 [%] without and with the **ARC** ; the additional costs linked to the addition of the **ARC** led to higher investment costs but to a lower **LCOS** (254 [€/MWh_{el}] without **ARC** and 234 [€/MWh_{el}] with). The energy considered to calculate the **LCOS** was, as for Hu et al., determined by the discharging power considering a certain amount of discharge hours per year, equal to 2,800 [h/year] for the figures of merit obtained.

Chapter 3

Thermodynamic model of the Carnot Battery

“ I am somewhat exhausted; I wonder how a battery feels when it pours electricity into a non-conductor ? ”

Arthur Conan Doyle, 1913, *"The Adventure of the Dying Detective"*

As it has been highlighted in the literature review, few models of Thermally Integrated Rankine-based Carnot Batteries capturing the impact of an off-design operation of the storage system have been developed. Such models are however needed to assess more clearly the potential of Carnot Batteries in energy systems.

However, the potential of integration is assessed in this work by optimizing a set of parameters by using a genetic algorithm performing a consequent number of simulations (as explained in 4.5.3) ; **a low computational demanding model** is required in order to determine the optimal sets of parameters given limited computational resources.

The general thermodynamic modeling of the Rankine-based CB is presented in a first time in Section 3.1 by identifying the equations and parameters used to model the VCHP, the TES, the ORC and the HEX.

A brief discussion on which parameters must and can be modelled more thoroughly to capture the off-design operation of the Rankine-based CB is then performed in Section 3.2, and a mapping of the CB performances is presented in 3.3 for different charging and discharging conditions and important design parameters.

3.1 General thermodynamic model

Rankine-based Carnot Batteries can be represented as in Figure 3.1 as being made of three sub-systems: a VCHP, used to extract heat from a low-temperature heat source to a higher temperature heat sink, a TES system used to store heat at high temperature and

an **ORC** used to convert heat into electricity by discharging heat from a high temperature heat source to a low temperature heat sink. A basic thermodynamic model is proposed for each of them and as those three subsystems interacts with each others through **HEX**, a simplified thermodynamic model is proposed for this component.

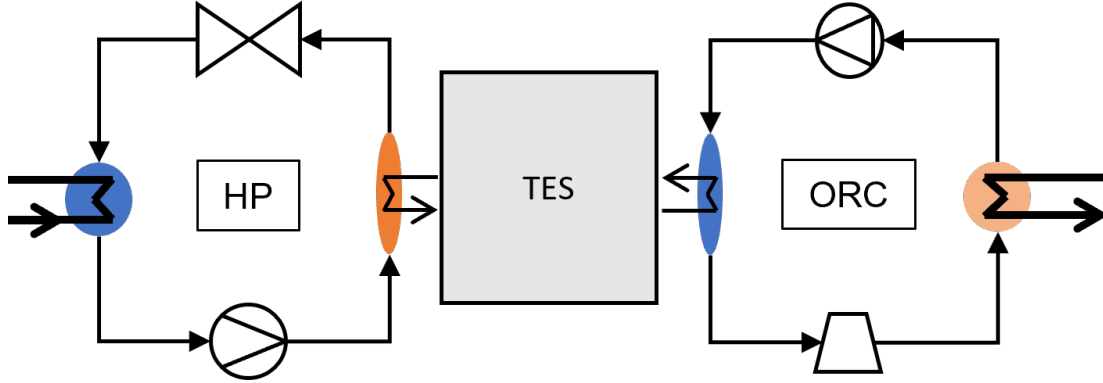


Figure 3.1: Rankine Carnot Battery conceptual schematic

3.1.1 Vapor Compressed Heat Pump cycle modelling

The **VCHP** cycle is modelled by four transformations shown in Figures 3.2a and 3.2b : a low-temperature **heat extraction** in a **HEX** done by evaporating the working fluid at low pressure (phase 0-1) ; a **compression** of the working fluid to a higher pressure (phase 1-2) ; a high-temperature **heat rejection** in a **HEX** done by condensing the fluid at high pressure (phase 2-3) ; an **expansion** of the working fluid performed by throttling it back to a lower pressure through a valve (phase 3-0). A summary of the **HP** cycle-dependent parameters is given in Table 3.1.

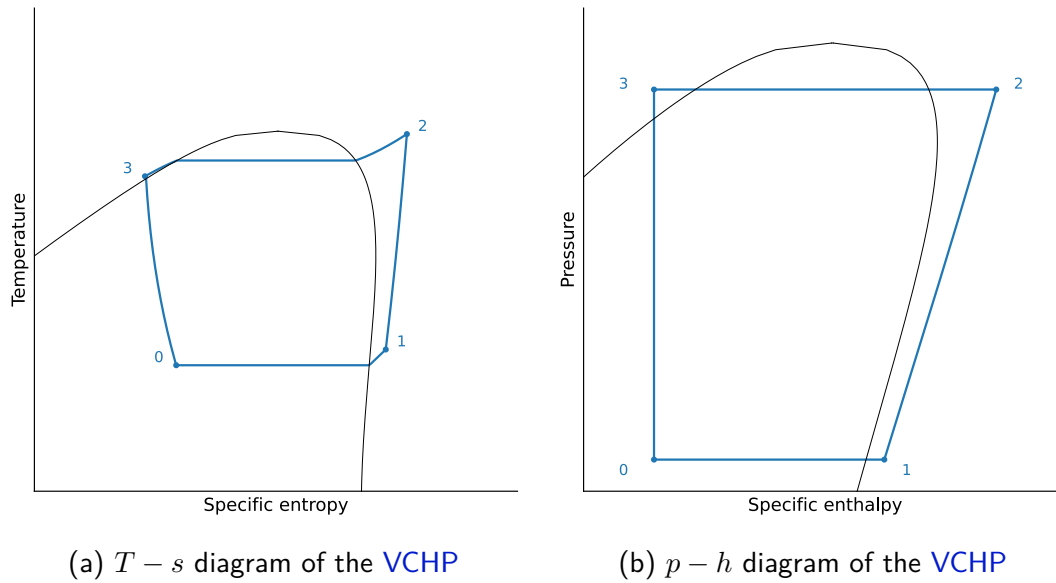


Figure 3.2: Diagrams of a typical **VCHP** cycle, using R1233zd(E) as the working fluid

Transformation 0-1 : Heat Extraction

The heat extraction process is done by evaporating the fluid at a low temperature T_0 (so at a low pressure p_0) fixed by the heat source. The fluid is then superheated at constant pressure to $T_1 = T_0 + \Delta T_{sup,HP}$ to avoid the formation of droplets in the compressors which could damage it.

$$h_0 \rightarrow h_0'' \quad T_0'' \rightarrow T_1 \quad (3.1)$$

The total amount of heat absorbed from the heat source $q_{so,HP}$ is defined as in Equation (3.2) :

$$q_{so,HP} \triangleq h_1 - h_0 \quad (3.2)$$

Transformation 1-2 : Compression

The compression of the working fluid increases the pressure of the working fluid from p_1 to p_2 . The work needed for the compression $w_{in} := h_2 - h_1$ can be calculated through the isentropic efficiency of the compressor $\eta_{is,comp}$, defined as follows :

$$\eta_{is,comp} \triangleq \frac{h_{2s} - h_1}{h_2 - h_1} \quad (3.3)$$

where h_2 is the enthalpy after the compressor and h_{2s} is the enthalpy that the working fluid would have if the compression would have been isentropic.

Transformation 2-3 : Heat rejection

During the heat rejection process, heat is first rejected at high temperature by desuperheating the working fluid from T_2 to the saturation temperature T_2'' followed by a condensation of the fluid at a pressure p_2 fixed by the heat sink. The fluid is also usually subcooled by a temperature difference $\Delta T_{sub,HP}$ after the fluid is condensed up to T_3 .

$$T_2 \rightarrow T_2'' \quad h_2'' \rightarrow h_2' \quad T_2' \rightarrow T_3 \quad (3.4)$$

The total amount of heat rejected to the heat sink $q_{si,HP}$ is defined as in Equation (3.5) :

$$q_{si,HP} \triangleq h_2 - h_3 \quad (3.5)$$

Transformation 3-0 : Expansion

The throttling of the working fluid is modelled as an isenthalpic expansion through a valve during which the fluid is expanded from a pressure p_3 to a pressure p_0 , thus closing the cycle.

$$h_3 = h_0 \quad p_3 \rightarrow p_0 \quad (3.6)$$

Performances of the cycle

The performance of a HP is captured by its Coefficient of Performances (COP), representing the amount of heat delivered at high temperature $q_{si,HP}$ per unit of electricity consumed w_{in} and defined as in Equation (3.7), and is hence always greater than unity factor :

$$\text{COP} = \frac{q_{si,HP}}{w_{in}} = \frac{q_{so,HP} + w_{in}}{w_{in}} \quad (3.7)$$

Heat extraction		Compression	
Superheating [K]	$\Delta T_{sup,HP}$	Isentropic efficiency [–]	$\eta_{is,comp}$
Heat absorption			
Subcooling [K]		$\Delta T_{sub,HP}$	

Table 3.1: HP cycle-related parameters

3.1.2 Thermal Energy Storage modelling

The Thermal Energy Storage system considered is a Sensible TES used in a hot storage configuration; heat is hence extracted from the VCHP cycle and stored by elevating the temperature of the storage media by a temperature difference $\Delta T_{spread,sto}$ (referred to as **the spread**) from $T_{TES,low}$ to $T_{TES,high}$ (which will be referred to as T_{sto} from hereon). When discharging heat the temperature of the storage media decreases by the same temperature difference.

Pressurized water tanks are commonly proposed as the storage system, as they are the most cost-effective solution for liquid storage system [3], which are easier to heat and cool down during the charging and discharging processes than solid media storage systems [16]. The heat storage can be done by using two tanks at different temperatures between which the pressurized water is circulated as shown in Figure 3.3a or by using a single tank in which a thermocline (a temperature gradient over the storage) is created as shown in Figure 3.3b.

The choice of the heat storage design is a trade-off between cost and performance, as using one tank is cheaper but mixing occurring in the tank reduces the discharge TES temperature over time ; on the other hand, two-tanks configurations are more expensive but ensure a constant temperature difference over the discharge HEX. As an ideal thermocline is considered in most studies using a single tank for TI-Rankine-based CB, the effect of mixing as not been assessed yet [3].

The amount of heat that the storage can store E_{sto} is directly proportional to the mass of water contained m_{water} , the mean heat capacity over the considered temperature $c_{p,water}$ range and the spread over the storage ΔT_{spread} .

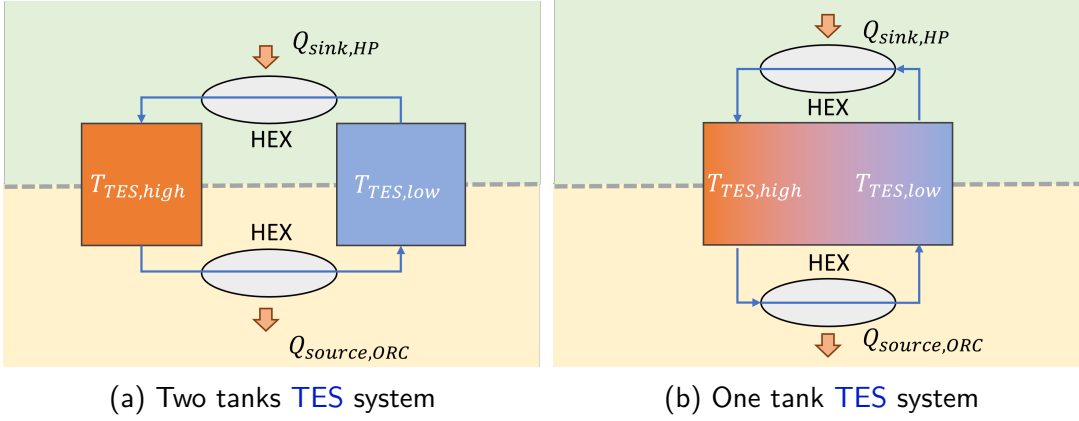


Figure 3.3: Different possible configurations of TES, with
 $T_{TES,high} = T_{TES,low} + \Delta T_{spread}$

Performance of the storage

The storage is characterized by its storage efficiency ϵ_{sto} , which reflects how much heat can be discharged from the storage per unit of heat injected in the storage. In the absence of direct heat exchange from/to the storage, ϵ_{sto} is smaller than one and captures the thermal losses and irreversibilities within the storage.

Thermal Energy Storage			
Maximum temperature [K]	T_{sto}	Storage spread [K]	$\Delta T_{spread,sto}$
Storage capacity [kWh]	E_{TES}	Storage efficiency [—]	η_{sto}

Table 3.2: TES characteristic parameters

3.1.3 Organic Rankine Cycle modelling

The Organic Rankine Cycle is modelled by four transformations shown in Figures 3.4a and 3.4b : a pressure increase of the liquid fluid in a pump (phase A-B) ; a **heat absorption** at high pressure from the heat source in a HEX (phase B-C) ; an **expansion** of the vapor through a turbine producing electricity (phase C-D) ; a **heat rejection** to the low temperature heat sink (phase D-A). The ORC cycle-dependent parameters are thus listed in Table 3.3.

Transformation A-B : Compression

The fluid is compressed under a liquid form in a pump from a pressure p_A to a pressure p_B ; the mechanical work needed $w_{pump} := h_B - h_A$ can be obtained by using the isentropic efficiency of the pump defined as follows :

$$\eta_{is,pump} \triangleq \frac{h_B - h_A}{h_{B,s} - h_A} \quad (3.8)$$

where h_B is the enthalpy of the fluid after the compressor $h_{B,s}$ is the enthalpy that the fluid would have if the compression was isentropic.

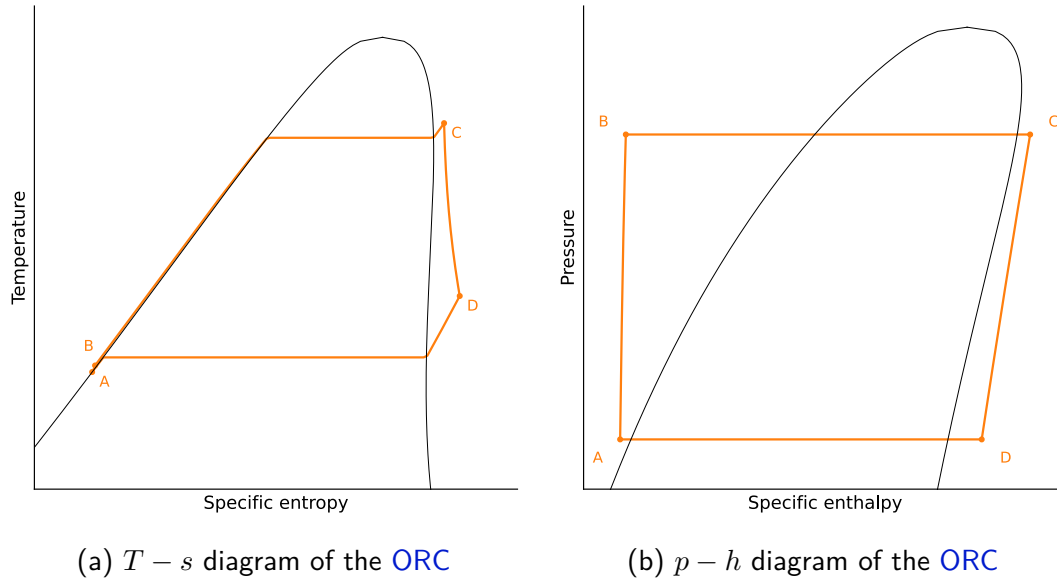


Figure 3.4: Diagrams of a typical VCHP cycle, using R1233zd(E) as the working fluid

Transformation B-C : Heat extraction

The fluid is preheated at constant pressure from T_B until it reaches the saturation temperature $T_{C'}$. The fluid is completely evaporated and is then superheated up to a temperature $T_C = T_{C''} + \Delta T_{sup,ORC}$ to avoid the formation of droplets in the turbine, which could damage it.

$$T_B \rightarrow T_{C'} \quad h_{C'}' \rightarrow h_{C''} \quad T_{C''} \rightarrow T_C \quad (3.9)$$

The total amount of heat absorbed from the heat source $q_{so,ORC}$ is defined as in Equation (3.13) :

$$q_{so,ORC} \triangleq h_C - h_B \quad (3.10)$$

Transformation C-D : Expansion

The working fluid is then expanded from a pressure p_C to a pressure p_D to produce mechanical work. The work $w_{exp} := h_C - h_D$ can be calculated through the isentropic efficiency of the expander $\eta_{is,exp}$ defined as follows :

$$\eta_{is,exp} \triangleq \frac{h_C - h_D}{h_C - h_{D,s}} \quad (3.11)$$

where $h_{D,s}$ is the enthalpy of the fluid after expansion and $h_{D,s}$ is the enthalpy that the fluid would have if the expansion was isentropic.

Transformation D-A : Heat rejection

During the heat rejection process, the fluid is first desuperheated from a temperature T_D to the saturation temperature $T_{A'}$; the fluid is then condensed before being subcooled

from $T_{A''}$ to $T_A = T_{A''} - \Delta T_{sub,ORC}$, reducing the risk of cavitation of the fluid in the feed-pump.

$$T_D \rightarrow T_{A'} \quad h_{A'} \rightarrow h_{A''} \quad T_{A''} \rightarrow T_A \quad (3.12)$$

The total amount of heat rejected to the heat sink $q_{si,ORC}$ is defined as in Equation (3.13) :

$$q_{si,ORC} \triangleq h_D - h_A \quad (3.13)$$

Performance of the cycle

The ORC is used to convert back heat into electricity by discharging heat from a high temperature heat source to a low temperature heat sink. The thermal efficiency η_{th} captures the performance of the thermal cycle, namely how much electricity $w_{out} := w_{exp} - w_{pump}$ is produced per unit of heat provided by at heat source $q_{so,ORC}$, as defined in Equation (3.14) and is hence always smaller than unity :

$$\eta_{th} = \frac{w_{out}}{q_{so,ORC}} = \frac{w_{out}}{w_{out} + q_{si,ORC}} \quad (3.14)$$

Feed-pump		Heat extraction	
Isentropic efficiency [-]	$\eta_{is,pump}$	Superheating [K]	$\Delta T_{sup,ORC}$
Expander		Heat Rejection	
Isentropic efficiency [-]	$\eta_{is,exp}$	Subcooling [K]	$\Delta T_{sub,ORC}$

Table 3.3: ORC cycle-related parameters

3.1.4 Heat Exchangers modelling

The CB overall thermodynamic cycle is thus composed of the cascading of the three cycles described previously (thus meaning that the TES is actually used as the HP heat sink and the ORC heat source, reducing the total number of parameters of the CB). Those three cycles interact with each other and with the HP cycle heat source and the ORC through HEX, which needs to be described to be able to characterize the simulation of a Rankine-based CB.

During a heat exchange, the hot fluid is always at a higher temperature than the cold fluid, by at least a temperature difference of ΔT_{pp} , which is referred to as the pinch-point temperature difference or **the pinch**. At the same time, fluids on both sides of the HEX undergo a pressure drop intrinsically linked to the heat exchange ; such a pressure drop Δp is often small but non negligible and affects the temperature of the fluids at the end of the phase change (as the saturation temperature is linked to the pressure).

Last, the HP heat source and the ORC heat sink can be characterized by their initial temperature $T_{source/sink}$ and their temperature difference over the HEX ΔT_{glide} , which is referred to as **the glide**.

3.2 Thermodynamic model used

Based on the models presented in Section 3.1 and on the parameters identified, a full operation of a Rankine-based CB can be modelled, considering the use of R1233zd(E), which has been identified several times as a high performing working fluid. As it has however been highlighted in the literature review, the performance of the CB is rather sensitive to some design parameters, namely the isentropic efficiencies used for the compressor, for the turbine and the feed-pump but also the subcooling in the feed-pump or the HEX design.

A brief discussion is hence held for each of those parameters on their impact on the performances and on how they can be modelled more thoroughly before presenting the off-design model considered for the rest of this thesis.

3.2.1 Compressor and Expander efficiency

Compressor and expander are usually modelled with constant isentropic efficiencies in the literature [10, 17, 4] for sake of simplicity of the model ; however, as it was already identified, the isentropic efficiency of a compressor or an expander is actually dependent on the pressure ratio and the temperature at which they are operated.

Lemort proposed in [39] a semi-empirical model for scroll compressors and expander, that was later extended to other volumetric machines in [38]. This type of machines offer relatively high pressure ratios for low rotational speeds and low flow rates while supporting two-phase flows, which can typically occur in expanders at the end of the expansion ; among them, screw expanders show state-of-the-art efficiencies of 84 [%] while having volumetric ratios lying going up to 8. Even though they show the largest electrical power among the volumetric machines, higher electrical powers than to 200 [kW_{el}] [3] might be needed, which can be obtained by using several volumetric machines in parallel. Using volumetric machines in series, as proposed by Kane et al. in with scroll [45], could be on the other side imagined to increase the overall volumetric ratio.

Using the full semi-empirical model proposed by Lemort is out of the scope of this work, as no existing volumetric machine was used and as the model initially aims at fitting real data with a mathematical model ; for this reason, Lemort's model is truncated as in [46], allowing to represent well enough volumetric machines' behavior with fewer parameters. The parameters used to model a volumetric machine are the volumetric ratio r_v , the maximum efficiency η_{mech} , the displacement volume V_s and the rotational speed N_s .

Thermodynamic model

The model can be used to represent the compression/expansion of a fluid from a pressure p_{su} to a pressure p_{exh} by decomposing the transformation in two phases, assumed adiabatic :

1. An isentropic volumetric compression/expansion, during which the fluid is compressed/expanded with a volumetric ratio r_v from the supply state (su) to the intermediary state (in).

$$v_{su} \rightarrow v_{in} \quad p_{su} \rightarrow p_{in} \quad s_{su} = s_{in} \quad (3.15)$$

The work related to the isentropic transformation w_1 is defined as follows :

$$w_1 = h_{in} - h_{su} \quad (3.16)$$

2. An isochoric pressure equalization, during which the intermediary fluid pressure p_{in} is equalized to the desired exhaust pressure p_{exh} ;

$$v_{in} = v_{exh} \quad p_{in} \rightarrow p_{exh} \quad s_{in} \rightarrow s_{exh} \quad (3.17)$$

The work related to the pressure equalization w_2 is defined as :

$$w_2 = v_{in} \cdot (p_{exh} - p_{in}) \quad (3.18)$$

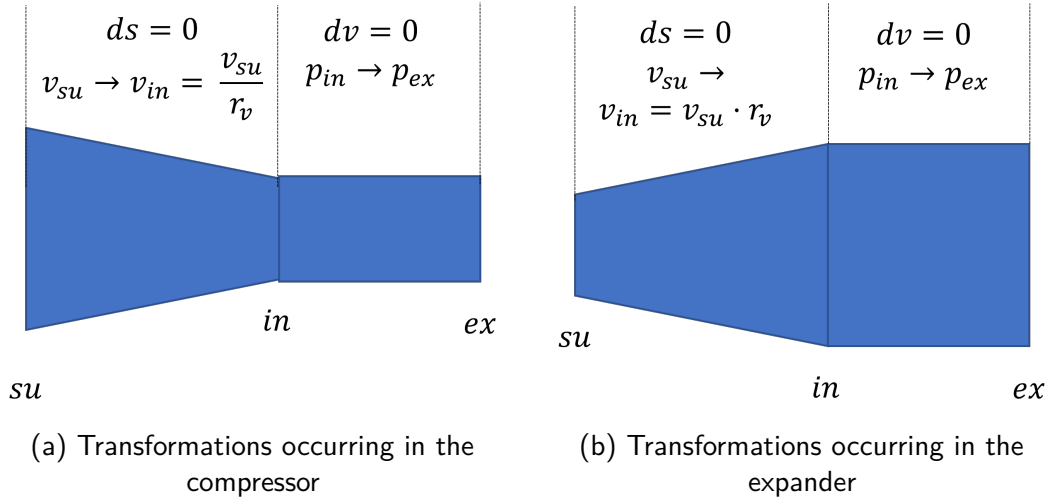


Figure 3.5: Transformations occurring in volumetric machines based on the model presented in [46]

The isentropic efficiency η_{is} is defined as before by the real work $w_{scroll} = w_1 + w_2$ and work that would have been provided/obtained if the transformation had been isentropic, considering a maximum achievable isentropic efficiency $\eta_{is,max}$:

$$\eta_{is,comp} = \frac{w_1 + w_2}{h_{ex} - h_{su}} \cdot \eta_{is,max} \quad \eta_{is,exp} = \frac{h_{ex} - h_{su}}{w_1 + w_2} \cdot \eta_{is,max} \quad (3.19)$$

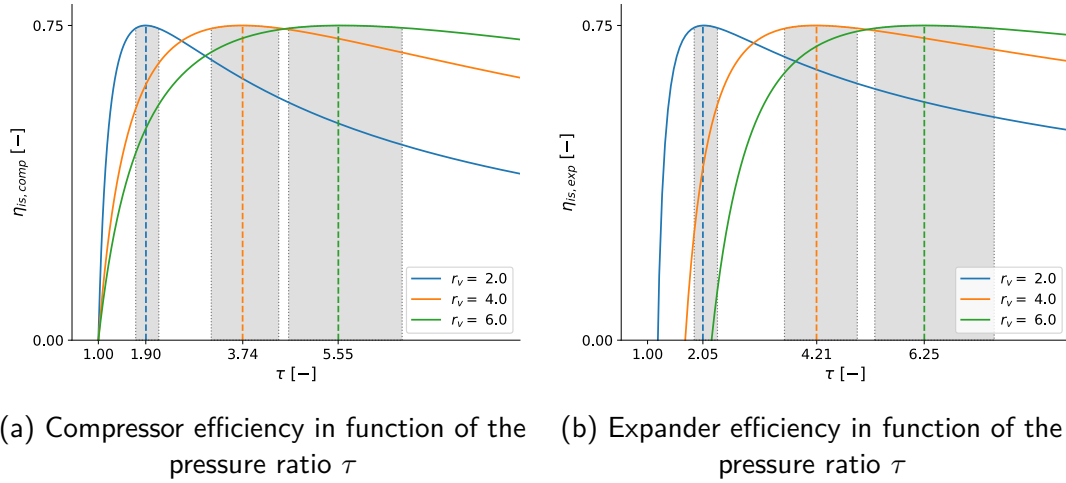


Figure 3.6: Lumped model for volumetric machines based on the model presented in [46], considering a maximum achievable efficiency $\eta_{is,max} = 75$ [%]. Shaded areas represent the pressure ratios for which $\eta_{is} \leq 0.99\eta_{is,max}$

The evolution of the isentropic efficiency in function of the pressure ratio τ between the supply and exhaust pressures is represented in Figure 3.6a for a compression and in Figure 3.6b for an expansion. The maxima of the curves are obtained for $\tau \approx r_v$ and are exactly located at $\tau = r_v$ if the fluid is a perfect gas.

The shaded areas in Figures 3.6a and 3.6b represent the pressure ratios for which the isentropic efficiency deviation from its maximal value is smaller than 1 [%]. It thus appear that small build-in volumetric ratios allow to obtain high performances at low pressure ratios only for a narrow range of pressure ratio close to the optimal pressure ratio ; on the other hand, larger build-in ratios perform offer good performances in a broader range of pressure ratios. Small build-in volumetric ratios are thus adapted for working machine designs where the pressure ratios are not expected to vary much over the operational range, while large volumetric ratio might be overall a better fit for highly fluctuating pressure ratios.

As the optimal choice of volumetric ratio (and thus the preferred behavior for an annual operation) cannot be determined beforehand, this parameter will be considered to be a design variable to determine along with the optimal design of the Rankine-based CB. The maximal efficiency, on the other hand is a design parameter, which only depends on the volumetric machine considered. A range of volumetric ratio between 1.5 and 5.5 are hence considered in this study, for a maximal isentropic efficiency of 75 [%].

3.2.2 Pump model

As already mentioned, even though the feed-pump isentropic efficiency and the feed-pump subcooling have been identified as having a significant impact on ORC, to the author's best knowledge no study evaluated the sensitivity of the P2P ratio on those parameters, fixed values being usually assumed without much discussion.

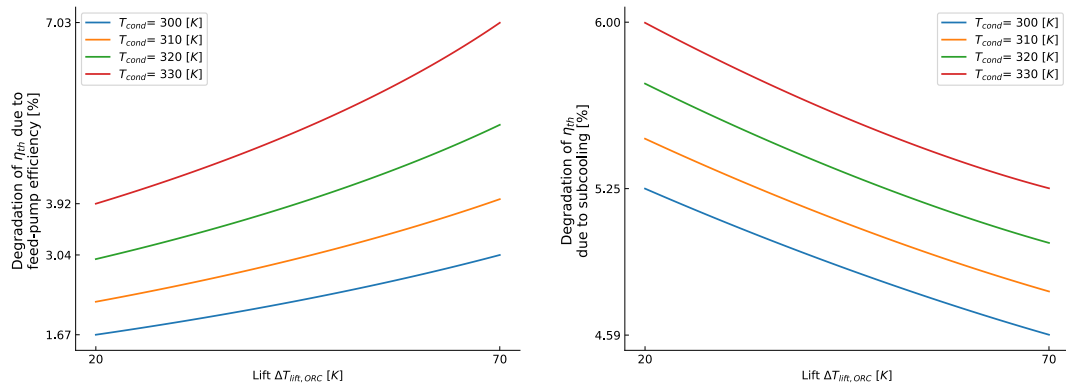
A sensitivity analysis on the feed-pump isentropic efficiency $\eta_{is,pump}$ and on the subcooling of the fluid before the feed-pump $\Delta T_{sub,ORC}$ was thus conducted by simulating an ORC using R1233zd(E) as the working fluid with CoolProp, a reference-grade open-source thermophysical property library [43] to evaluate the impact those parameters on the thermal efficiency. The thermal efficiency was evaluated for different condensing temperatures T_{cond} and different lifts $\Delta T_{lift,ORC}$ at the ORC.

This impact was evaluated for the feed-pump isentropic efficiency by the adimensional degradation of the thermal efficiency calculated as in Equation (3.20), considering a subcooling of 5 [K], a superheating $\Delta T_{sup,ORC} = 5$ [K]:

$$\%_{\eta_{is}}(T_{cond}, \Delta T_{lift}) \triangleq 1 - \frac{\eta_{th}(\eta_{is,pump} = 50\%, T_{cond}, \Delta T_{lift})}{\eta_{th}(\eta_{is,pump} = 80\%, T_{cond}, \Delta T_{lift})} \quad (3.20)$$

For the subcooling, this impact was evaluated in the same way, considering the thermal efficiency for the highest and lowest subcooling as in Equation (3.21), considering an isentropic efficiency of 50 [%] and a superheating $\Delta T_{sup,ORC} = 5$ [K]:

$$\%_{\Delta T_{sub}}(T_{cond}, \Delta T_{lift}) \triangleq 1 - \frac{\eta_{th}(\Delta T_{sub} = 5 \text{ [K]}, T_{cond}, \Delta T_{lift})}{\eta_{th}(\Delta T_{sub} = 15 \text{ [K]}, T_{cond}, \Delta T_{lift})} \quad (3.21)$$



(a) Impact on η_{th} of using $\eta_{is,pump} = 50$ [%] instead of $\eta_{is,pump} = 80$ [%] (b) Impact on η_{th} of using $\Delta T_{sub} = 5$ [K] instead of $\Delta T_{sub} = 15$ [K]

Figure 3.7: Sensitivity analysis of the impact of the feed-pump characteristics on the thermal efficiency of the ORC

It appears that the impact of the subcooling on the thermal efficiency show a small dependence on the condensing temperature and on the lift considered. The degradation in the thermal efficiency obtained by using a subcooling of 15 [K] instead of a subcooling of 5 [K] is close to 5 [%], this figure approximately increasing by 0.2 additional percent per 10 [K] increase in the condensing temperature. On the same time considering a 20 [K] glide leads to an degradation of the thermal efficiency higher by 0.6 additional points than when considering a 70 [K] glide, almost independently from the condensation temperature.

Concerning the feed-pump isentropic efficiency, the impact is more marked for high condensing temperatures and increases with the lift. The thermal efficiency is 7 [%] lower when considering a $\eta_{is,pump} = 50$ [%] instead of $\eta_{is,pump} = 80$ [%] for $T_{cond} = 330$ [K] and $\Delta T_{lift} = 70$ [K] ; for the same lift but a condensing temperature $T_{cond} = 300$ [K], the degradation of the thermal efficiency is of only 3 [%].

It thus appears that **the subcooling must be minimized as much as possible**, as it has a non-negligible impact on the thermal efficiency ; it must nonetheless be stressed that the actual minimal value for the subcooling depends on the component chosen and the system operation. For this reason, without prior knowledge on the component used and the thermodynamic states of the ORC, the subcooling has been imposed as a design parameter ; this is nonetheless done without a significant degradation of the cycle performances. Based on minimal values for the subcooling that could be found in literature [42], a subcooling $\Delta T_{sub,ORC} = 5$ [K] is chosen, keeping in mind that this value might lead to an 6 [%] overestimation of the ORC thermal efficiency if a small lift and high condensing temperature are observed in the ORC.

Concerning the feed-pump isentropic efficiency, **its impact on the thermal efficiency is mild at low condensing temperatures**. A semi-empirical model for feed-pump isentropic model was actually proposed by Landelle et al. in [40], taking into account the pressure differential but also the shaft speed and the electromechanical losses. However, this model was proposed for [kW] scale feed-pumps and as the aim of this thesis is to investigate the potential of integration of CB in energy system without prior knowledge on the size of the CB (and thus of the feed-pump), there is a risk of using this model out of the defined scope. A feed-pump isentropic efficiency of $\eta_{is,pump} = 50$ [%] is thus considered as a conservative estimate, compared to the values used in most of the studies reviewed which often use values between 50 and 70 [%] [16, 17, 23, 7, 11, 29], as the investment burden of buying a highly performing feed-pump might exceed the obtained thermodynamic benefits.

3.2.3 Heat exchangers model

Pinch-point temperature differences have been shown by Dumont et al. [11] to be one of the main drivers of the performances of a CB, and that values as small as possible must be used. The impact of pressure losses on the other hand has not been assessed yet in a clear way ; even though, high pressures losses are expected to decrease the P2P ratio as they induce higher compression work and lower expansion work overall. As this parameter is usually design- and operation-dependent and that no technology of HEX was chosen beforehand, fixed values were assumed for all of the HEX.

Pressure losses were thus considered as fixed over all the different HEX and were taken equal to 0.5 [bar] as in [32] whereas pinch-point temperature differences of 2 [°C] were considered as in [11].

3.3 Performance Mapping

Based on the general model presented in Section 3.1 and taking into account the impact of some parameters discussed in Section 3.2, an evaluation of the CB performances can be performed. The numerical model used is presented in Appendix A.1. The values for the fixed parameters used in the basecase are presented in Table 3.4 while the parameters investigated for the sensitivity analyses are presented in Table 3.5.

Fixed parameters

The CB model uses a lot of different parameters ; some of them are to be optimized in any case as they are non-idealities that degrade the overall performances in a very clear trend. Such design parameters, as for example pressure drops, pinch-point temperature differences in the HEX or the subcooling of the ORC feed-pump would ideally be taken as small as possible from a thermodynamic performance point of view ; others such as the compressor, expander and pump efficiencies, the HP subcooling or the superheating in the HP and the ORC should be as high as possible. Optimizing those parameters can however only be improving the design of the related components, leading to higher costs, even though non-idealities cannot be avoided.

Build-in volumetric ratios, on the other hand, are parameters whose optimal value depends on the cycle characteristics (mainly on the pressure ratios actually). As already explained in Section 3.2.1, small build-in volumetric ratios are best fitted for compressions or expansions with rather constant and small pressure ratios, while large build-in volumetric ratios can deal with broader ranges of pressure ratios without significant degradation of the isentropic efficiency. For this reason, and as the determination of the optimal values of the build-in volumetric ratios is to be done during the techno-economic optimization, fixed values will be taken for the sensitivity analyzes, considering intermediate build-in ratios.

The values considered for the afore-mentioned fixed parameters for the sensitivity analyses are given in Table 3.4.

$\eta_{is,max}$ [—]	0.75	$\eta_{is,pump}$ [—]	0.5
$r_{v,HP}$ [—]	2.5	$r_{v,ORC}$ [—]	2.5
$\Delta T_{pp,so,ORC}$ [K]	2	$\Delta T_{pp,so,ORC}$ [K]	2
$\Delta T_{pp,so,ORC}$ [K]	2	$\Delta T_{pp,so,ORC}$ [K]	2
$\Delta p_{so,HP}$ [bar]	0.5	$\Delta p_{si,HP}$ [bar]	0.5
$\Delta p_{so,ORC}$ [bar]	0.5	$\Delta p_{si,ORC}$ [bar]	0.5
$\Delta T_{sup,HP}$ [K]	5	$\Delta T_{sup,ORC}$ [K]	5
$\Delta T_{sub,HP}$ [K]	5	$\Delta T_{sub,ORC}$ [K]	5
$\Delta T_{glide,si,ORC}$ [K]	10		

Table 3.4: Fixed design parameters

Parameters of interest

At the same time, parameters such as the source and sink temperatures do not rely on a design choice but are mostly imposed, so that, even if the outcome of a sensitivity analysis on their values is predictable, exploring their impact on the performances allows to understand the range of performances that can be obtained by the CB in different cases.

From a more operational point of view, if minimizing the glide at the heat source to reduce the temperature difference between the evaporating temperature of HP cycle and the heat source temperature improves the system performances (by reducing the compression work), maximizing it leads to a higher utilization of the Waste Heat source, which can allow to increase the power flows. A sensitivity analysis on the glide might thus help to understand which trade-offs appear between the overall cycle efficiency and the use of the Waste Heat source.

Last, by impacting the thermodynamic performances of both the charging and the discharging cycle but also conditioning the amount of energy that can be stored by the CB, the spread in the TES or the storage temperature are parameters for which conducting sensitivity analyses might be pretty interesting to already grasp the trade-offs that can exist between storing large amounts of energy and storing energy efficiently.

T_{sink}	T_{source}	$\Delta T_{glide,so,HP}$	T_{sto}	$\Delta T_{spread,sto}$
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Table 3.5: CB parameters studied throughout the sensitivity analyzes

3.3.1 Heat source temperature

A sensitivity analysis is here conducted on the impact of the heat source temperature T_{source} on the COP of the HP for different storage temperatures. As the aim of this sensitivity analysis is to highlight the role of the heat source temperature, the heat sink temperature (which does not impact the HP cycle) is kept at a temperature of 10 [°C], the storage spread (representing how much energy can be stored in the TES) is fixed to 10 [°C], and the heat source glide (determining how much the waste heat source is used) is fixed to 10 [°C].

Results

The results of the sensitivity analysis are shown in Figure 3.8. As one could expect from the expression of an ideal Carnot Heat Pump cycle, decreasing the temperature difference between the heat source and the heat sink improves the COP and the maximum values of COP are always obtained for the lowest feasible storage temperature, as it was highlighted by Dumont et al. in [11]. Given the low glide at the heat source (and, thus the low use of the Waste Heat source) and the low pinches in the HEX, relatively high values of P2P ratio, comparable to the ones that can be found in the literature, can be obtained. It must be noted that, as the glide at the source is taken relatively low, some configurations with low storage temperatures and high source temperatures (and thus

high VCHP evaporating temperatures) are not allowed.

3.3.2 Heat source glide

If the system performances presented in Figure 3.8b could be even further improved at high heat source temperatures by choosing lower build-in volumetric ratio for the HP, the amount of electricity used to charge the VCHP would still be very low with respect to heat extracted from the waste heat source. In that sense, high-COP CB extracting heat from waste heat source will be limited by the heat availability ; the economic interest of using waste heat with CB using a low glide at the heat source becomes thus limited if electricity production using waste heat recovery ORC is possible, as the improvement on the thermal efficiency provided by the VCHP cycle is counterbalanced by the low use of the heat source.

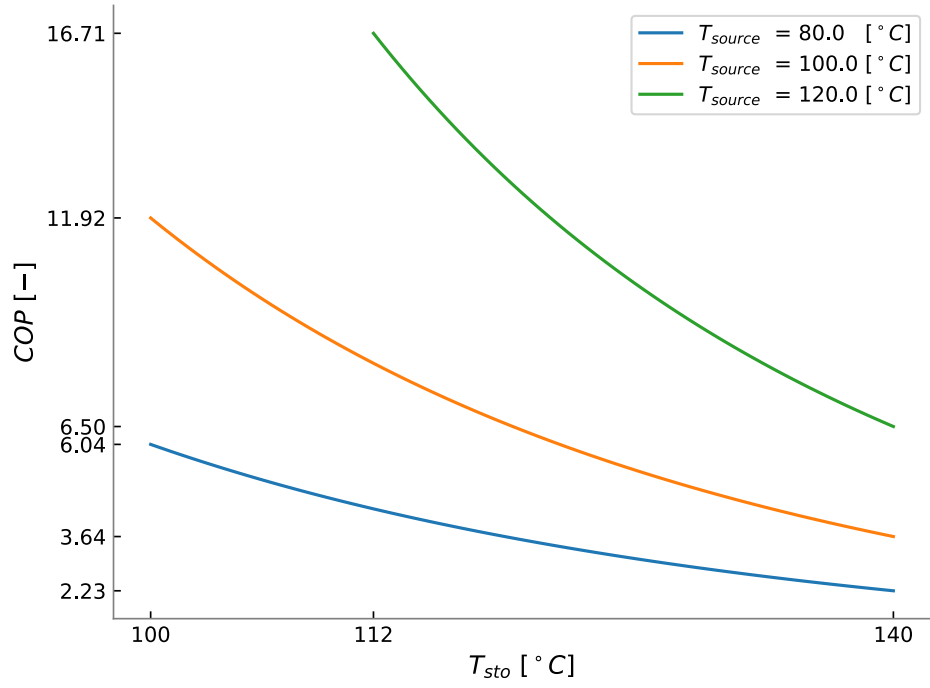
A higher use of the Waste Heat source might thus be needed to efficiently use the waste heat source from an economic point of view ; but as a larger heat exploitation means a larger glide (since the amount of heat extracted from the heat source is approximately proportional to the glide of the heat source), the HP performances might be reduced. A sensitivity analysis is hence conducted on the impact of the glide at the heat source on the COP for different storage temperatures.

The spread at the storage is kept equal to 10 [°C] and the heat sink is kept equal to 10 [°C] while the higher heat source temperature (namely $T_{source} = 120$ [°C]) is considered. The latter choice is made because, as explained in 4.2, a 120 [°C] waste heat source is expected to be available in the flue gases of the biomass powerplant which will sell green heat to the UCLouvain.

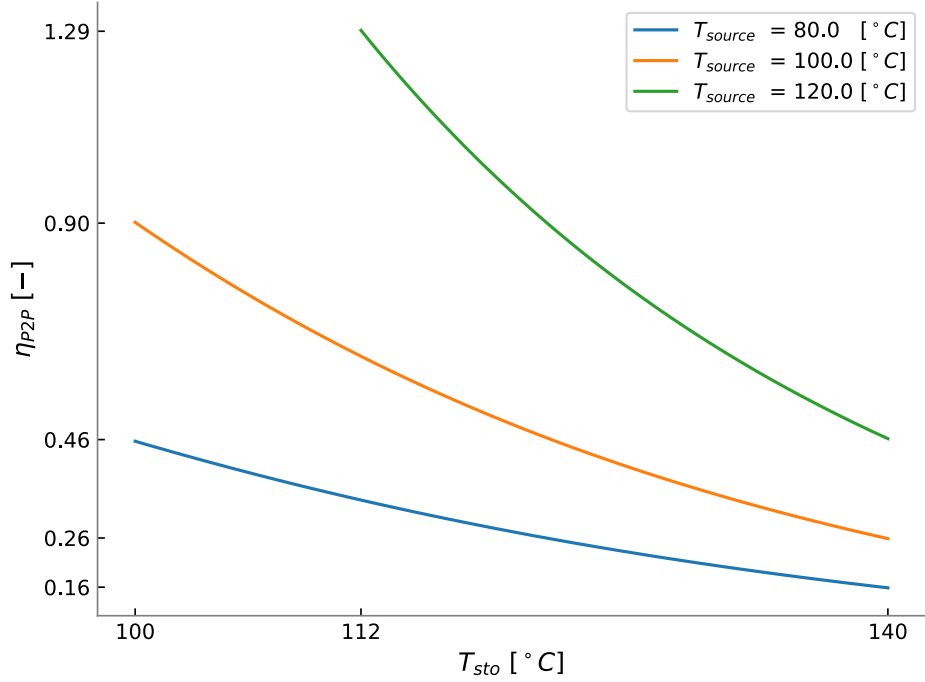
Results

The results of the sensitivity analysis are shown in Figure 3.9. Without any surprise, the impact of a larger glide is more marked for designs with lower storage temperature, whose COP is decreased sixfold ; the waste heat use, however is about twelve times higher, if the heat extracted from the waste heat source can be estimated to be proportional to the temperature decrease of the source. In others words, **by trading the thermodynamic efficiency with a higher use of the waste heat source, the maximum discharging power can be doubled**; the increase in the maximum discharging power is even higher for $T_{sto} = 140$ [°C], as the P2P ratio is only decreased by a factor 3.3.

If from an energetic point of view this conclusion might be outrageous, it might be good news from an economic point of view for some energy systems suffering from low feed-in tariffs and high electricity prices ; an increased CB charging capacity means a higher input energy flux, so that the smaller storage efficiency is balanced by a higher amount of energy discharged from it than for a highly efficient design providing negligible amounts of energy. Such a conclusion, must be verified for the energy system considered throughout the thermo-economic optimization performed in Chapter 6.



(a) The COP reaches higher values for higher heat source temperatures and lower storage temperatures.



(b) The P2P ratio is dominated by the COP behavior ; the values obtained are close to the ones obtained in the literature [12, 17]

Figure 3.8: Impact of the heat source and storage temperatures on the CB performances

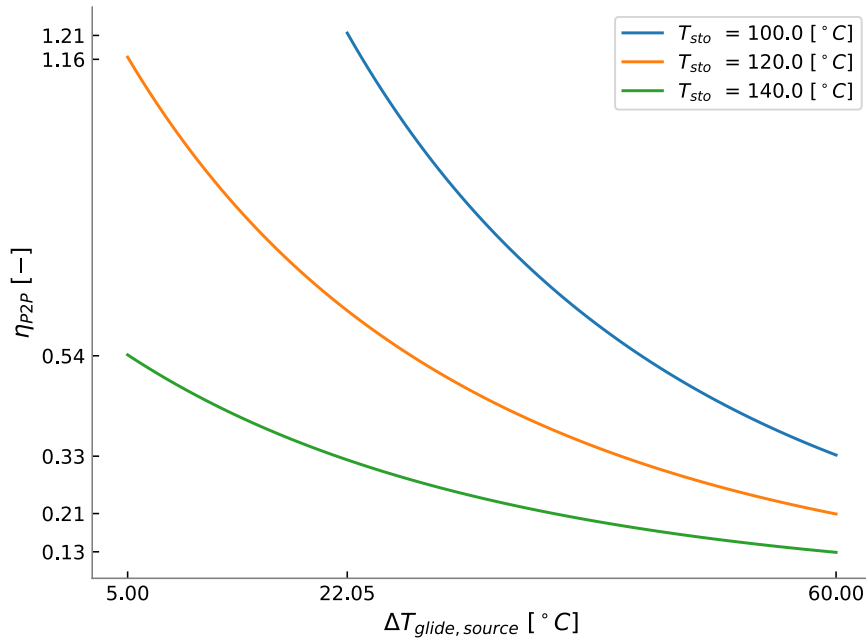


Figure 3.9: P2P ratio evolution with the glide at the VCHP heat source

Once again, some configurations with low glides and low storage temperatures are not allowed as the storage temperature would be inferior to the VCHP evaporating temperature.

3.3.3 Sink temperature

At the same time, the impact of the sink temperature on the thermal efficiency must be assessed as, except if a fixed- and low-temperature heat demand can be determined, the heat sink temperature for the ORC varies over the year as it is the case for each of the two integration scenarios determined in Section 6.

Based on those two integration scenarios, a sensitivity analysis on η_{th} is performed for low temperature heat sinks (namely considering the range of value for the ambient temperature in Louvain-la-Neuve over the year) and another one is performed for high-temperature high sinks (namely considering the range of return temperatures of the DHN over the year). The storage spread is fixed to 10 [K] and the heat source temperature is fixed to 120 [°C]. A higher heat source glide of 25 [°C] is however considered in order to consider an operation of the VCHP cycle on a wider range of storage temperatures. The trend on the P2P ratio is however not affected, it has been identified that the COP decreases monotonously with the glide at the VCHP heat source and that the degradation of the COP over the considered storage range is much more important than the variation of the thermal efficiency on the same range.

Results

The results of the sensitivity analysis are shown in Figures 3.10 and 3.11. It appears that for low heat sink temperatures, the impact of the heat sink temperature on the P2P ratio is marked for low storage temperatures and negligible otherwise ; the P2P ratio for $T_{sto} = 100$ [°C] is however almost twice as big as the P2P ratio for $T_{sto} = 120$ [°C] and between 3 times and 4 times higher than when using $T_{to} = 140$ [°C]. As the temperature lift is rather important, higher values of volumetric ratios than 2.5 might be preferred to improve the P2P ratio ; this is however not a strict necessity as expanders with large r_v have shown to perform well on larger pressure ratio ranges than expanders with small r_v for the expander model used.

The impact of the heat sink temperature on the P2P ratio is however much higher for heat sink temperatures in the range of the DHN return temperatures. Low temperature storage designs does not seem adapted to a high variability in the heat sink temperatures, while higher storage temperature design show a milder yet significant sensitivity to this parameter.

The steep decrease in the thermal efficiency of low storage temperature designs can be partly explained by the high reduction in the lift of the ORC leading to low pressure ratios which are not adapted anymore to the volumetric ratio chosen. Choosing $r_{v,ORC} = 1.5$ [–] for $T_{sto} = 100$ [°C] leads to a drop of the thermal efficiency close to zero for a heat sink temperature of 70 [°C] instead of 63.4 [°C] for example.

It thus appears that for a scenario where the heat sink temperature varies significantly and is high, intermediate storage temperatures seem to be the best guess to take as they show a mild dependency on the heat sink temperature for a given volumetric ratio while showing good performances. Low volumetric ratio must also be preferred, as the generally low pressure ratio drastically impacts the expander efficiency.

For low heat sink temperatures however, the decrease in performances is much less important, and low storage temperatures design achieve much better performances and high volumetric ratios should be preferred due to the higher pressure ratios obtained.

3.3.4 Storage spread

The impact of the storage spread has already been identified by Dumont et al. as a parameter asking for trade-offs [11]: the higher the storage spread, the higher the energy density of the storage for a given mass of material but the lower the P2P ratio, and vice-versa. However, to the difference of this study, the subcooling is chosen as a fixed parameter in this model, which might lead to different conclusions. A sensitivity analysis is hence conducted for different storage temperatures, considering $T_{so,HP} = 120$ [°C] and a moderate glide $\Delta T_{glide,so,HP} = 25$ [K], for a low heat sink temperature $T_{si,ORC} = 10$ [°C] and for a high heat sink temperature $T_{si,ORC} = 60$ [°C]. It must be noted the minimum storage temperature considered for pressurized tanks is 90 [°C] so $T_{sto} - \Delta T_{spread,sto} > 90$ [°C].

Results

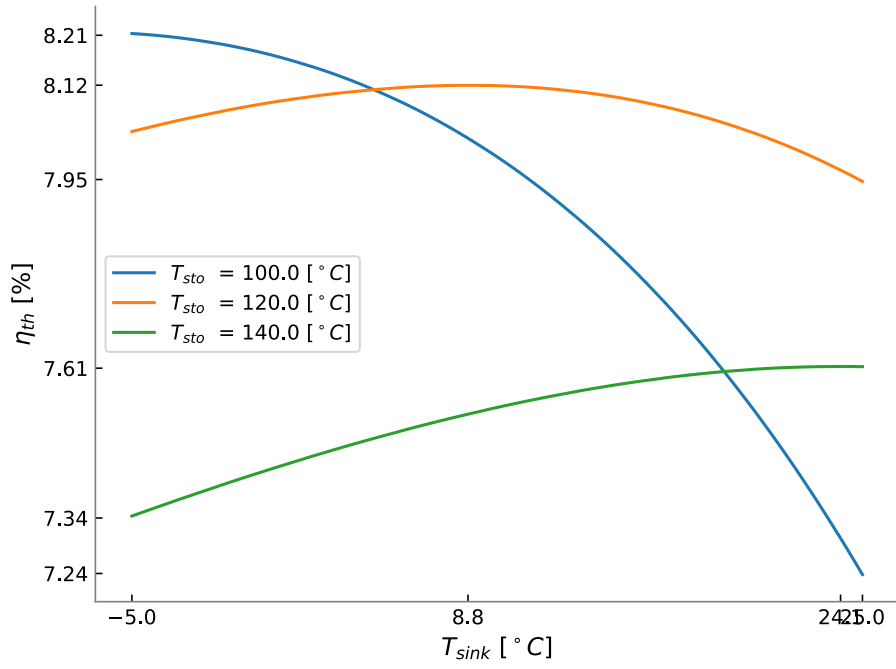
The results of the sensitivity analysis are shown in Figures 3.12 and 3.13. Assessing the impact of the storage spread on the P2P ratio is rather complex as it impacts both the VCHP and the ORC cycles by mainly impacting the lift in the cycle, even though not in the same fashion. In both cases, however, the major part of the heat exchanged comes from the latent heat h_{lv} due to the phase change of the working fluid.

As the heat exchange at the VCHP is mainly done by the condensation of the working fluid, the pinch-point is generally located at the close to the vapor saturation point ; below a certain threshold (mainly depending on the spread and the subcooling), the pinch point is located after the subcooling of the fluid so that increasing the spread leads to a direct decrease in the condensing temperature. Past this threshold increasing the large storage spread does not have much more impact on the condensing temperature.

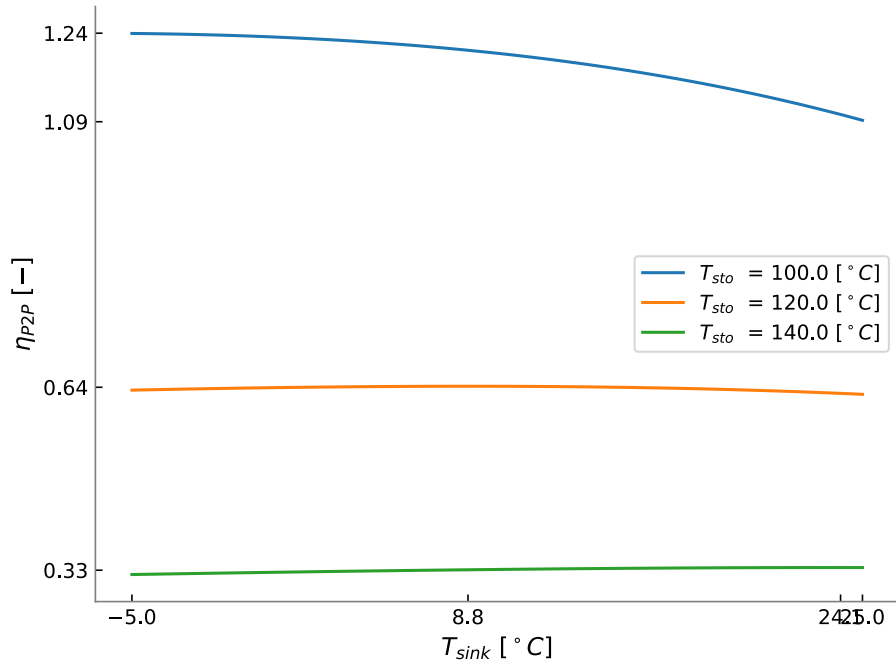
This mild impact is nonetheless beneficial and can be explained based on the iterative procedure used to locate the pinch-point in the HEX described in Appendix 3 ; as the pinch point is located at the vapor saturated point and that $h_{lv}/q_{so,HP}$ remains approximately constant, the difference between the storage temperature and the pinch-point temperature will be mainly proportional to the spread. This lead to a slight improvement of the COP when increasing the spread.

An opposite trend is observed for the ORC, as the pinch is located after the superheating of the vapor for low superheating temperatures and low spread values ; past a threshold, the pinch point is located at the liquid saturated point and increasing the spread imposes a lower evaporation temperature.

Additionally, the change in the lift in both cycles leads to a change in the pressure ratio, which can lead to inadequate volumetric ratio choice hence impacting the real isentropic efficiencies. This explains why for high storage temperatures and $r_{v,ORC} = 2.5$ a lower evaporation temperature is beneficial in the ORC ; the reduction in the pressure ratio actually allows a higher expander work relatively to the heat extracted from the storage, up to a certain point.

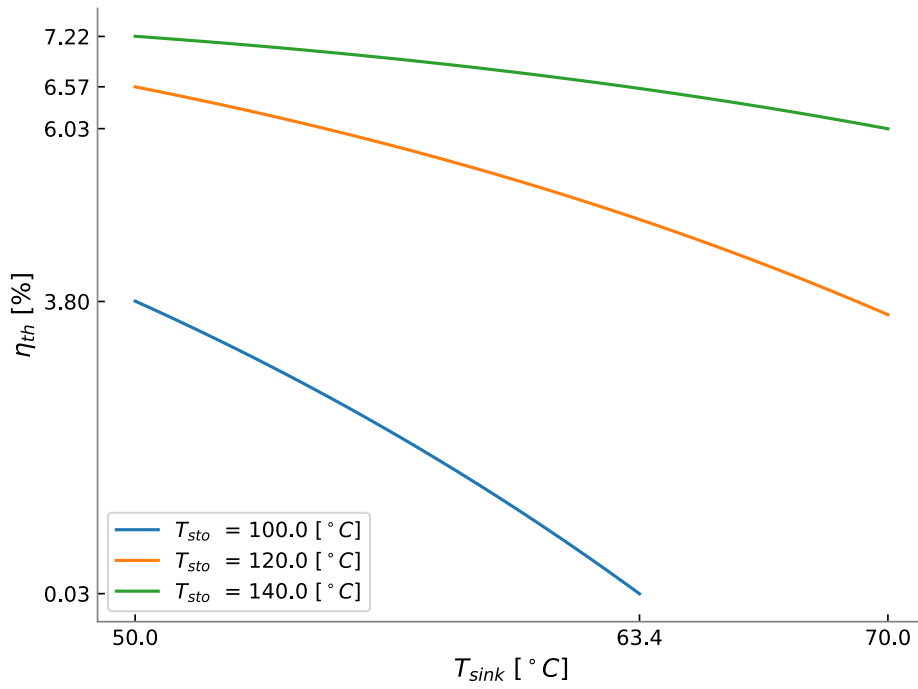


- (a) The maximum value of η_{th} is obtained for larger $T_{si,ORC}$ when considering high storage temperatures, due to the variation in the pressure ratio impacting $\eta_{is,pump}$ through the volumetric ratio $r_{v,ORC}$ chosen

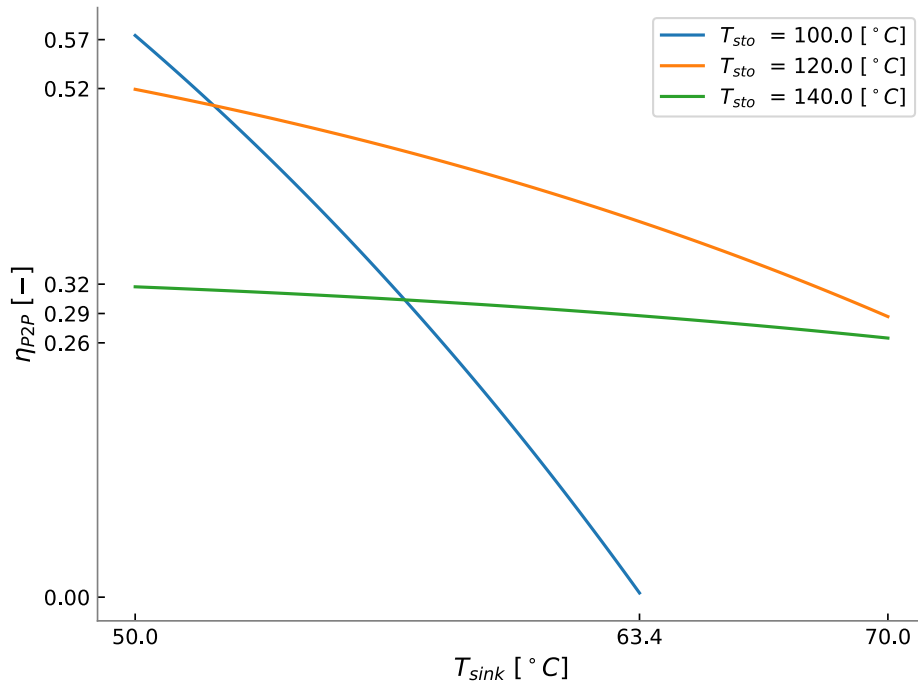


- (b) Overall, the **P2P ratio** of high storage temperature designs is not very sensitive to the increase in the heat sink temperature ; a mild impact is observed for low T_{sto}

Figure 3.10: Impact of the heat sink temperature on the **CB** performances for low heat sink temperatures

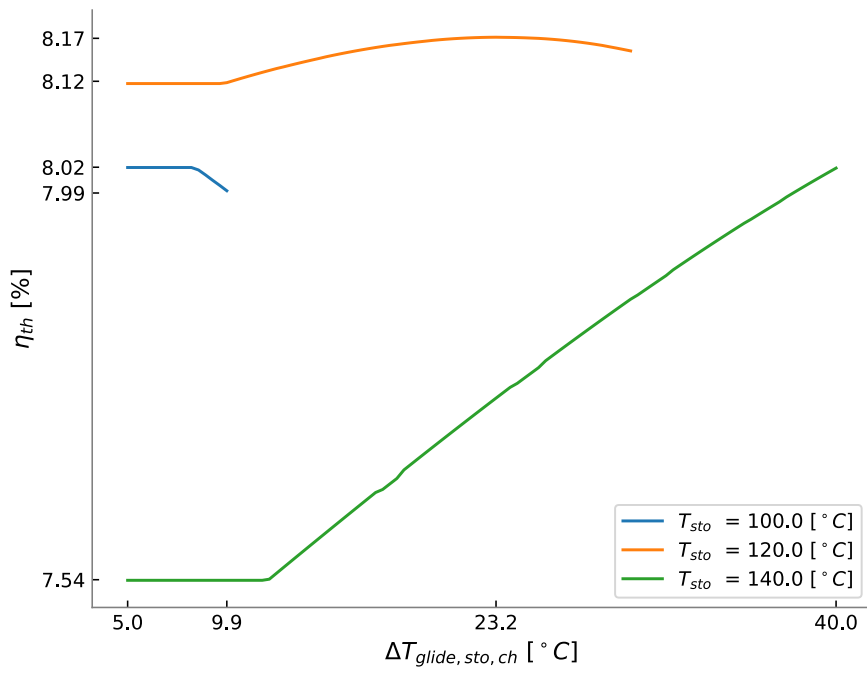


(a) Given the volumetric ratio, increasing the heat sink temperature leads to a degradation of the expander isentropic efficiency ; better η_{th} are obtained for high storage temperatures

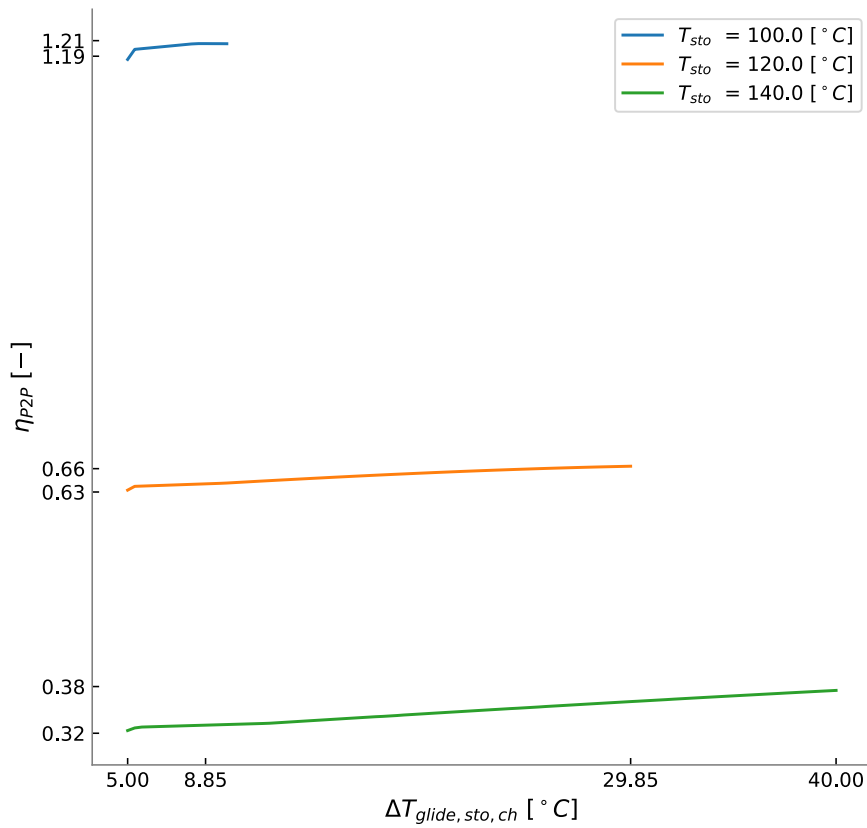


(b) Over the studied range of heat sinks, using low storage temperatures show to be particularly sensitive to $T_{si,ORC}$ even though performing better for low heat sink temperatures ; high storage temperatures do not reach high performances but show on the other hand rather constant performances.

Figure 3.11: Impact of the heat sink temperature on the CB performances for high heat sink temperatures

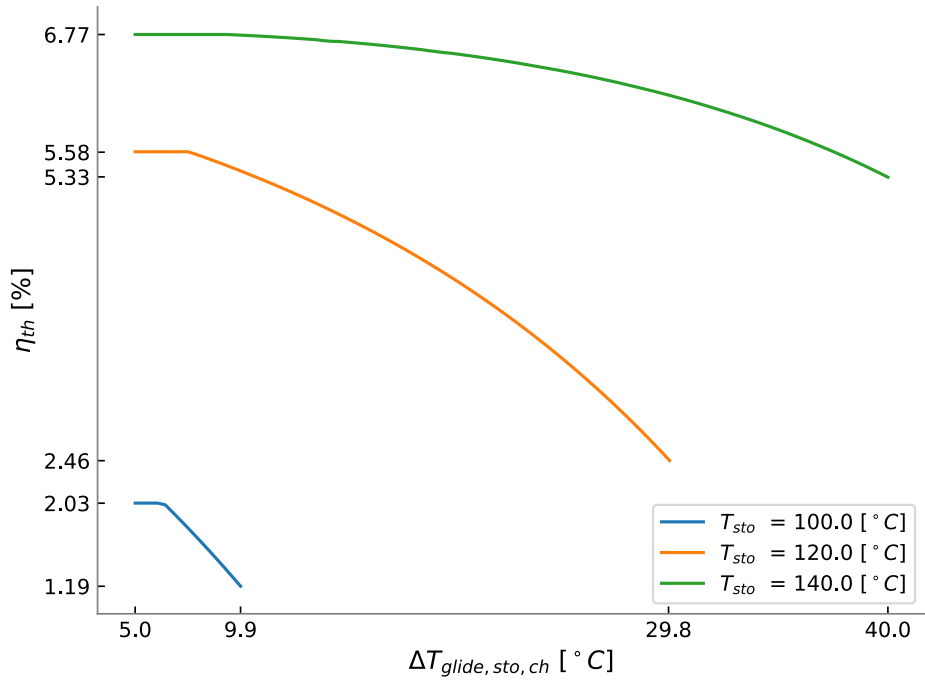


(a)

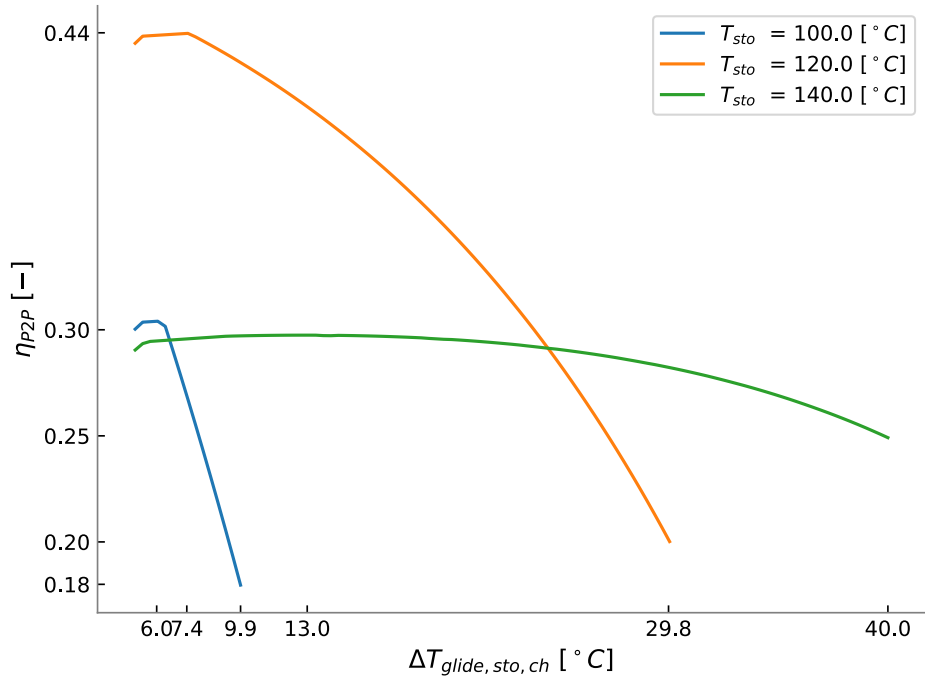


(b)

Figure 3.12: An optimum **P2P ratio** can be obtained for a spread leading to a pinch-point located at the vapor saturated point in the **VCHP** and a pinch-point located at the exit of the vapor superheating in the **ORC**.



(a)



(b)

Figure 3.13: An optimum **P2P ratio** can be obtained for a spread leading to a pinch-point located at the vapor saturated point in the **VCHP** and a pinch-point located at the exit of the vapor superheating in the **ORC**.

Chapter 4

Energy system model

“ In framing a system which we wish to last for ages, we should not lose sight of the changes which ages will produce. ”

James Madison, 1787, *U.S. Constitutional Convention*

In-line with its 2035 carbon neutral goal and due to the upcoming end of life of the natural gas **CHP** units currently used, UCLouvain plans to **change** its **energy system** by increasing the use of renewable and low-carbon energy sources in its energy system as shown in Figure 4.1

- A Biomass **CHP** powerplant will be operated by Veolia, using wood residues from the recycling parks of the neighbourhood of Louvain-la-Neuve and designed to meet the heat demand of UCLouvain. The **CHP** powerplant thus meets the requirement of UCLouvain for green heat while being able to supply low-carbon electricity ;
- At the same time, the volatility in the electricity prices favors energy systems to be more self-sufficient. The installation of **PV panels** and **WT**, producing electricity with near-zero carbon content, is thus considered by UCLouvain to reduce its dependency to the grid. Furthermore, the renewable electricity overproduction can be re-injected into the grid at a defined feed-in tariff.

Organisation of the chapter

As a good knowledge of the energy system in which to integrate the storage system is needed to determine which integration scenarios for the storage presented in Chapter 3 are worth to explore, a quick analysis of the system demand and infrastructures is performed in Section 4.1.

A technological, economic and environmental characterisation is then presented for each of the different production units that will be considered in the energy system in Sections 4.2, 4.3 and 4.4.

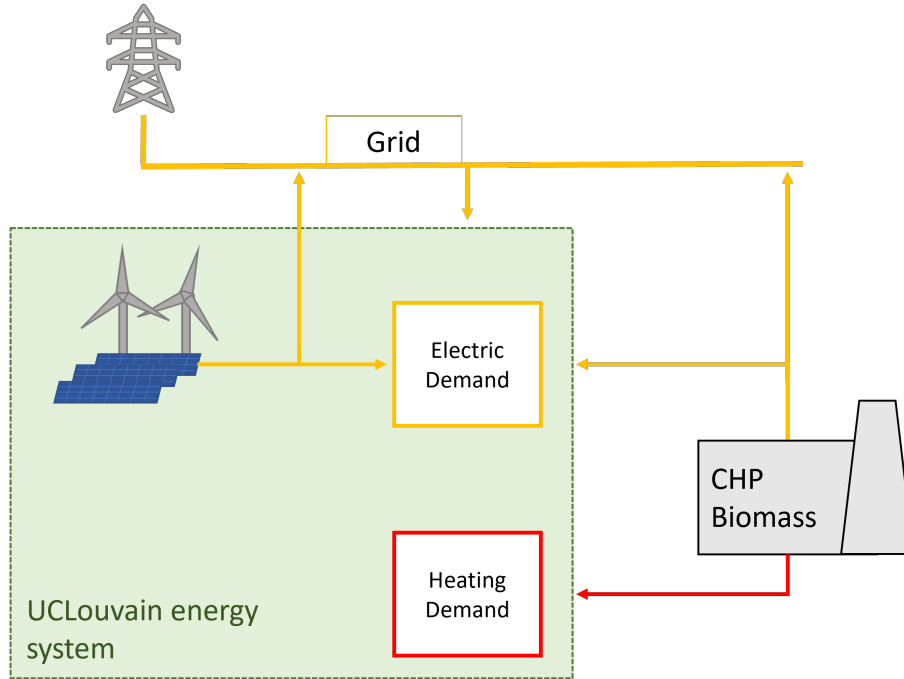


Figure 4.1: The energy system considered for UCLouvain is composed of PV panels and WT, and relies on a CHP biomass powerplant owned by Veolia and on the grid to match the electric demand for each hour. The heat production is ensured by the CHP powerplant, designed to match the heat demand of UCLouvain.

Last, a preliminary multi-objective optimisation of UCLouvain energy system without storage is done in Section 4.5 to grasp the system's behavior and determine how the Carnot Battery could be used best in this energy system.

4.1 Demand and infrastructures

UCLouvain energy system is composed of the academic buildings but also student accommodations, and commercial buildings, and can be characterized by its heat demand and electricity demand.

4.1.1 Heat demand

Heat is distributed in the energy system through a 12 km-long DHN at intermediate temperatures (50 to 80 [°C]). As described in [47], the heating of the heat transfer fluid is centralized in a heating station located in the higher part of Louvain-la-Neuve and the DHN is subdivided in three main branches feeding different areas of the campus. In this work, the DHN go and return temperature retrieved data is considered as an imposed value ; distribution losses inside of the DHN accounting for 9.4 [%] for the actual heat demand are also included in the heat demand definition of UCLouvain energy system,

The heat demand covers the totality of the academics buildings' needs as well as a high share of the student-related buildings' needs (mostly accommodations but also

	Academic buildings	Student-related buildings	Commercial Buildings
Share of the total heat demand [%]	65	15	20

Table 4.1: Consumer share of UCLouvain heat demand [47]

buildings related to student animation). Some commercial buildings of Louvain-la-Neuve are also fed by the DHN ; the share of the heat demand for year 2015 for the different consumers are reported in Table 4.1 from [47].

The hourly heat demand profile of UCLouvain energy system in 2021 is shown in Figure 4.2a ; the annual demand was of 47.5 [GWh_{th}], while the mean hourly heat demand over the year is equal to 5.43 [MWh_{th}] for a peak demand equal to 24 [MWh_{th}]. The hourly ambient temperature profile for year 2021 is shown in Figure 4.2b.

As part of the heat demand corresponds to sanitary-purposed heating, the temperature must always stay above 50 [°C] everywhere in the network, thus imposing a minimal return temperature for the DHN. At the same time, the go temperature is fixed by a proportional law with respect to the ambient temperature¹; varying the mass flow rate of the five 450 [m³/h]-pumps is thus the main way to match the heat demand. The pumps' work provide a 1 [bar] increase in pressure while the static pressure in the DHN is comprised between 3 and 9 [bars] depending of the altitude.

The temperature profiles of the go and return temperatures of the DHN for year 2019 are shown in Figure 4.3 and statistical data on the temperatures are given in Table 4.2.

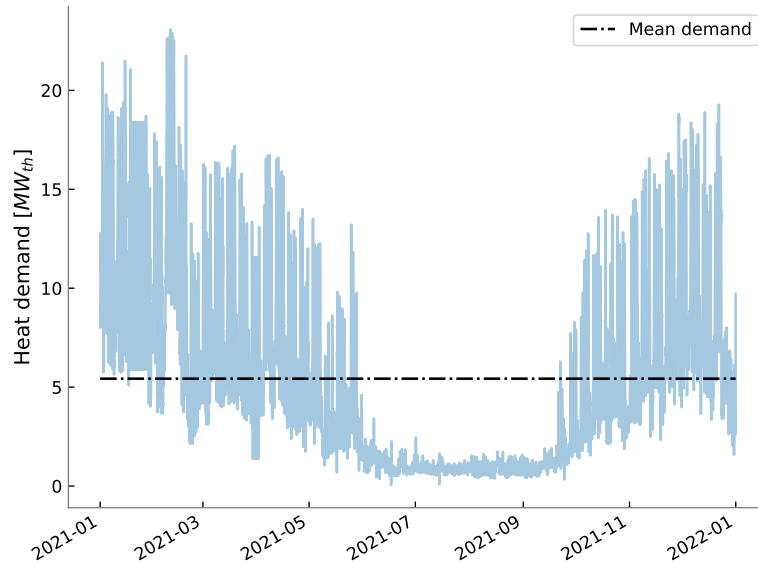
	Mean value μ [°C]	Standard deviation σ [°C]
Go temperature T_{go}	70.32	3.66
Return temperature T_{return}	57.79	1.96
Lift $\Delta T_{lift} \triangleq T_{go} - T_{return}$	12.54	4.53

Table 4.2: Statistical data on the DHN temperatures

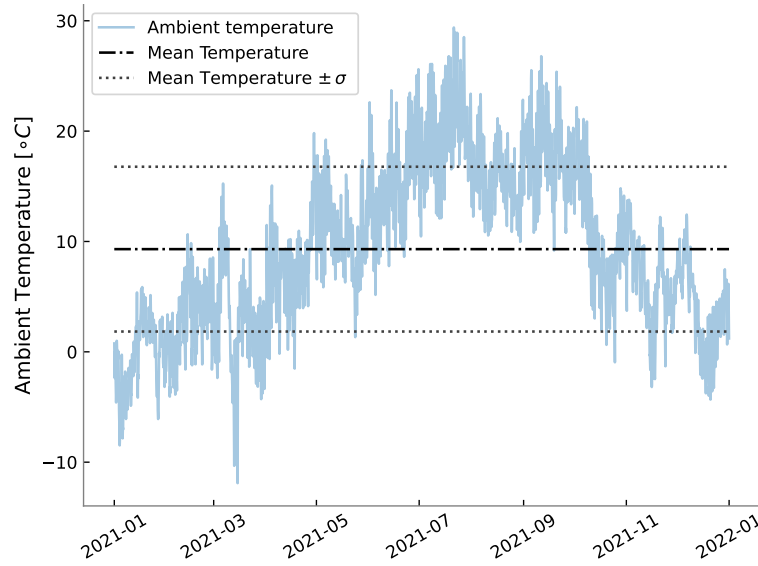
4.1.2 Electric demand

As for the heat demand, the electric demand covers all of the academic buildings and part of student-related and commercial buildings. The electricity is assumed to be transported around the campus with negligible losses and only active power is considered for the pricing and the demand.

¹It must nonetheless be mentioned that the solar irradiance has a significant impact on the heat demand of the buildings so that the actual go temperature is lower on a sunny day compared to a cloudier day, considering the same ambient temperature profile.



(a) Hourly heat demand profile for year 2021



(b) Hourly ambient temperature profile for year 2021

Figure 4.2: During summer months, the heat consumption is reduced to the sole heating of sanitary water as the ambient temperature is greater than the threshold temperature at which buildings start to be heated.

Shares for the different sectors could not be obtained as for the heat demand ; nonetheless, the use of some research equipment have a high impact on the electric load. Running experiments on the Cyclotron requires about 1 [MW_{el}] for example, so that it is usually ran during the weekend or during holidays when the electric demand is lower than during typical weeks.

The hourly electric demand profile for year 2021 is shown in Figure 4.4, for an annual

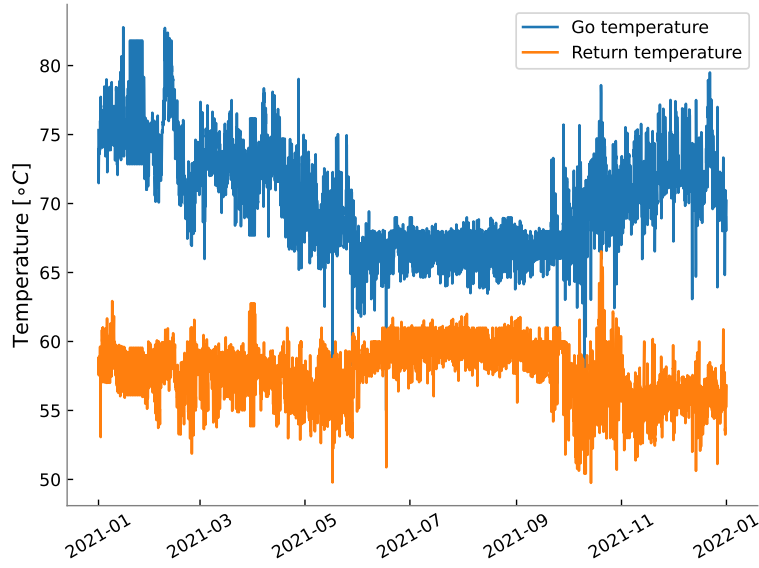


Figure 4.3: Temperature profile of the DHN go and return flows for year 2021

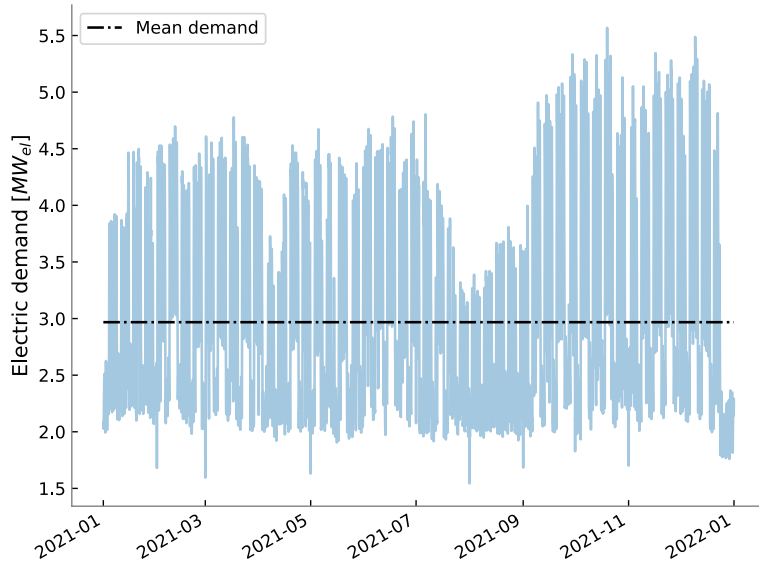


Figure 4.4: The electric demand is marked by a week-long periodic pattern. A typical 2 [MW_{el}] step in power requirement can be observed between weekdays and weekend days. Easter, summer and Christmas holidays also have a significant impact on the electric load.

demand of 26 [GWh_{el}], an average electric load of 2.97 [MW_{el}] and a peak demand of 5.57 [MW_{el}].

4.2 CHP unit

As mentioned before, UCLouvain aims to become carbon neutral by 2035 and thus looks forward green heat and electricity supply ; the operation of a biomass-run **CHP** powerplant by Veolia, designed to match the university's heat demand, is the option that was chosen by UCLouvain. Also, as the co-produced electricity has a lower carbon footprint than the grid, there is an opportunity for UCLouvain to reduce its electricity-related greenhouse emissions.

4.2.1 Technological overview

The **CHP** unit uses a high pressurized steam cycle. The steam flow exits the biomass boiler at approximately 400 [°C] and 80 [bars] and passes through a high-pressure turbine in which electricity only is produced. The steam flow then passes through a low-pressure turbine where part of the flow is extracted by a bleeding to produce heat while the other part is used to produce additional electricity. By reducing the heat demand (and thus the heat extraction), as it can be the case in summer time, the electricity production is thus increased as the mass-flow rate at the low-pressure turbine is increased.

Biomass boilers work most efficiently when they can run at nominal load [48] and are prone to higher smoke emissions and unstable combustion when working at part-load conditions [49], making them preferably used as baseline production units rather than load-follower ones. On top of this, as biomass combustion is considered as almost carbon neutral (see Section 4.2.3) and as electricity can either be sold UCLouvain energy system or exported to the grid, the **CHP** unit is operated by Veolia at full boiler duty all over the year, adapting the heating power (and thus the heat extraction at the low-pressure level) to the heat demand. Storage tanks are also considered to cover the peak demand that the powerplant cannot directly provide ; heat recovery from the flue gases is not considered by Veolia.

It must be added that technically additional distribution losses must be considered on top of the DHN ones, as the **CHP** powerplant will be located a few kilometers away from the heating station in Louvain-la-Neuve. Those losses are estimated to be about 2% of the total heat demand.

4.2.2 Technical characteristics

The **CHP** unit was designed based on the heat demand from UCLouvain energy system to be able to provide green heat at anytime of the year. The boiler combustion chamber has a capacity of 26.75 [MW] for a maximum heating power of 19 [MW_{th}] and a maximum net electric power of 7.98 [MW_{el}].

As the boiler chamber is used at its maximum heat duty all over the year, the electric and heat power are linked ; based on data provided by Fabian Servais, Head of Technics at Veolia [50], a polynomial logarithmic regression of the efficiencies with respect to the heating power of the **CHP** unit was performed in order to characterize the **CHP** behavior,

as shown in Figure 4.5.

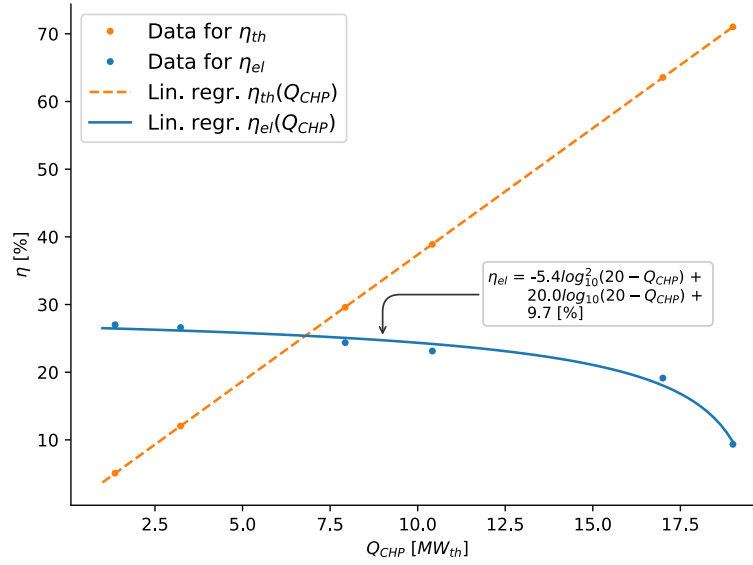


Figure 4.5: Biomass unit performances in function of the heating power ; increasing heat production leads to a decrease in electricity production.

The use of the heat storage to complement the CHP thermal power means that a charging strategy for the CHP storage systems has to be fixed to ensure a continuous heat supply all over the year. As heat and electric production are linked, charging the storage system (which is seen as an additional heat load from the CHP unit perspective) decreases the electricity that is produced by the CHP unit. Such a strategy depends on technical but also economic factors, which were not communicated. The charging strategy that was thus considered was a 'charge as soon as possible' strategy ; whenever the heat load is lower than the CHP maximum heating capacity, the storage is charged back as much as possible.

Data provided	Volumetric flow rate $\dot{V}_f$ [$m^3 N/h$]	38,500
	Moisture content MC [%]	20
	Exhaust temperature T_f [$^{\circ}C$]	120
Modelled combustion	Biomass composition	$CH_{1.494}O_{0.713}$
	Excess-air ratio λ [—]	1.6

Table 4.3: Data provided and estimated parameters used to model the exhaust gases composition

About 38,500 [Nm^3/h] of flue gases are evacuated at the stack's exhaust at a temperature close to 120 [$^{\circ}C$]. As the biomass exact composition is unknown as it is composed of waste wood coming from recycling parks, approximation on its nature and the combustion were made to model the flue gases composition. Considering a moisture content

$MC = 20$ [%], an excess air factor $\lambda = 1.6$ and using a woody biomass composition as given in [51] a deviation from the actual flue gases flow rate of less than 1 [%] was obtained, which was considered as accurate enough.

4.2.3 Economics and greenhouse emissions

Even though the biomass powerplant was designed in the first place to supply heat to UCLouvain, its operation leads to the production of electricity. According to Fabian Servais, this by-product electricity production is estimated to 54.8 [$GW^{he}/year$] for a 45.7 [$GW^{th}/year$] heat production.

Economics

Heat produced by the **CHP** powerplant is sold to UCLouvain at a fixed price, independently from the heat power required ; the electricity price, on the other hand, is determined based on the monthly Endex index.

Greenhouse emissions

Biomass combustion is considered by the Walloon and European legislation as carbon neutral ; however, taking into account other indirect emissions (such as staff mobility emissions, emissions related to the transport of biomass from the recycling parks to the **CHP** site, etc) makes the **CHP** heat and electricity production low carbon instead of carbon neutral. Based on the European Directive [2], the carbon accountability for biomass-based **CHP** units is done according to the exergetic content of the considered energy vector, as shown in Equation 4.1 :

$$EC_i = \frac{E}{\eta_i} \frac{C_i \cdot \eta_i}{C_{el}\dot{\eta}_{el} + C_{th}\dot{\eta}_{th}} \quad (4.1)$$

where EC_i is the carbon footprint of the energy vector i , η_i the conversion efficiency of the energy vector i . The Carnot factor for electricity C_{el} is by definition considered equal to one while the Carnot factor for heat C_{th} is considered to be equal to 0.3546.

Following this accountability, the **GHE** related to the electricity production and the heat production are given in Table 4.4.

	Heat - related GHE	Electricity - related GHE
Total GHE [$ktCO_{2,eq}$]	177.14	587.86
Energetic GHE [$kgCO_{2,eq}/MWh$]	3.73	10.73

Table 4.4: Greenhouse emissions share for heat and electricity production, based on [50] values and on EU Directive [2]

4.3 PV panels

The **PV panels** panels are modelled using the Python pvlib library [52].

4.3.1 Technological model

The [PV panels](#) panels are modeled as fixed mounted [PV panels](#) arrays. Typical mono-crystalline silicon [PV panels](#) panels (*Sunpower SPR X-19-240-BLK*) [53] are used for the simulations, having a lifetime of 25 [years] and an inverter efficiency $\eta_{inv} = 0.95$, considering a solar azimuth of 180° and a surface tilt of 20° .

4.3.2 Economics and typical LCOE

PV panels main costs usually lies within the investments costs, or [Capital Expenditure \(CAPEX\)](#), and not the [Operation and Maintenance \(O&M\)](#) costs ; the Annual Technology Baseline from the [National Renewable Energy Laboratory \(NREL\)](#) [54] for fixed mounted [PV panels](#) panels gives conservative estimates of 1,573.7 [$\$/kW_{peak}$] for [CAPEX](#) and 17.7 [$\$/kW_{peak} \cdot year$] for [O&M](#) while the estimated LCOE is equal to 84.8 [$\$/MWh$]. UCLouvain's maximum [PV panels](#) potential on its buildings' roof is about 1 [MW_{peak}], which corresponds to 4,167 *Sunpower SPR X-19-240-BLK* [PV panels](#) panels.

4.4 Wind turbines model

The Wind turbines was modelled with the online Renewables Ninja tool, which is based on the Virtual Wind Farm model proposed in [55] and provides estimates of wind production for different locations and types of wind turbines. The Virtual Farm model extrapolates the wind speed at the Wind Turbine's hub height based on hourly wind speed data from MERRA's database and converts wind speed to power outputs using manufacturer power curves considering a filtering due to the geographical distribution of wind turbines and correcting bias.

4.4.1 Technological model

As the exact model used by UCLouvain could not be obtained, data was extracted from Renewable Ninja tool for the year 2019 for a 900 [kW] *PowerWind 56 900* wind turbine model (whose characteristic curve is shown in Figure 4.6), as it showed the best capacity factor of the Wind Turbines modelled in the Renewable Ninja tool considering that the Wind Turbines would be located around Louvain-la-Neuve. The lifetime of a wind turbine is assumed to be equal to the one used for the [PV panels](#) panels ($l_{WT} = 25$ [years]).

4.4.2 Economic model and typical LCOE

Wind turbines, as [PV panels](#) panels, usually have Operation and Maintenance costs which are much lower than the investment costs. Conservative LCOE estimates of the NREL for land-based Wind Turbines are of 38.3 [$\$/MWh$] in 2023 for investment costs of 1,323.4 [$\$/kW$] and O&M costs of 43 [$\$/kW \cdot year$], considering the wind speed distribution in Louvain-la-Neuve 2019 (NREL Class 8 wind speed Resource).

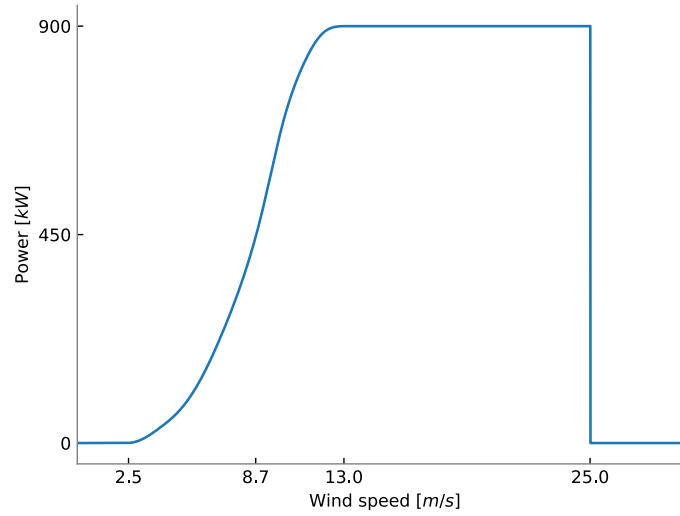


Figure 4.6: Power curve of the *PowerWind 56 900* wind turbine

As costs estimates for the purchase of Wind turbines by UCLouvain are much higher (almost twice as expensive NREL estimates), it is expected that the LCOE will be higher than estimates from literature ; it is planned that up to 12 Wind Turbines could be installed by UCLouvain.

4.5 Optimization without storage

Once the different production units to consider in the energy system presented, a first evaluation of the system behavior can be done by determining the number of wind Turbines and of [PV panels](#) panels to must be installed. Conjointly to the reduction of the [GHE](#) linked to energy production GHE_{tot} , a minimization of the annual costs for operating the energy system C_{tot} is also sought by UCLouvain. As both objectives usually compete with each other, a compromise between the environmental and economic objectives might have to be found. A quick introduction to multi-objective optimization is given in [Appendix B](#).

4.5.1 Simulation settings

The hourly operation of the UCLouvain energy system is simulated to obtain the annual costs and greenhouse emissions linked to the operation of the energy system, based on the historical consumption of electricity and heat of UCLouvain for year 2021 and on the models for the [CHP](#) unit, [PV panels](#) panels and Wind Turbines based on the models presented in [Sections 4.2, 4.3 and 4.4](#). The model parameters used for the simulations are shown in [Table 4.5](#) and detailed here-below.

Cost related parameters	
Inflation rate i_{rate} [%]	6
Lifetime l [years]	25
GHE related parameters	
Emission factor for grid electricity c_{grid} [$kgCO_{2,eq}/MWh$]	158
Emission factor for CHP electricity $c_{CHP,el}$ [$kgCO_{2,eq}/MWh$]	10.73
Emission factor for CHP heat $c_{CHP,heat}$ [$kgCO_{2,eq}/MWh$]	3.73

Table 4.5: Simulation parameters used for the basecase situation

Costs related parameters

The annual cost C_{tot} , defined in (4.2), can thus be decomposed into annualized investment costs C_{inv} , variable costs C_{var} and benefits C_{ben} .

$$C_{tot} = C_{inv} + C_{var} - C_{ben} \quad (4.2)$$

- The **annualized investment costs** C_{inv} for any equipment k are calculated based on the **CAPEX** and the installed power $P_{installed,k}$ using Equation 4.3, assuming an inflation rate i_{rate} (expressed in percent and taken equal to 6 [%] for the basecase simulation) and an equipment lifetime l (expressed in years and taken equal to 30 years for the basecase simulation) :

$$C_{inv,k} = CAPEX_k \cdot P_{installed,k} \cdot \frac{(1 + i_{rate})^{l_k} \cdot i_{rate}}{(1 + i_{rate})^{l_k} - 1} \quad (4.3)$$

As the price electricity from the **CHP** unit is fixed by the Endex index, the Endex index of the year 2020 was chosen as being more representative of the system behavior than the one of the year 2021, which ended up as particularly asymmetrical due the sudden increase of the electricity price at the end of the year ;

- The **variable costs** C_{var} are composed of **O&M** costs and fuel-related costs C_{fuel} , both being considered proportional to the installed power or the fuel consumption. Heat and electricity prices are considered as constant over the year and it is considered that there is no restriction on the minimal or maximal amount of electricity purchased from the grid ;
- The **benefits** C_{ben} consider the incomes generated by injecting electricity into the grid, due to the overproduction of electricity of production units for example. The electricity feed-in tariff on the other hand is calculated based on the **Belgian Index Spot Market (BELPEX M)**, according to the following linear pricing :

$$c_{feed-in}(h) = \begin{cases} c_{high} \cdot \text{BELPEX M} - t_{high} & \text{if } h \in [7; 22] \\ c_{low} \cdot \text{BELPEX M} - t_{low} & \text{else} \end{cases}, \quad (4.4)$$

where $c_{high/low}$, $t_{high/low}$, the feed-in tariff $c_{feed,in}$ and BELPEX M are expressed in $[\text{€}/\text{MWh}_{el}]$ and where h is the time in hour. It is also considered that there is no minimum or maximum to the electricity injection to the grid. As for the CHP electricity price, the BELPEX M was subject to extremely high variations at the end of the year 2021 ; for this reason, the BELPEX M of the year 2020 was used in the simulations.

Greenhouse emissions related parameters

The greenhouse emissions, defined in Equation (4.5), can be decomposed into the heat related and the electricity related parts. Electricity and heat emissions factors (namely the amount of equivalent carbon dioxide released per unit of energy) for each production unit are needed to characterize the energy system carbon footprint.

$$GHE_{tot} = GHE_{Heat} + GHE_{Elec} \quad (4.5)$$

PV panels and WT emission factors for electricity production are considered to be equal to zero ; the manufacturing and end-of-life of the production units, even though non-null, are not considered in this work. At the same time, the emission factors defined in 4.2 are used for the CHP unit and a fixed emission factor is considered for the electricity from the grid and is taken to be equal to the Belgian mean in 2021 (namely $f_{grid,2021} = 158 [\text{kgCO}_{2,eq}/\text{MWh}_{el}]$ [56]).

No carbon credit or similar negative carbon emissions are considered in this work : direct consumption and production of energy for UCLouvain are the only emissions taken into account so that the only way to decrease the annual GHE of the system is to decrease its electric carbon footprint.

4.5.2 Numerical model used

The UCLouvain energy system is optimized to minimize the costs and GHE of UCLouvain ; since no restriction on the electricity export or purchase to or from the grid are considered and as the CHP unit heat production (and thus electricity production) is considered to depend for each hour on UCLouvain demand, the operation of the system can be summarized as follows:

1. Compute the electricity balance $\Delta Dem_{Elec} \triangleq Cons_{Elec} - Prod_{Elec}$ for each hour h of the year ;
2. Inject electricity to the grid or purchase electricity from the available CHP power and from the grid if needed at the tariff value for hour h ;
3. Compute the fuel-related costs, the benefits and the GHE for each hour h
4. Compute the annual costs based on the investment costs and hourly fuel-related costs
5. Compute the annual greenhouse emissions based on the hourly GHE

A more complete definition of the numerical model used is detailed in the algorithmic scheme given in Appendix C, along with the obtained performances.

4.5.3 Optimization method

For optimization problems such as the one considered, being highly non-linear, genetic algorithms optimization methods such as [Non-dominated Sorted Genetic Algorithm II \(NSGA II\)](#) have proven to be particularly efficient. [NSGA II](#) is an elitist non-dominated algorithm that finds a set of best designs (namely a Pareto set) according to the objective functions [57].

For each of the m generation, an elite population E_{i+1} of N individuals (namely a set of N decision vectors) is selected from a global population of $2N$ individuals G_i . A child population C_{i+1} is then created based on the characteristics of elite individuals, and a new global population G_{i+1} is made by merging the elite population with the child population. A more detailed of [NSGA II](#) is given in Appendix A.1.

Optimization framework

The optimization of the number of [PV panels](#) panels and Wind Turbines is implemented in Python through the [Robust design optimization of renewable Hydrogen and dErived energy cArrier systems \(RHEIA\)](#) framework [58], which provides multi-objective optimization for deterministic and stochastic problems using [NSGA II](#). An additional parameter is also defined in [RHEIA](#) for the crossovers in [NSGA II](#) : the crowding degree of crossover η_{CO} , representing the strength of crossover (more exactly the resemblance of the children with the parents individuals).

Best practices rules for deterministic optimization given in [RHEIA](#) documentation [59] advise to run the population for at least 75 generations up to 250 generations (the gain in optimization being rather limited above this amount of simulations). The population size is also advised to be at least of 20 individuals to avoid converging towards local optima and to be kept below 50 individuals, as a higher population size does not significantly improve the exploration of the design space, the latter advise being given for optimization problems with less than 10 optimization variables. It must nonetheless be kept in mind that the convergence of the optimization is problem-dependent.

However, as the performances of the numerical model used to simulate the energy system without [CB](#) (which is described in Section 4.5.2) allow it a higher population size is considered to obtain a finer Pareto optimal front and along with running the optimization process for more than 250 generations ; a population size of 80 individuals is thus used for 400 generations. Default values for the crossover probability p_{CO} , the mutation probability p_{mu} and the crowding degree of crossover η_{CO} proposed by [RHEIA](#) are used and are shown in Table 4.6.

Optimization variables	Population size	Number of generation
2	80	400
Crossover probability	Mutation probability	Crowding degree of crossover
0.9	0.1	0.2

Table 4.6: **NSGA II** parameters used for the optimization

4.6 Simulation Results

4.6.1 Basecase results

Considering the heat demand of the UCLouvain energy system, it appears that the **CHP** unit mean electric power is actually much higher than the electric consumption of UCLouvain (6.572 [MW_{el}] vs 2.968 [MW_{el}]) so that UCLouvain can be practically independent from the grid considering the only electricity production from the **CHP** unit. As the heat demand is fixed, installing **PV panels** and **WT** can only reduce the **GHE** linked to the electricity production.

The Pareto front is shown in Figure 4.7 along with the **PV panels** and **WT** installed capacity ; in-depth details for the best economic case (design with the minimum costs), the best **GHE** case (design with the minimum **GHE**) and an intermediate case are given in Table 4.7 :

- The very low pricing for the electricity from the **CHP** unit and the low feed-in tariff makes the no-**PV panels**-nor-**WT** option **the best economic option**; the annual system **GHE** in this case are equal to 0.529 [ktCO_{2,eq}] ; heat related **GHE** are equal to 0.196 [ktCO_{2,eq}] while the electric related **GHE** are equal to 0.333 [ktCO_{2,eq}]. The electricity from the grid only represents 1.41 [%] of the annual electricity consumption but represents 11.0 [%] of the electricity carbon footprint. The mean electric carbon footprint for the energy system in such a configuration is equal to 12.8 [kgCO_{2,eq}/MWh_{el}], which is extremely low compared to the Belgian mean ($f_{grid,2021} = 158$ [kgCO_{2,eq}/MWh_{el}]) ;
- On the other hand, **the best GHE option** is to install the maximal capacity for both the **WT** and the **PV panels** panels. By doing so, the annual **GHE** are reduced by 47.1 [%] for a cost increase of 40.6 [%] compared to the best economic case. More specifically, the electric carbon footprint decreases by 74.7 [%]. With such a design, the system is 71.4 [%] self-sufficient in electricity (self-sufficiency denoting how much of the energy consumed is produced, as opposed to the self-consumption, denoting how much of the energy produced is consumed) and consumes 62.6 [%] of the renewable electricity produced (leading to an excess production of 11.1 [GWh_{el}]). It might be noted that the electricity is mainly injected during day-time in such a design, and that the end of a

Even in such a design, electricity must eventually be purchased to the grid but

	Best economic case (No PV panels and no WT)	Intermediate Case	Best GHE case (Full Renewable potential)
Annual costs [M€]	2.397	2.800	3.371
Annual GHE [ktCO _{2,eq}]	0.529	0.343	0.280
Mean electric carbon footprint [kgCO _{2,eq} /MWh _{el}]	12.8	5.7	3.3
Electricity costs share [%C _{tot,el}]	100		
Investment (Renewables) [%C _{tot,el}]	0	0.69.6	102.6
Purchase (CHP) [%C _{tot,el}]	96.3	33.0	13.1
Purchase (grid) [%C _{tot,el}]	3.7	0.4	0.2
Benefits (export to grid) [%C _{tot,el}]	0	-3.0	-15.9
Heat GHE share [%GHE _{tot}]	37.0	57.0	69.8
Electricity GHE share [%GHE _{tot}]	63.0	43.0	30.2
CHP [%GHE _{tot}]	52.0	40.6	28.3
Grid [%GHE _{tot}]	11.0	2.4	1.9
Self-sufficiency (Electricity) [%Dem _{el}]	0	49.9	71.4
Self-consumption (Renewables) [%Prod _{ren}]	0	89.5	62.6

Table 4.7: System breakdown of the electricity costs, GHE emissions and use of renewable electricity for the three design analyzed. Electric shares are calculated based on the total electric-related costs taking into account benefits, which are defined as negative since they reduce the annual costs.

in small proportions compared to the overall demand, the grid dependency falling down to 0.13 [%] of the annual demand. In this scenario, most of the electricity produced by the CHP powerplant over the year is hence not consumed and the biomass unit is used as a back-up unit for low-production periods.

- The **intermediate design** studied is the design located in the middle of the Pareto front (namely the 40th individual of the Pareto front, starting from the lowest annual costs). This design shows a 16.8 [%] increase in the annual costs for a 35.2 [%] decrease in the annual GHE with respect to the best economic case, meaning that the first half of the Pareto curve include the designs reduce the annual GHE in the most cost-effective way. The amount of renewable electricity injected into the grid in this design is relatively limited compared to the actual electricity production (10.5 [%] of the annual production). This higher self-consumption is actually due to the fact that the PV panels and WT capacity is much lower than in the best GHE option. It can also be noted that the installed power allows the

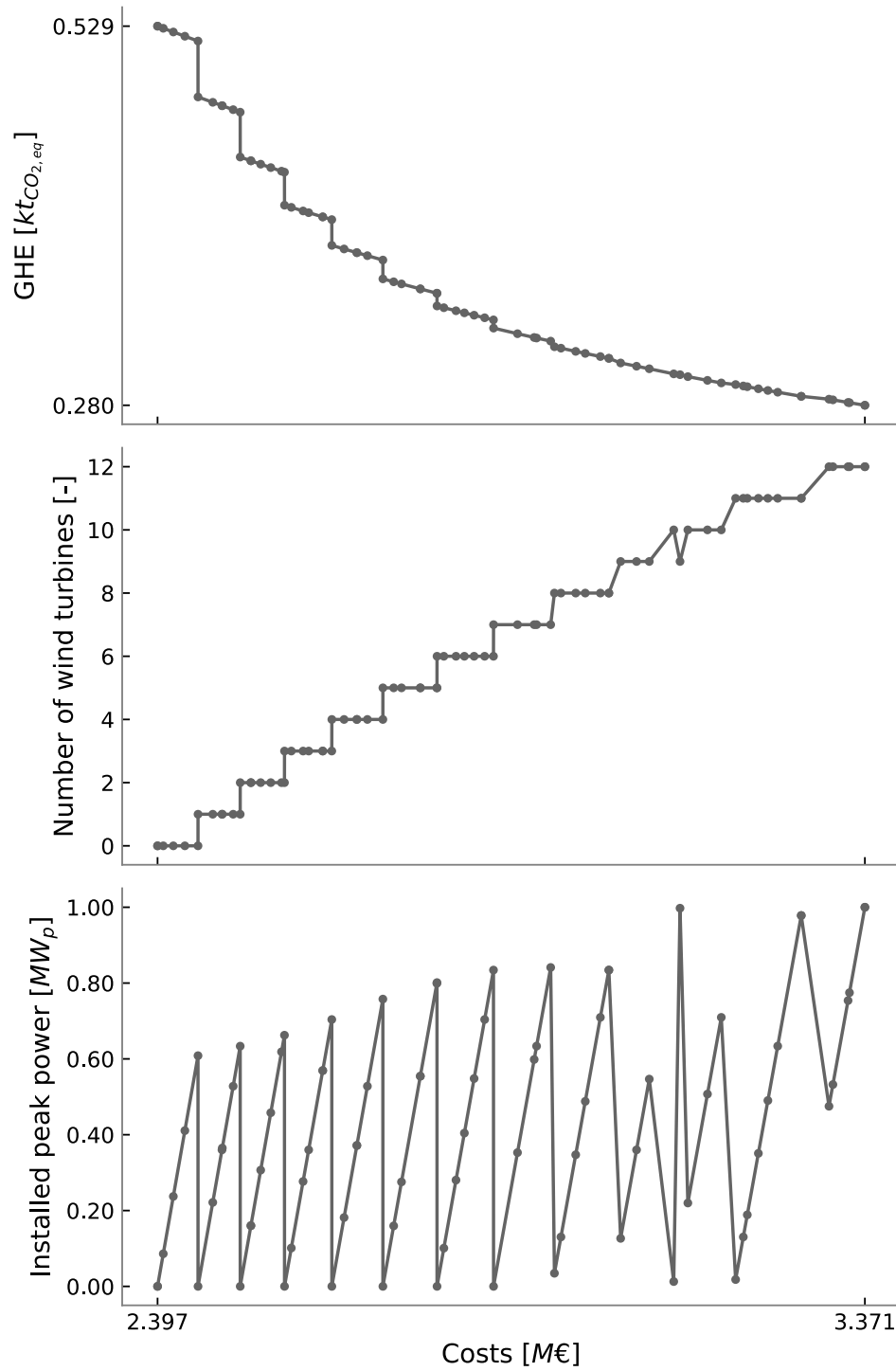


Figure 4.7: Pareto front and installed renewable capacity for the designs obtained from the optimization of the system. The saw pattern in the **PV panels** shows the economic attractiveness of **WT** electricity compared to the **PV panels** one, given that the **PV panels** capacity is much more continuous due to their much smaller peak capacity.

energy system to be 49.9 [%] self-sufficient in its electricity consumption.

4.6.2 Sensitivity analyzes

In order to understand up to which point the optimal designs obtained for the system are dependent on the values used for the parameters, sensitivity analyzes were conducted on them. For each Pareto optimal designs, costs and GHE were calculated varying one parameter value and keeping all the other values equal. It appears that the most impactful parameters are the **CHP** electricity price $c_{CHP,elec}$, the inflation rate i_{rate} and the feed-in tariff c_{grid} . Other parameters such as the biomass emissions or the lifetime of the renewable production units that are fixed by the manufacturer were not investigated, whereas 'floating' parameters (such as the carbon footprint of electricity from the grid) did not show to strongly influence the energy system behavior.

- The **CHP electricity price** $c_{CHP,elec} = c_{CHP,elec,basecase} + c_{add,CHP,elec}$ has a huge impact on the determination of the Pareto front as the optimal designs do not have the same self-sufficiency for electricity. The additional cost $c_{add,CHP,elec}$ is considered constant over the year and can represent a margin that Veolia could make by increasing the tariff ; an additional cost of only 15 [€/MWh_{el,CHP}] leads to a 16 [%] increase in the costs of the best economic case, almost completely dependent on the **CHP** electricity while it only leads to a 4.6 [%] increase in the costs of the best **GHE** case, which is highly self-sufficient.

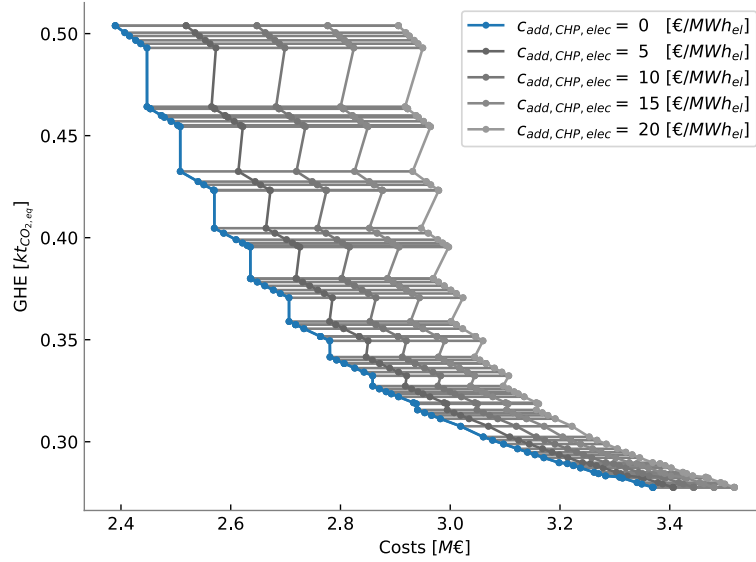


Figure 4.8: The cost at which is sold electricity from the **CHP** unit has a huge impact on the Pareto front shape, as low self-sufficient design suffer more from a higher pricing than highly self-sufficient ones. The basecase Pareto Front is shown as the blue curve.

- The value of the **inflation rate** is also a parameter to which the system costs are strongly sensitive ; the higher the i_{rate} , the higher the investment costs and the lower the i_{rate} , the lower the investment costs. This parameter is however not *per*

se a parameter that is supposed to vary a lot, and the choice of its value is usually done along with the determination of the desired cash-back period ;

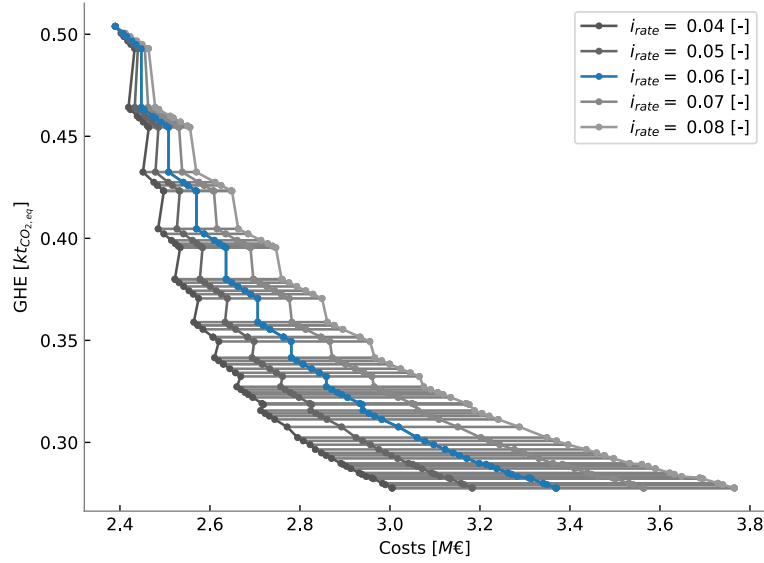


Figure 4.9: The inflation rate has a quasi linear impact on the annual costs for the designs of the Pareto front : a deviation of 3.5 [%] in the annual costs is observed with respect to the basecase scenario ($i_{rate} = 6$ [%], in blue) for each percent deviation from the basecase value.

- Last, a reduction of **the feed-in tariff** $c_{feedin} = c_{feedin,basecase} - c_{decr,feedin}$ does not seem to have an impact comparable to the one of an increase of the **CHP** electricity. By reducing the feed-in tariff by a fixed amount over the year $c_{decr,feedin}$ and bounding c_{feedin} to positive values, it appears that the best **GHE** case only suffers from a 5.9 [%] increase in the annual operation costs ; in other words, benefits due to injection of electricity are very limited.

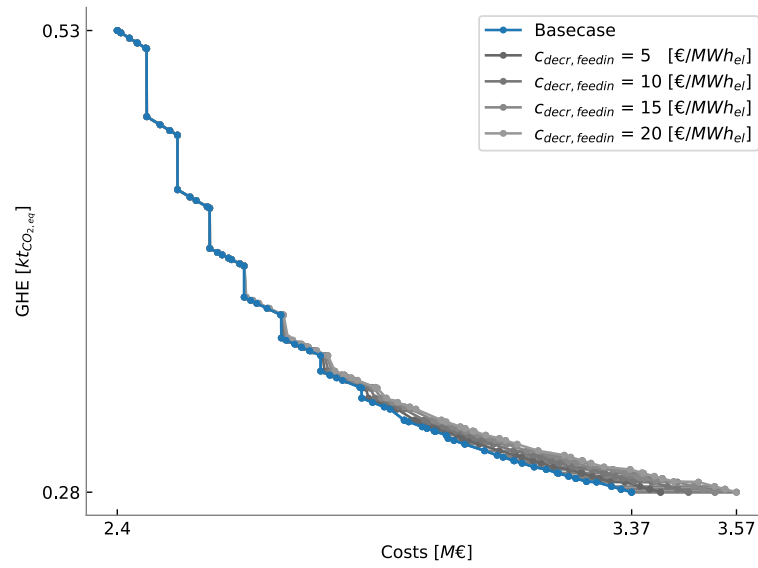


Figure 4.10: Systems injecting high amount of electricity into the grid are disadvantaged by a less profitable feed-in tariff compared to the basecase Pareto Front (shown as the blue curve). This economic prejudice is however limited and is only remarkable for an important decrease of the feed-in tariff.

Chapter 5

Economic model of the Carnot Battery

“ Nothing in life is certain except death, taxes, and the second law of thermodynamics. ”

Seth Lloyd

Once the storage system characterized by a thermodynamic model, an economic model has to be proposed for the Carnot Battery, in order to determine the specific costs of such costs. The estimation of the costs of a project is a crucial part during the design phase to evaluate its feasibility. The accuracy of the costs estimates depends on the complexity of the plant and the level of knowledge on its characteristics, the reference information available and an appropriate evaluation of the project contingencies [60].

This section presents the methodology used to get the economic model in Section 5.1 ; after holding in Section 5.2 a brief discussion on the difficulty to find appropriate economic models for CB, the economic model used for the CB is presented in Section 5.3.

5.1 Existing cost estimation techniques

Cost estimation techniques often show a trade-off between accuracy range and effort and labor to make the estimates ;

- **Rough but quick estimates** can be made by estimating the equipment costs C_{equip} based on its characteristic size A_{equip} using costs correlations for equipment with similar characteristics f_{equip} :

$$C_{equip} = f_{equip}(A_{equip}) \quad (5.1)$$

The capacity exponent ratio method shown in (5.2) is often used as the cost correlation in rough estimates, based on a reference equipment having a characteristic

size A_{ref} and a cost C_{ref} . The exponent factor n is determined through correlation of data, but is usually close to 0.6.

$$C_{equip} = C_{ref} \cdot \left(\frac{A_{equip}}{A_{ref}} \right)^n \quad (5.2)$$

Cost correlation of projects (which include the cost of the module, its installation and other costs such as engineering labor) are usually less accurate than module costs as the cost decomposition are usually not known and as specific project circumstances can have large impact on the project costs.

- **Finer but more complex estimates** can be made by decomposing the equipment into its components. Each component cost $C_{equip,i}$ is then calculated based on its typical characteristics A_i through cost correlation f_i , determined by the costs of components with similar characteristics.

$$C_{equip,i} = f_i(A_i) \quad (5.3)$$

Alternatively the Bare Module cost technique can be used ; a logarithmic polynomial is used to determine the initial component cost $C_{equip,i}^0$ as shown in Equation (5.4), where the polynomial order p_{max} is generally limited to 2 and where the $K_{i,p}$ are determined by correlation on the data. The component cost is then obtained by multiplying the initial component cost by a Bare Module factor $F_{BM,i}$ which depends on the type of component

$$\log_{10} C_{equip,i}^0 = \sum_{p=0}^{p_{max}} K_{i,p} (\log_{10} A_i)^p \quad (5.4)$$

$$C_{equip,i} = F_{BM,i} \cdot C_{equip,i}^0 \quad (5.5)$$

The equipment cost is equal to the sum of the price of its components multiplied by an installation factor i_{inst} (which range between values as low as 1.18 to values as high as 4.16) taking into account the costs for the integration of the project and of the labor.

$$C_{project} = i_{inst} \cdot \sum_i^{components} C_{equip,i} \quad (5.6)$$

- For rough or fine estimates, **historical costs** that were used to compute the overall costs **have to be updated** due to the effect of inflation over the year. This is usually captured by multiplying the old cost C_{old} by the ratio of the current cost index $I_{current}$ and old cost index I_{old} as in Equation (5.7).

$$C_{current} = C_{old} \cdot \frac{I_{current}}{I_{old}} \quad (5.7)$$

5.2 Finding appropriate models

As it has been highlighted in the literature review, the economic aspect of [Rankine-based CB](#) has been very scarcely explored. This is mainly due to the fact that this technology

is still under development and that very few Rankine-based CB have been constructed, their size ranged usually from the [W] scale to the [kW] scale [25]; even though the construction of an experimental [MW]-scale prototype is considered in [27]. Most of the studies assessing the thermo-economic potential of CB evaluated the cost of a CB by performing a component-wise analysis using the Bare Module cost technique based on existing correlations.

As the aim of this master's thesis is to provide a tool to screen the economic potential of integration of CB without prior knowledge on the sizing range that can be expected for the different components, up-to-date economic correlations that can be used over large ranges would be needed. The study of Norregard et al. [30] was on the other hand the only one to considered specific investment costs which did not depend on the power installed for the each of the three sub-systems (namely the VCHP, the TES and the ORC), as the expected installed power was in the range of the [MW] scale.

If this method is expected to give rougher estimates than the one used by the rest of the existing studies as it does not capture the reduction in specific costs usually obtained for larger sizes of CB, the lack of research for this topic cannot guarantee an improved accuracy. As it was highlighted by Dumont et al. [11], HEX must be oversized to obtain extremely low pinch-point temperature difference and as the technology of HEX to use has not been defined yet, using usual economic correlation for HEX might not be appropriate for example.

However, none of the aforementioned studies have considered off-design models able to capture the variations in the performances of the CB in operational conditions. An intermediate cost estimate method, similar to the one used by Norregard et al. and using project costs (also referred to as SIC, as opposed to the module costs also referred to as Specific Equipment Cost (SEC)), is thus proposed in this thesis by considering economic correlations for the three sub-systems of the CB. Even if the level of detail of the economic model will be lower than if an economic correlations was found for each of the components, this method will allow to determine for which sub-systems a more detailed economic model is needed given the off-design operation of the CB.

5.3 Economic model proposed

Economic correlations for VCHP, TES and ORC showing similar characteristics to the one of the sub-systems are hence sought. Considering the VCHP, correlations for High-temperature Waste Heat Recovery HP using industrial waste or flue gases should be preferred, as cooling down flue gases from the biomass CHP will provide high-temperature waste heat to the VCHP cycle in the investigated integration scenarios, hence improving the performances compared to the case where the ambient would be considered as the charging cycle heat source. Considering the TES, a cost estimate of pressurized water tanks operating between 90 and 140 [°C] is needed ; considering the ORC, finding a cost estimate for Waste Heat Recovery ORC based on their discharging power would be a good fit, as the maximum temperatures to consider for the discharging cycle are lower than 140 [°C]

5.3.1 Heat Pump

The cool-down of flue gases has however been identified in the literature as leading to several degradation to the materials used : for flue gases of biomass boilers, the acid dew-point (which characterize the temperature at which acids condensate) is around 100 [°C] [61, 62] and the formation of corrosive hygroscopic salts which could damage equipment such as economizers used for waste heat recovery was also highlighted by Retschitzegger et al. in [63]. Overall, and even if a 'safe operating zone' has been identified by Li et al. around the water dew-point [64], located around 65 [°C] for the flue gas considered, cooling down gases lead to a degradation of the equipment which has to be replaced more often ; this could lead to additional costs for the **VCHP** compared to the use of a less damaging heat source.

Pieper et al. explored in [65] the allocation of investment costs for different types of heat pumps supplying **DHN**; half of the investment costs of heat pumps using flue gases were identified to be linked to the purchase of the module for the cases reviewed in the study. If the heat pumps considered cooled down the flue gases down to 40 [°C] and showed **COP** ranging between 3.1 and 5.3, the heat sink temperatures were rather low (about 50 [°C]) compared to the **VCHP** expected operation (with a condensing temperature higher than the storage temperature). A piece-wise defined economic correlation was provided for heating capacities (namely, thermal powers at the sink) ranging between 0.5 and 10 [MW_{th}].

On the other hand, Kosmadakis et al. investigated in [37] high-temperature waste heat **VCHP** rejecting heat up to 150 [°C] and using R1233zd(E) as the working fluid, for configurations with single-stage intermediary **HEX**, double-stage intermediary **HEX** and without intermediary **HEX**. The **COP** ranged between 3 and 4.5 for heat source temperatures of 100 [°C] in the configuration without **HEX**. A **SEC** estimation was provided for heating capacities ranging between 133 [kW_{th}] and 4500 [kW_{th}] for single-stage intermediary **HEX**, using the Bare Module method. An installation factor of 4.16 was additionally considered to obtain the **SIC**.

If the value of installation factor used by Kosmadakis et al. might be rather high compared to the ratio given by Pieper et al, the gap observed between the costs estimates reported by those two articles is rather small : Kosmadakis et al.'s give **SIC** estimates between 411 and 740 [€/kW_{th}] while Pieper et al. report values between 440 and 630 [€/kW_{th}]. If the former costs estimate does not consider additional costs due to the flue-gases cool-down degradation as the latter, which is based upon project data, it uses a **HP** model more similar to the one of the **CB**. It appears that the **SIC** used by Norregard et al. in their techno-economic assessment a **SIC** of 500 [€/kW_{th}] for [MW]-scale **HP** fits pretty well the value obtained by the aforementioned cost estimates.

The correlation of Pieper et al. is hence used as the basecase economic model for the **VCHP**, considering a extension to 133 [kW_{th}] using the values reported by Kosmadakis et al. for that heating power :

$$SIC_{VCHP}(Q_{heating}) = a \cdot Q_{heating} + b \quad (5.8)$$

where a and b are defined piece-wise as in Table 5.1

	0.133 – 0.5 [MW]	0.5 – 1 [MW]	1 – 4 [MW]	4 – 10 [MW]
a	0.300	0.2	0.023	0.003
b	0.74	0.63	0.53	0.46

Table 5.1: Piece-wise definition of the costs correlation for the VCHP defined in (5.8)

5.3.2 Thermal Energy Storage

As it has already been mentioned, pressurized water tanks are considered as the most cost-effective option for STES for temperatures between 90 [°C] and 140 [°C] and are often used in the thermo-economic studies. However, it appears from their costs often represent a major part of the investment costs, with total costs shares between 20 [%] and 40 [%] [12, 17] ; Frate et al. even report in [66] a cost share close to 60 [%] for P2P ratio values in the range of 30 [%].

Frate et al. calculated in [66] values of SEC per unit of electricity stored based for different P2P ratio, corresponding to SEC between 400 [€/kWh_{th}] and 600 [€/kWh_{th}] for 500 [kW_{el,ch}] CB while values of about 250 [€/kWh_{th}] are obtained for 5 [MW_{el,ch}] CB. Fan et al. considered in [16] a simple cost correlation based on the volume of the tanks used and on the price of industrial water, reporting SEC values of 28.9 [€/kWh_{th}] when considering two tanks configurations. The values reported by Hu et al. in [17] are on the other hand extremely high, close to 4,395 [€/kWh_{th}] which seem rather high given the cost-effectiveness that was put forwards in the literature for this storage system.

The cost correlation of Hu et al. for the SEC is considered for two tanks in this work using an installation factor of 4.16 to obtain the SIC as in Equation (5.10) keeping in mind that this cost estimate might underestimate the real cost of the storage tanks.

$$SIC_{TES}(V_{tank}) = (0.0014 \cdot \rho_{water} + 167.19 \cdot 2) \cdot V_{tank} \cdot i_{inst} \quad (5.9)$$

5.3.3 Organic Rankine Cycle

ORC is a technology pretty well known and for which a large range of projects have already been completed. Lemmens reviewed in [67] the SIC for a large amount of projects over a broad power capacity ; among those, the costs of eight Waste Heat Recovery ORC projects with electrical power ranging between 90 [kW] and 5500 [kW] are given. The SIC reported range between 1,849 [€/kW_{el}] and 3,820 [€/kW_{el}], which is in the range of values obtained by Quoilin et al. in [46] (2,136 [€/kW_{el}] to 4,260 [€/kW]) for kW-scale Waste Heat Recovery ORC but much higher than the value of 800 [€/kW_{el}] used by Norregard et al in their study [30].

A polynomial logarithmic regression of second order of the SIC of the project reviewed by Lemmens is performed to obtain a cost correlation based on the electrical power $P_{el,ORC}$.

$$SIC_{ORC}(P_{el}) = -4887 + 5712 \cdot \log(P_{el}) - 991.5 \cdot \log^2(P_{el}) \quad (5.10)$$

Chapter 6

Integration of the Carnot Battery

“ I Have Not Failed. I Have Just Found 10,000 Things That Do Not Work. ”

Thomas A. Edison

The aim of this thesis is to provide a tool to investigate the potential of integration of a [Rankine-based CB](#) in energy system ; thermodynamic and economical models for the Carnot Battery having been proposed and the optimal design of UCLouvain energy system being already investigated, determining how to integrate the storage into the energy system is necessary. Two integration scenarios are proposed in Section 6.1. A preliminary analysis of the potential of integration of [TI-Rankine-based CB](#) in UCLouvain energy system is then provided in Section 6.2 to understand what is the potential of integration *a priori* of such a storage system in UCLouvain energy system.

6.1 Integration scenarios

The [CB](#) could be used as a storage system charging on the excess production from the [PV panels](#) and the [WT](#) of UCLouvain energy system and discharging electricity to the system to match the demand when needed. As it has been highlighted in Section 3.3, the use of a high-temperature heat source leads to an important boost in the [P2P ratio](#). At the same time, the biomass [CHP](#) unit produces 38,500 [m³N] of flue gases per hour at a temperature of 120 [°C] ; if the flue gases are cooled down to 60 [°C], this represents a heat source thermal power of about 820 [kW_{th}] which represents a significant energy flux that could be tapped, assuming that Veolia allows the free use of those flue gases. If it is true that the limitation in the heat source use (fixed by the flue gases flow-rate) will have an impact on the [CB](#) sizing, the improvement in performances due to the high-temperature heat source is considered to be overall more interesting than using the ambient as the [VCHP](#) heat source.

At the same time, the ambient temperature in Louvain-la-Neuve is rather low, about 10 [°C], which allows to use it as a low-temperature [ORC](#) heat sink over the year, boosting

even further the **P2P ratio**. However, even though **TI-Rankine-based CB** are usually considered as electric storage system, they can technically provide heat at intermediate temperatures ; as it was assessed in Section 3.3, **P2P ratio** values of about 40 [%] could be obtained when rejecting heat at heat sink temperature between 50 [°C] and 70 [°C]. As UCLouvain heat demand is satisfied by buying heat to Veolia **CHP** powerplant, a reduction in the annual expenses but also the **GHE** could be obtained if the thermal energy rejected during the discharging cycle was used to supply part of the UCLouvain heat demand. This would investigate the idea proposed by Steinmann et al. to use **CB** as sector coupling technologies [32].

Two integration scenarios are hence considered in this work : a **P2P ratio-enhancing scenario**, using the flue gases as the **VCHP** heat source and the ambient as the **ORC** heat sink and a **Fully thermally integrated scenario**, rejecting heat to the **DHN** when discharging electricity to the system.

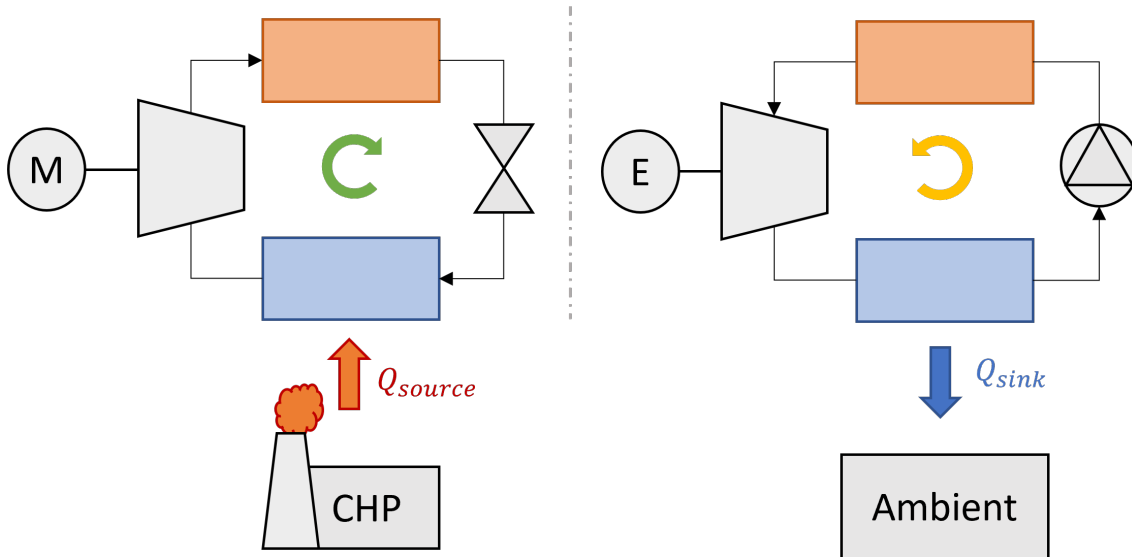


Figure 6.1: In Scenario 1, the **CB** extracts heat from the **CHP** flue gases when charging and rejects heat to the ambient when discharging. The amount of heat that can be charged in is limited by the heat that can be extracted from the flue gases ; the amount of heat that can be rejected (and thus the discharging electrical power) is not limited.

6.1.1 Scenario 1 - Power-to-power enhancement

In Scenario 1, the **CB** is charged whenever there is an excess production of electricity from the **PV panels** and **WT**, using the flue gases as the heat source at a fixed temperature of 120 [°C]. As the thermal power available in flue gases is limited, the **CB** heating power (and thus charging power) is bounded each hour either by the excess electricity production, either by the heat source capacity. The thermal power that can be rejected at the **ORC** is however assumed not to be limited as its rejects heat to the ambient, whose mean temperature over the year is close to 10 [°C], so that the discharge power is never limited. This integration scenario is shown in Figure 6.2.

6.1.2 Scenario 2 - Full thermal integration

The CB is charged in Scenario 2 as in Scenario 1, using the flue gases as the charging cycle heat source and being limited in its charging power by its thermal power. The electricity is discharged to the grid whenever the electric is greater than the PV panels and WT production whereas the ORC rejects heat to the DHN. Even though the average heat demand is an order of magnitude greater than the flue gases thermal power, it must be verified that the heat rejected into the DHN is smaller or equal to the heat demand ; the electric discharge is hence bounded by the deficit of electricity production.

It must be added that when discharging heat to the DHN, the lift over the heating station (i.e., the temperature difference between the return flow and the go-flow) is considered to fix the glide at the heat sink.

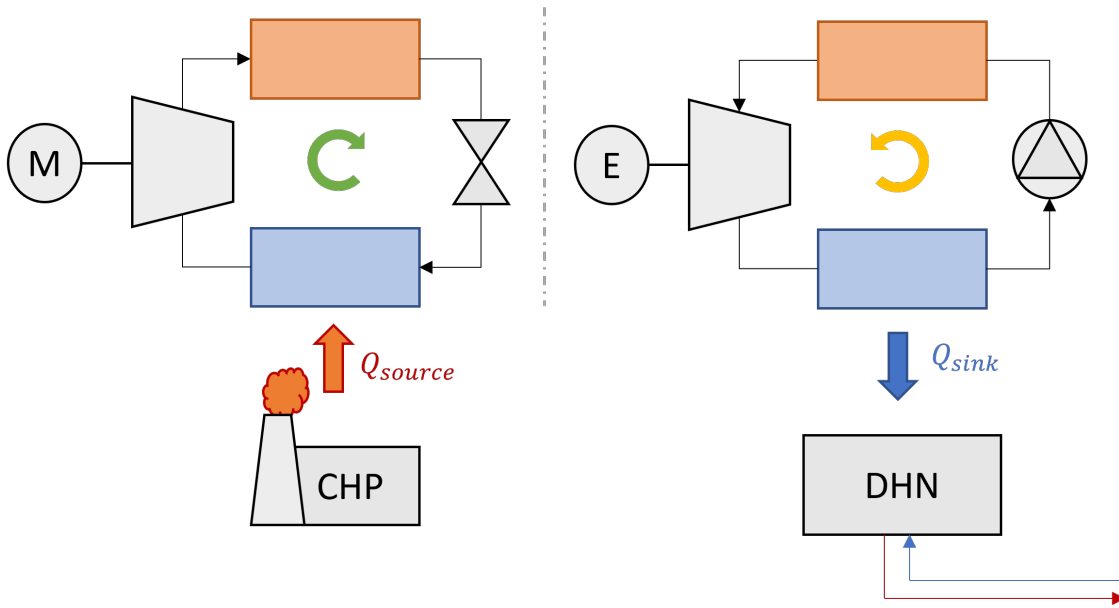


Figure 6.2: The CB extracts heat from the CHP flue gases when charging and rejects heat to the DHN when discharging. The amount of heat that can be charged in is limited by the heat that can be extracted from the flue gases ; the heat demand at a given time limits the amount of heat that be discharged (and thus the discharging electrical power) at that time.

6.2 Preliminary analysis

Before trying to optimize the energy system design, a preliminary analysis can be performed to determine *a priori* the potential of integration of a Carnot Battery given the economic and energetic system framework.

6.2.1 Economic potential

Energy storage systems can be used within energy systems with high shares of renewable production to reduce the dependency on external actors and improve self-consumption. To observe an economic benefit, investment costs must be counterbalanced by a reduction in the purchase costs to observe an economic benefit. When the excess production can be valued, the loss of revenues due to the injection of energy in the storage instead of injecting it in the grid is also to consider. The economic benefits due to the addition of a storage system to an already existing system can be determined as in Equation (6.1).

$$Ben_{sto} = \underbrace{\sum_i^{year} c_{purch,i} \cdot E_{avoided,i}}_{\triangleq C_{avoided}} - \left(\underbrace{\sum_j^{year} c_{feedin,j} \cdot E_{excess,j}}_{\triangleq C_{loss \text{ revenue}}} + CAPEX_{sto} \right) \quad (6.1)$$

where $C_{avoided}$ is the annual reduction in costs linked to a smaller purchase of energy, $C_{loss \text{ revenue}}$ represents the annual loss of revenue due to a lower injection of electricity to the grid and $CAPEX_{sto}$ represents the costs linked to the installation of the storage system. In the case of a CB, the heat extracted from the flue gases to charge the battery is considered as free and for Scenario 2, in which heat is provided to the DHN when discharging, Equation (6.1) can be extended by considering both the avoidance in electricity and heat purchase to calculate $C_{avoided}$.

Generally speaking, both the purchase price c_{purch} and the feed-in tariff c_{feedin} vary over the year; so do the storage charging and discharging power of the CB, meaning that defining an 'hourly' P2P ratio becomes practically impossible as the energy charged in is not always discharged at once nor in the same conditions. The exact economic benefit of adding a storage system can thus hardly be computed *a priori*, and using average values and making assumptions is required to screen the economic potential of using a storage.

Considering the average electricity purchase cost and feed-in tariff and the P2P ratio over the year, the economic benefits for Scenario 1 (providing only electricity) can be estimated as in Equation (6.3) with $E_{excess,tot}$ the total amount of electricity charged into the storage system over the year and $CAPEX_{sto}$ the investment costs depending on the design chosen.

$$Ben_{sto} \approx \overline{c_{purch}} \cdot E_{excess,tot} \cdot \epsilon_{P2P} - \overline{c_{feedin}} \cdot E_{excess,tot} - CAPEX_{sto} \quad (6.2)$$

$$= E_{excess,tot} \cdot \overline{c_{purch}} \left(\epsilon_{P2P} - \frac{\overline{c_{feedin}}}{\overline{c_{purch}}} \right) - CAPEX_{sto} \quad (6.3)$$

It thus appears that if the P2P ratio is too low for the difference between the purchase cost and the feed-in tariff, **whatever the investment costs, adding a storage system is not beneficial** for reducing the operational costs. In UCLouvain energy system, even though $c_{purch} > c_{feedin}$ most of the year and $\overline{c_{purch}} > \overline{c_{feedin}}$, it appears that designs leading to high P2P ratio and low investment costs per unit of energy stored are mandatory for both scenarios to observe an economic benefit when adding a Carnot

Battery.

For Scenario 2, providing both heat and electricity when discharging, Equation (6.3) can be modified to take into account the avoided purchase of heat and hence reads as in Equation (6.4) assuming a storage efficiency $\eta_{sto} = 1$, where c_{heat} is the heat purchase cost.

$$Ben_{sto} \approx E_{excess,tot} \cdot \left(\overline{c_{purch}} \left(\epsilon_{P2P} - \frac{\overline{c_{feedin}}}{\overline{c_{purch}}} \right) + \overline{c_{heat}} \cdot (1 - \eta_{th}) \cdot COP \right) - CAPEX_{sto} \quad (6.4)$$

Even though the **P2P ratio** is expected to be much lower than in Scenario 1 and too low to expect an economic benefit by the sole increase in electricity self-consumption, high heat purchase costs might allow the storage system use to be beneficial overall.

6.2.2 Greenhouse emissions avoidance

By improving the self-consumption of renewable energy, the addition of energy storage systems to an already existing energy system can also lead to a reduction in the **GHE**, the importance of this emission avoidance being mainly driven by the carbon footprint of the energy that is not purchased.

In the case of UCLouvain energy system, the low-carbon footprint of both heat and electricity from the **CHP** and the very low dependency on the grid does not allow a high reduction of the **GHE**, especially for Scenario 1, as the heat-related **GHE** cannot be reduced in such a scenario. The potential of **GHE** avoidance is technically higher in Scenario 2 as the heat demand can be reduced, but is impacted by a lower **P2P ratio**; as the **CHP** electricity carbon footprint is about 3 times higher than the one of **CHP** heat, decent values must be obtained for the **P2P ratio** to observe a significant potential of **GHE** avoidance.

It must also be added that the avoidance costs (namely the investment costs needed to obtain a given reduction in the **GHE**) of adding more renewable must be higher than the one of adding the energy storage system to observe an interest of installing storage system instead of increasing the renewable capacity when it is possible. Based on the results obtained in Section 4.5, the avoidance costs for considering the intermediate case instead of the best economic case were of 1.721 [M€/ktCO_{2,eq,avoided}]; when considering the best **GHE** case instead of the intermediate case, they were of 9.027 [M€/ktCO_{2,eq,avoided}]. The fivefold increase in the avoidance cost is mainly linked to the decrease of the share of electricity-related **GHE** from 0.333 [ktCO_{2,eq}] in the best economic option to 0.147 [ktCO_{2,eq}] for the intermediate case and 0.088 [ktCO_{2,eq}] in the best **GHE** option.

Chapter 7

Conclusion and future work

The potential of Carnot Batteries has been identified to have been merely investigated in operational conditions for real energy systems. Based on a semi-empirical model for volumetric machines, an operational model has been proposed. The impact of different parameters identified as having a high impact on the **P2P ratio** has been assessed through sensitivity analyses ; the impact of the subcooling of the **ORC** feed-pump and of its isentropic efficiency has been explored ; results show that, even if the impact of those parameters is mild, it is non-negligible and the proposed model would benefit from a clearer modelling of such components.

HEX have again been identified as being key components of the **CB** ; however, as assessing the potential of integration of **CB** in energy systems means exploring **CB** designs over a large range of power, a choice of **HEX** cannot be made. Defining a clearer range of power for the **CB** to investigate might allow to use more precise models and would provide a significantly more realistic operational model.

The UCLouvain energy system behavior has been presented in details and an operational model has been presented for the system considered. A study of the optimal sizing for the **PV panels** and **WT** has been conducted and showed two things :

1. Without considering the use of storage system, a 47 [%] decrease in the **GHE** could be achieved by increasing by 41 [%] the expenses for the operation of the energy system to install 12 900 [kW_{el}]-**WT** and 1 [MW_{peak}] of **PV panels**, leading to an electric carbon footprint of 3.3 [kgCO_{2,eq}/MWh_{el}] comparable to the heat carbon footprint of 3.73 [kgCO_{2,eq}/MWh_{th}]. As no thermal production unit are considered, the system carbon footprint cannot be fall down below an annual **GHE** of 0.194 [ktCO_{2,eq}] ;
2. Due to the feed-in tariff and the purchase costs of electricity to the **CHP**, the integration of a storage system can hardly be beneficial from an economic point of view except if very low investment costs can be achieved for a high storage efficiency. However, as the energy system is highly sensitive to the purchase costs of electricity from the **CHP** (the system still purchasing 29 [%] of its electricity for the design leading to the lowest **GHE**), the introduction of a storage system might improve the system economic resiliency.

The scarcity of techno-economic assessments, the lack of actual projects and the diversity encountered in the cost estimates found in literature highlighted the need to research gap existing for the economic aspect of Carnot Batteries, especially if potential integration screening is to be performed without prior knowledge on the storage system size.

Last, two integration scenarios for the CB were identified, both using the flue gases from the CHP powerplant as the heat source of the charging cycle. the first scenario proposed focused on a P2P ratio enhancement by using a low-temperature sink while the second allows the heat rejection at the ORC heat sink to decrease the system heat demand, trading a high P2P ratio with a more complete integration in the energy system. Other integration scenarios, could nonetheless be investigated, as the heating thermal power of the charging cycle is bounded by the thermal content of the flue gases in the presented scenarios ; simultaneously, considering the use of a TI-Rankine-based CB in different system configurations (namely, considering to use others technologies such as standalone biomass boilers for the same heat and electricity demands) might help to determine in which energy systems and under which conditions a TI-Rankine-based CB has a potential of integration.

7.1 Perspectives for future work

Based on the work that has been presented in the present document, a list of the challenges to overcome and of the perspectives for future work can be make :

- Identifying a HEX model taking into account the operational conditions for a given power range of CB while being low-computationally demanding would significantly improve the mapping of CB in real operational conditions ;
- Developing more realistic and up-to-date economic models for CB, taking into account the specificities of the technology, to allow to assess more accurately the economic potential of CB ;
- Successfully coupling of the presented CB model to the UCLouvain energy system model to determine if there is or not a potential of integration of TI-Rankine-based CB considering the proposed scenarios and the economic framework of the system.

Appendix A

Numerical model used for the Carnot Battery

A.1 Numerical model

The numerical model that is used to simulate the Carnot Battery operation can be decomposed into a model solving for the HP cycle, whose main operations are described in Algorithm 1 and a model running for the ORC cycle, similarly described in Algorithm 2. The model used for the isentropic efficiency of the compressor/expander and the pressure losses do not allow to solve the cycle at once, so that an iterative process described in Algorithm 3 takes place during the cycle solving.

Algorithm for the HP cycle

The Algorithm for operating the HP cycle presented in 1 can be summarized as follows:

- (i) States 3 and 0 are obtained at the HP heat sink, based on T_{source} , $\Delta T_{glide,so,HP}$, $\Delta T_{pp,so,HEX}$, $\Delta T_{sup,HP}$ and considering that the pressure drop Δp_{so} appears between State 0 and State 0". State 3 can only be characterized by its temperature and pressure while state 0 can be fully determined ;
- (ii) States 1 and 2 are obtained at the HP heat source by determining iteratively the pinch-point location as described in Algorithm 3. States 1 and 2 can be fully determined ;
- (iii) State 3 is fully determined based on the isenthalpic expansion of the fluid from state 2 to state 3 ;
- (iv) A sanity check is performed to verify the feasibility of the cycle (the temperature at state 3 must be higher than the temperature at state 0).

Algorithm for the ORC cycle

The Algorithm for operating the ORC cycle can be summarized as follows :

Algorithm 1 Methodology used to solve the HP cycle

-
- (i) DeterminePPsource(Δp_{so}) ▷ Δp_{so} between states 0 and 0"
→ Set (p_0, T_0 , State 1)
 - (ii) IterativePP(State 3, State 2, "Sink") ▷ Iterative determination of pinch-point
→ Set (State 2, State 3)
 - (iii) DetermineState0() ▷ Based on isenthalpic expansion
→ Set (State 0)
 - (iv) VerifyCycle() ▷ Sanity check
-

- (a) States B and C are obtained at the ORC heat source by determining iteratively the pinch-point location as described in Algorithm 3. State B can only be characterized by its pressure : state C can be fully determined ;
- (b) States D and A are obtained at the ORC heat sink by determining iteratively the pinch-point location as described in Algorithm 3. States D and A can be fully determined ;
- (c) State B is characterized based on state A and on the feed-pump efficiency ;
- (d) A sanity check is performed to verify the feasibility of the cycle (the temperature at state D must be lower than temperature at state A and the expander must produce a positive work greater than the work consumed by the feed-pump).

Algorithm 2 Methodology used to solve the ORC cycle

-
- (i) IterativePP(State B, State C, "Source") ▷
→ Set (p_B , State C)
 - (ii) IterativePP(State A, State D, "Sink") ▷ Iterative determination of pinch-point
→ Set (State A, State D)
 - (iii) DetermineStateB() ▷ Based on pump isentropic efficiency expansion
→ Set (State B)
 - (iv) VerifyCycle() ▷ Sanity check
-

Algorithm for the Iterative procedure

As mentioned in items (ii),(a) and (b) , an iterative procedure is needed when no information on the two states of the HEX is available beforehand : indeed, as the pinch-point location has an impact on the temperature at which the fluid is condensed/evaporated, the pressure and the enthalpy of the states inside of the HEX depend on the pinch-point

location.

The iterative process can be described as follows, considering the low-enthalpy side of the HEX (State Low) and the high-enthalpy side of the HEX (State High) at that information on the other states of the cycle to solve are available:

- (1) Make a guess on the temperature at which is evaporated/condensed the working fluid :
 - For the heat sinks, the initial guess is that the pinch-point is located at State Low ; for the ORC heat source, it is that the pinch-point is located at State High ;
 - The following guesses locate the pinch-point for intermediate enthalpy values.

For the value of subcooling, superheating and glide used for the CB, no other case has to be considered ;

- (2) Determine the pressure and the enthalpy of the states within the HEX using the scroll model for the compressor/the expander and information on the other states of the cycle to solve ;
- (3) Determine the temperature of the heat sink/source at the pinch-point based on the glide (which is known) and the guessed pinch-point location assuming a linear profile for the temperature of the heat sink/source as in Equation (A.1)

$$T|_{PP} = T_{sto} - \Delta T_{spread,sto} \cdot (h_{PP} - h_{low}) / (h_{high} - h_{low}) \quad (A.1)$$

where h_{low} refers to the enthalpy of the working fluid at State Low, h_{high} refers to the enthalpy of the working fluid at State High and h_{PP} refers to the enthalpy of the working fluid at the guessed pinch-point location and is such that ;

- (4) Come back to (1) to converge towards the real pinch-point location if last guess was not good.

Algorithm 3 Iterative procedure definition

```

procedure ITERATIVEPP(State Low, State High, SinkOrSource)
  do
    (1) MakeGuessT(SinkOrSource)           ▷ Initial guess depends on the HEX
    (2) DeterminePressureEnthalpyHEX()      ▷ Using Scroll Model
    (3) DetermineTPP()                     ▷ Assuming linear temperature profile
    while (4) VerifyGuess() = False        ▷ Check if pinch-point well located
  end procedure
  
```

Implementation

The Carnot Battery model was implemented using CoolProp in Python. As mentioned in [68], using low-level interface Tabular Taylor Series Expansion (TTSE) methods to calculate thermodynamic properties provides a significant speedup compared to the use of full equations of states without any significant deviations. The use of full equations of state could nonetheless not be totally avoided as some inputs pairs were not supported yet for R1233zd(E) at the time of redaction of this work, but the overall speedup was nonetheless significant.

The elapsed time to run once the overall CB model (setup of the CB + HP cycle + ORC cycle) for a typical set of parameters is $0.62ms$, averaged on 100,000 runs on a portable computer with INTEL(R) CORE(TM) i7-7500U CPU @ 2.70GHz processors. The setup of the CB parameters does not represent a significant part of the computational effort, which is mainly concentrated in the iterative processes where calls to TTSE methods to calculate thermodynamic properties represents most of the time.

Appendix B

Multi-objective optimization problem

Optimization problems for which different objectives are to be optimized simultaneously are called **multi-objective optimization problems**, in which a multi-objective function $\mathbf{f} = (f_1, f_2, \dots, f_m)$ is optimized for a decision vector $\mathbf{x} = (x_1, x_2, \dots, x_n)$ over a feasible set X and under constraints $\mathbf{g} = (g_1, g_1, \dots, g_p)$ on the decision variables.

$$\begin{aligned} \min_{\mathbf{x}} \quad & \mathbf{f}(\mathbf{x}) \\ \text{s.t.} \quad & \mathbf{g}(\mathbf{x}) = 0 \end{aligned} \tag{B.1}$$

Problem of interest without storage systems

In the case of UCLouvain energy system, the multi-objective function is composed of the annual costs and annual greenhouse emissions for operating the energy system $\mathbf{f} = (C_{tot}, GHE_{tot})$:

- The **annual cost** C_{tot} can be defined in function of the annualized investment costs C_{inv} , the annual variable costs C_{var} and the benefits C_{ben} obtained by exporting energy out of the energy system as in Equation (B.2) ;

$$C_{tot} = C_{inv} + C_{var} - C_{ben} \tag{B.2}$$

- Similarly, the **annual greenhouse emissions** GHE_{tot} can be defined based on the heat-related and electricity-related emissions GHE_{Heat} and GHE_{Elec} as in Equation (B.3).

$$GHE_{tot} = GHE_{Heat} + GHE_{Elec} \tag{B.3}$$

If not energy storage system is considered, the decision vector can be reduced to the installed **PV panels** power and the number of installed wind turbines $\mathbf{x} = (PV_{peak}, WT)$, the feasible set X being bounded to maximum power that can be installed due to physical constraints.

As the **CHP** powerplant has been designed to match UCLouvain heat demand all over the year and as it is supposed that the grid capacity is always higher than the demand, the hourly electric and the heat demand are satisfied all over the year.

Alternatively said, defining the energy balance as in Equation (B.4) and whatever the choice of x , the electricity balance ΔDem_{Elec} and the heat balance ΔDem_{Heat} of the energy system cannot be negative.

$$\Delta Dem \triangleq Prod - Cons \quad (B.4)$$

In the case of heat, the demand of UCLouvain is perfectly matched *perfectly matched* ($\Delta Dem_{Heat} = 0$) and no heat exportation are considered in the presented model.

The optimization problem for the energy system can thus be written such as :

$$\begin{aligned} \min_{\mathbf{x}} \quad & \mathbf{f}(\mathbf{x}) = (C_{tot}(x), GHE_{tot}(x)) \\ \text{s.t.} \quad & \Delta Dem_{Heat} = 0, \\ & \Delta Dem_{Elec} \geq 0 \end{aligned} \quad (B.5)$$

Appendix C

Numerical model for the energy system without Carnot Battery

The numerical model used to simulate the operation of UCLouvain energy system over the year is given. The *powerCHP()* function in Algorithm 4 determines the available electric power considering the heat demand and the charge of the storage tanks of the CHP unit at hour h , based on the model proposed in Section 4.2.

The numerical model was implemented in Python using when possible commands from the NUMBA package [69] to accelerate the code execution ; the mean execution time for simulating a year that was obtained on a portable computer with INTEL(R) CORE(TM) i7-7500U CPU @ 2.70GHz processors over 100,000 runs is equal to 0.278 [ms], not taking into account the data acquisition from distant files by the pvlib library and the lecture of the files stored locally (which can take up to 10 [s] but is needed only once to be able to run simulations).

Algorithm 4 Numerical model algorithm used for the energy system without storage

$$C_{PV} = (CAPEX_{an,PV} + OPEX_{fix,PV}) \cdot P_{inst,PV}$$

$$C_{WT} = (CAPEX_{an,WT} + OPEX_{fix,WT}) \cdot P_{inst,WT}$$

$$GHE = 0$$

▷ Compute electricity balance for each hour of the year

for h in range(hours per year) **do**

$$\Delta Dem_{Elec} = Cons_{Elec}[t = h] - P_{Ren}[t = h]$$

$$P_{CHP} = \text{powerCHP}(Cons_{Heat}, h)$$

if $\Delta Dem_{Elec} < 0$ **then** ▷ If overproduction of electricity

$$P_{inj,grid} = \Delta Dem_{Elec}$$

$$P_{purch,CHP} = 0$$

$$P_{purch,grid} = 0$$

else ▷ If underproduction of electricity

$$P_{purch,CHP} = \min(\Delta Dem_{Elec}, P_{CHP})$$

if $P_{purch,CHP} + \Delta Dem_{Elec} < 0$ **then** ▷ If still not enough electricity

$$P_{purch,grid} = \text{abs}(P_{purch,CHP} + \Delta Dem_{Elec})$$

else

$$P_{purch,grid} = 0$$

end if

end if ▷ Compute Costs and Benefits

$$C_{Elec} = (P_{purch,grid} \cdot c_{grid}) + (P_{purch,CHP} \cdot c_{CHP,Elec}[t = h])$$

$$B_{Elec} = (P_{inj,grid} \cdot c_{feedin}[t = h])$$

$$C_{Heat} = Dem_{Heat} \cdot c_{CHP,Heat}$$

$$C += C_{Elec} + C_{Heat} - B_{Elec}$$

▷ Compute GHE

$$GHE_{Elec} = (P_{purch,grid} \cdot f_{grid}) + (P_{purch,CHP} \cdot f_{CHP,Elec})$$

$$GHE_{Heat} = Dem_{Heat} \cdot f_{CHP,Heat}$$

$$GHE += GHE_{Elec} + GHE_{Heat}$$

end for

Appendix D

Non-Dominated Sorting Algorithm description

Processes description

The **elite selection process** is done by dividing recursively the global population into non-dominated (i.e. Pareto optimal) individuals and dominated individuals (*sortNonDominated()* function in Algorithm 5). If selecting the ND non-dominated individuals for the elite selection does lead to an elite population size greater to N , the individuals showing the best 'genetic diversity' are selected to complete the elite population (*distance()* function in Algorithm 5) and the elite selection process stops ; if not, the entirety of the ND individuals are selected to complete the elite population and the elite selection process continues.

The **child population creation process** (*newPop()* function in Algorithm 5) is done based on crossovers and mutations of the characteristics of the elite population.

Algorithm 5 NSGA II algorithm pseudo-code

```
for  $i$  in range( $m$ ) do                                ▷ For each generation of the  $m$  generations
     $D = G_i$ 
     $E_{i+1} = \emptyset$ 
    while size( $E_{i+1}$ ) <  $N$  do                            ▷ Select the  $N$  best-performing individuals
         $ND, D = \text{sortNonDominated}(D)$ 
        if size( $E_{i+1}$ ) + size( $ND$ ) <  $N$  then
             $E_{i+1} = E_{i+1} \cup ND$ 
        else
             $E_{i+1} = E_{i+1} \cup \text{distance}(ND, \text{size}(E_{i+1}))$ 
        end if
    end while
     $C_{i+1} = \text{newPop}(E_{i+1})$                             ▷ Create new population based on the retained elite
     $G_{i+1} = E_{i+1} \cup C_{i+1}$                             ▷ Merge elite and child populations
end for
```

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