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Pumped Thermal Energy Storage based on Organic Rankine Cycles

TESI DI LAUREA MAGISTRALE IN
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Abstract

Increasing demand for sustainable and renewable energy sources it leads to great challenges because of high fluctuating power transmission which creates a significant unbalance between supply and demand. Energy storage is the key factor for the feasibility of clean and sustainable energy by increasing the share of renewable power production. Pumped thermal energy storage (PTES) is an alternate and relatively new technology for storing electrical energy.

In this thesis paper, a comprehensive review of PTES technology and PTES system based Organic Rankine Cycles (ORC) is studied in detail. PTES use waste heat sources as additional thermal energy at a high temperature then the heat sink is generally referred to as thermally integrated pumped thermal energy storage (TI-PTES) to improve the roundtrip efficiency.

In ORC based PTES, the systems comprise of a Vapor compression heat pump (VCHP) and Organic rankine cycle (ORC) as the charging and discharging cycles. This system can be integrated with different thermal energy storage configurations. The storage system includes Sensible thermal energy storage (STES), Latent thermal energy storage (LTES) and Thermochemical energy storage (TCES).

In this study review of different types of working fluid and turbomachinery for PTES were addressed as it is essential to improve the performance of the system. The choice of operating conditions depends on the system configuration, working fluid and turbomachinery. This study aims to provide insights about selection of suitable fluids and components enhancing the overall performance and roundtrip efficiency.

A sensitivity analysis based on Carnot efficiencies is conducted to evaluate the impact of the operating temperature ranges on the performance and efficiency of the PTES system. The aim of this analysis is to address the relationship between the operating temperature limits and efficiency of the system which can be beneficial for designing and optimizing the PTES systems for various applications.

Keywords: Pumped Thermal Energy Storage (PTES), Pumped Heat Electricity Storage (PHES), Organic Rankine Cycles (ORC), Carnot Battery, Thermal integration.

Abstract in Italiano

La crescente domanda di fonti energetiche sostenibili e rinnovabili porta a grandi sfide a causa dell'elevata fluttuazione della trasmissione di potenza che crea un significativo squilibrio tra domanda e offerta. Lo stoccaggio di energia è il fattore chiave per la fattibilità di energia pulita e sostenibile aumentando la quota di produzione di energia rinnovabile. L'accumulo di energia termica pompata (PTES) è una tecnologia alternativa e relativamente nuova per lo stoccaggio di energia elettrica.

In questo documento di tesi, viene studiata in dettaglio una revisione completa della tecnologia PTES e dei cicli organici Rankine (ORC) basati sul sistema PTES. PTES utilizza fonti di calore di scarto come energia termica aggiuntiva ad alta temperatura, quindi il dissipatore di calore è generalmente indicato come accumulo di energia termica pompata termicamente integrato (TI-PTES) per migliorare l'efficienza di andata e ritorno.

Nel PTES basato su ORC, i sistemi comprendono una pompa di calore a compressione di vapore (VCHP) e un ciclo rankine organico (ORC) come cicli di carica e scarica. Questo sistema può essere integrato con diverse configurazioni di accumulo di energia termica. Il sistema di accumulo comprende l'accumulo di energia termica sensibile (STES), l'accumulo di energia termica latente (LTES) e l'accumulo di energia termochimica (TCES).

In questo studio è stata affrontata la revisione di diversi tipi di fluido di lavoro e turbomacchine per PTES poiché è essenziale per migliorare le prestazioni del sistema. La scelta delle condizioni operative dipende dalla configurazione del sistema, dal fluido di lavoro e dalla turbomacchina. Questo studio mira a fornire approfondimenti sulla selezione di fluidi e componenti adatti che migliorano le prestazioni complessive e l'efficienza di andata e ritorno.

Viene condotta un'analisi di sensibilità basata sulle efficienze di Carnot per valutare l'impatto della temperatura operativa sulle prestazioni e sull'efficienza del sistema PTES. Lo scopo di questo analisi è quello di affrontare la relazione tra l'intervallo di temperatura operativa e l'efficienza del sistema che può essere vantaggiosa per la progettazione e l'ottimizzazione dei sistemi PTES per varie applicazioni.

Parole chiave: Accumulo di energia termica pompata (PTES), Accumulo di energia termica pompata (PHES), Cicli Rankine organici (ORC), Batteria di Carnot, Integrazione termica

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1.INTRODUCTION:

1.1 The Need for Energy Storage:

Global electricity generation is highly dependent on conventional fossil fuels such as coal, natural gas, and liquid fuels. This may lead to the depletion of fossil fuels, and they have negative impact on the environment by releasing greenhouse gases (GHG) and other pollutants. The change from fossil fuels to renewable energy sources has been the top priority of our society since global warming became a major problem and climatic changes started to affect the way the world works. To solve this problem many countries tried to find an agreement to reduce carbon emissions. This transition can improve the efficiency of the grid by increasing the share of renewables and offset the need to depend on pollution-emitting peak power plants. The major challenge is to support the transition to be reliable and sustainable energy system by overcoming several challenges such as increasingly growing energy needs and climatic change. However, by Increasing the renewables share it leads to great challenges because of high fluctuating power transmission which creates a significant unbalance between supply and demand. This is the main reason why research about ways to store electric energy produced from renewable sources is of high priority. Energy storage is the key factor for the feasibility of clean and sustainable energy by increasing the share of renewable power production[1]. The rapid scaling up of energy storage systems will be critical to address the hour-to-hour variability of wind and solar PV electricity generation on the grid, especially as their share of generation increases rapidly. Grid-scale storage important for many reasons like short term balancing and operating reserves, ancillary services for grid stability and long-term energy storage for restoring power following a blackout. At the community level, energy storage requires more resilient and flexible energy systems with high levels of energy security through integration of local energy producers.

According to International energy agency (IEA), annual additions of renewable are expected to around 305 GW in the period of 2021-2026 which means that renewable power production is going to increase by 60% in the last five years[2]. To achieve net zero carbon targets, almost the majority of the energy has to be provided by renewable electricity. This increase in renewable electricity will be primarily based on wind and solar. Since the renewables are based on nature may produce less power than required or excess power that may be wasted. So, the most important factor is to store enough renewable energy when there is excess or surplus generation. But we need adequate energy to be stored to satisfy the demand and this can be achieved by building more renewable plants and coupling with efficient energy storage.

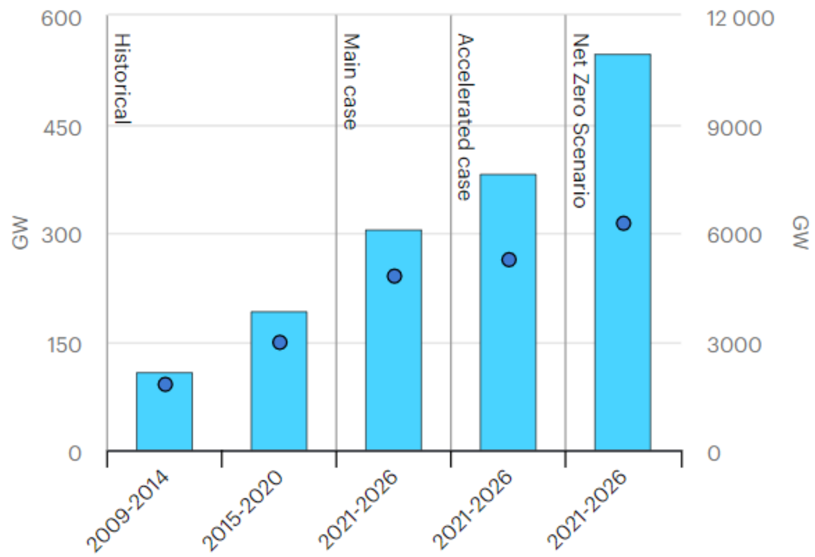


Figure1-1. Average annual renewable capacity additions and cumulative installed capacity to meet IEA Net Zero by 2050 Scenario

Electrical energy storage (EES) refers to the process of converting surplus electricity into a form that can be stored and reconverted to electricity when needed. Such a process enables the electricity to be generated at times of low energy demand, low generation cost and mange intermittent energy sources and to be used at times of high energy demand, high generation cost or when renewable energy sources are unavailable. Different energy storage technologies are available today and each storage system is based on different physical principles. Therefore, they have different performance characteristics such as different roundtrip efficiency, specific costs, energy density, power to capacity ratio, etc. Each energy storage technology can be integrated at different levels like generation, distribution, or transmission. By balancing power grids and saving surplus energy, energy storage represents a concrete means of improving energy efficiency and integrating more renewable energy sources into electricity systems.

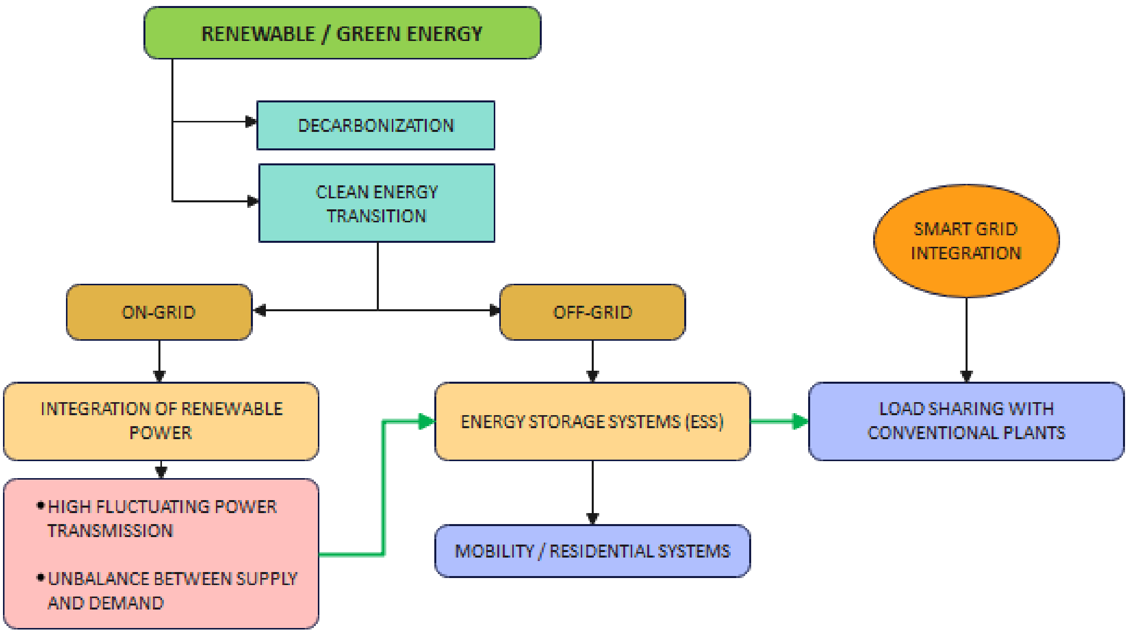


Figure 1-2. Energy transition to renewables with ESS integration

1.2 Energy Storage Systems:

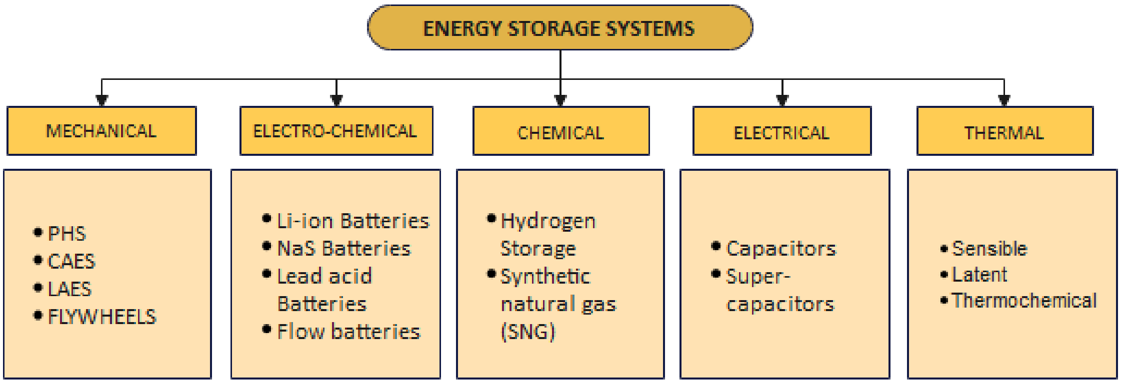


Figure 1-3. Classification of Energy Storage Systems (ESS)

There is a wide range of energy storage systems (ESS) that are commercially available where specific storage technologies have better performance than others and thus being more relevant for certain applications. Energy storage and power management are recently becoming very important as many countries are making a huge transition to renewable sources. Energy storage technologies will be the future to ensure secure and continuous supply to the consumer from a distributed and intermittent supply base both on and off grid applications. According to International energy agency (IEA), the global installed storage capacity is forecast to expand by 56% between 2021 and 2026 to reach over 270 GW by 2026 also 450 GW by 2050[3]. The major reason for this shift is the increasing need for system flexibility and storage around the world to fully utilize and integrate larger shares of variable renewable energy into power systems. The integration of energy storage systems into other sources of energy production will improve the system efficiency. Since all electricity generated is not always utilized, excess energy during off peak times can be stored used to sustain varying power demands at different times. So, adding more renewable sources for electricity production implies that electrical energy storage systems must be improved and investigated further to meet its growing market. ESS can be classified into mechanical, thermal, chemical, electrochemical and electrical storage systems[1], [4].

Different energy storage systems can be used for grid services based on the duration of storage, energy and power density and operating limits. So, the different services can be classified into short term storage or energy managements systems, ancillary services and long-term storage. For short term storage the energy storage system is required to the manage the supply and demand of electricity in the real time which includes some services like frequency regulation, voltage support to maintain the grid stability and reliability of the grid. Supercapacitors, flywheels and super conducting magnetic energy storage (SMES) are some of the suitable systems for short-term services. To support the transmission of electricity on the grid ancillary services are required which include services like black start and maintain frequency and voltage stability in the grid. Long-term energy storage is important for integrating renewable power systems to store electricity for long periods of time and used when it is required. Systems like PHS, CAES, PTES are the most preferred for long term storage services.

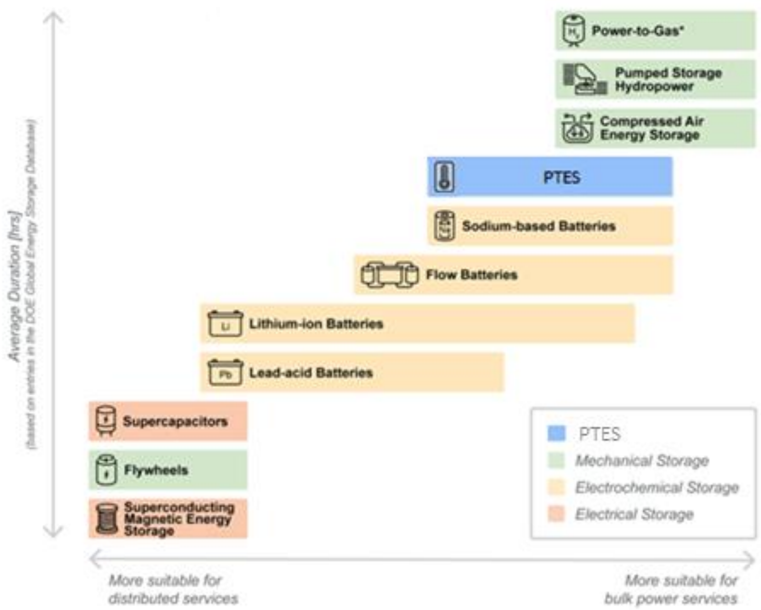


Figure 1-4. ESS classification based on different services and storage time.

1.2.1 Mechanical Energy Storage:

Among the mechanical storage systems, the pumped hydro storage (PHS) system is the most developed commercial storage technology, and it has the large share of energy storage capacity in the world. Other well-known energy storage technologies include Compresses air energy storage (CAES), Liquid air energy storage (LAES) and flywheels. PHS and CAES have longer startup times but provide large storage capacities, but both suffer high capital cost, geographical constraints, and environmental issues.

Pumped hydro storage is stored by pumping water from a lower reservoir to a higher reservoir (hill). Energy is then recovered through a turbine when the water is allowed to flow downwards. It accounts for about 131 GW of PSH capacity representing over 97% of installed global energy storage capacity [5]. PHS have large capacity and high discharge duration. It also has a longer lifetime, but the major disadvantage of this mature technology is the geographical constraint and longer construction period.

CAES stores energy in the form of compressed air which requires large storage volumes so compressed air is often stored in the underground caverns. In conventional CAES, fuel can be added to drive a combustion turbine but in the adiabatic case the heat is transferred from the compression phase and stored in a thermal energy storage [6]. This storage is highly dependent on the availability of underground caverns so only limited deployments.

Liquid air energy storage (LAES) cryogenically cools and liquefies air using off-peak electricity during charging and during discharge the stored liquid air is pumped, heated, and expanded in turbines to generate electricity. LAES stores electricity in the form of temperature potential and later utilizes it when needed. The major drawback is the low round trip efficiency and only practical on a large scale.

Flywheels energy store is done mechanically due to the kinetic energy of the rotor mass spinning at very high speeds. Energy stored in the flywheel can be reused by reducing the speed of the flywheel with a torque, while the kinetic energy is returned to the electrical motor, which acts as an electric generator. There are some major limitations to be considered in this type of storage which is the small capacity and unexpected dynamic loads or external shocks that can lead to failure and also, they are only able to supply power for short durations.

1.2.2 Electrochemical and Chemical Energy Storage:

Electrochemical energy storage devices can make a major contribution for storing renewable energy. Electrochemical energy storage is based on systems with high energy density and high-power density. In the near future both high power density and high energy density are required in the same material. These systems can be mainly classified into two types as Secondary batteries and flow batteries [7].

Secondary batteries like lead acid, Nickel-Cadmium, lithium ion, sodium-Sulphur, metal-air batteries are the most commonly used secondary batteries in the world. Lithium ion dominates the energy market because of its high energy and power density, high efficiency, and longer life cycle. Sodium Sulphur technology is becoming increasingly attractive for large commercial scale energy storage because of high energy density, longer lifetime and almost zero maintenance. This battery can discharge continuously for 6-8 hours. So, sodium sulfur batteries can be used for long term storage services because of high energy density and can operate with high temperature range for long cycle life. Nickel-cadmium batteries have high energy density and low maintenance but disposing of toxic materials like Nickel and cadmium is difficult.

Flow batteries have the potential to store energy for hours or days and it can be classified into redox flow batteries and hybrid flow batteries. The main advantage of the flow batteries is that the power and energy rating design can be done independently which makes them suitable for power and energy related applications.

Supercapacitors are electrochemical double layer capacitors. The two important properties are high capacitance values and the capability to charge and discharge very fast due to high inner resistance. The efficiency is very high, but these are not suitable for long term storage due to their high discharge rate.

In Hydrogen storage, an electrolyzer is used to convert/split water into hydrogen and oxygen using electricity. Heat is required throughout the process because it is an endothermic reaction. Hydrogen can be stored under pressure for nearly infinite periods of time. Methane (Synthetic natural gas) can be synthesized to store energy in pressure tanks or underground tanks. The major drawback of this type oof storage is low efficiency because of conversion losses.

1.2.3 Pumped Thermal Energy Storage:

Thermal energy storage can be classified into three types of heat storage and each storage type consists of different materials with specific properties. Different types of energy storage are sensible heat storage, latent heat storage and thermochemical storage. The storage medium can be solid, liquid or phase change materials (PCM). The thermal storage system capacity is determined by the specific heat capacity and mass of the medium used. The overall efficiency of thermal energy storage is 60-70% but its high energy and lower self-discharge are some notable advantages. They are ecofriendly and the initial investment cost is considerably low which makes them suitable for large energy storage systems. They have no geographical constraint, so it is easy to couple them with existing conventional energy production systems.

According to International Renewable Energy Agency (IRENA), thermal energy storage is estimated to triple by 2030 from 2019 to over 800 GWh[8]. By supporting the shift to renewables, efficiency and greater electrification, TES investments can help to fulfil long-term climate and sustainability goals.

Pumped thermal energy storage (PTES) is an alternate and relatively new technology which is inexpensive and site independent for storing electrical energy and are commonly known as "Carnot Batteries"[9][10]. It is also referred to as Pumped heat electricity storage (PHES) or Compressed heat energy storage (CHEST). The advantages of pumped thermal energy storage over other storage technologies are no geographical constraint, low-cost, high-energy density, and lower self-discharge rate.

Thermal energy storage is utilized by storing heat in hot or cold storage medium under different conditions. This stored heat is kept in a insulated reservoir and can be used later to generate electricity in industrial and residential applications. Thermal storage systems are used to act as an intermediary between thermal energy demand and supply, making them crucial for the integration of renewable energy sources.

Technologies	Power Rating (MW)	Storage duration (ms-hrs)	Energy Density (kWh/m3)	Power Density (kw/m3)	Efficiency (%)	Self-discharge rate (%)	Service life (years)
PHS	30-5000	>24 hrs	0.05-2	0.5-1.5	70-85	-	30-60
CAES	5-1000	>24hrs	0.4-20	0.5-10	40-80	-	20-50
LAES	1-300	1-6 hrs	50-200	0.1-1.5	45-70	1-2	20-40
FLYWHEELS	0.1-20	ms- < 1 hr	20-80	800-2000	70-90	-	15-20
Li-ion Battery	0-100	min- hr	150-500	50-5000	80-90	0.1-0.3	5-15
Na S battery	0.05-34	1-24 hrs	150-280	50-300	70-90	0.05-20	5-20

Leadacid Battery	0-40	sec-hr	50-100	10-700	65-90	0.1-0.3	5-15
Flow Battery	0.05-3	sec-10hrs	20-70	0.5-34	60-85	0.1-0.2	5-20
Hydrogen	0.1-50	sec-24hrs	600-1200	5-800	20-66	10	10-20
SMES	0-10	ms-min	0.2-13.8	1000-4000	80-95	10-15	20-30
Super-capacitors	0.02-10	ms-hr	1-35	1000-5000	60-90	5-40	8-20
PTES	0-300	>24 hrs	15-1200	-	> 90%	1-2	15-50

Table 1-1. Technical properties of different Energy storage systems (ESS) [11], [12]

2.Concept of Pumped Thermal Energy Storage:

The Pumped thermal energy storage is an energy storage technology which transforms electricity into heat and store it as thermal energy in a storage medium in an effective and inexpensive way and transform it back into electricity when needed [9][10]. The input electricity is used to establish a temperature difference between two environments which is the higher and lower temperature reservoirs. In general, this process includes three phases which are charging phase, storage, and the discharge phase.

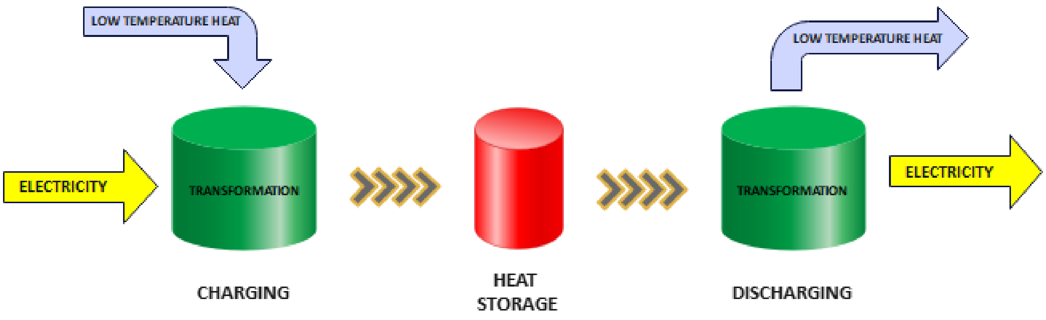


Figure 2-1. Technological concept of PTES

In the charging phase, electricity is stored as thermal energy using heat pumps or with direct resistance heating to establish a temperature difference between the hot and cold reservoir. When a heat pump is used for heating the storage, the heat pump requires low temperature heat as well as electricity to work efficiently. Because of this reason direct resistance heating could be another option but with disadvantage of less cost effective [13].

In the storage phase, the heat is stored in the storage medium in an effective way and ideally with minimum energy losses. The heat storage can either be sensible, latent, or thermochemical storage or a combination of these storages. One of the emerging concepts for TES are phase change materials (PCM) which are latent energy storage materials. The major advantage of PCM is discharge of a large volume of energy at constant temperature which could lead to smaller and lower storage system [11].

During discharge phase, the stored thermal energy is used to generate electricity using heat engines like Brayton cycle, rankine cycle. The heat engine absorbs heat from the high temperature reservoir and converts the heat into electricity.

Whatever the thermodynamic cycle, the PTES charges thermal exergy into the thermal reservoir which moves heat from a thermal reservoir at a lower temperature to higher temperature. The thermal exergy is conserved for the required amount of time in the thermal reservoir. During this phase, thermal exergy losses naturally occur. Finally, the PTES discharges the stored thermal exergy by converting it into electricity with a heat engine.

The main aim of the Carnot Battery is to store heat and produce electricity, but it can be also utilized as storage unit which could be used for industrial processes like steam generation. In addition, as a by-product of this process we obtain low temperature heat so different applications can be combined with this technology depending on the system integration and economic feasibility.

2.1 Pumped Thermal Energy Storage Classification:

After the basic configuration discussed in the previous chapter with respect to external heat sources and heat sinks, the PTES system can be further classified with respect to the thermodynamic cycle used for the charge and the discharge cycle. So, the PTES are mostly classified in the literature into Brayton based PTES, Rankine based PTES.

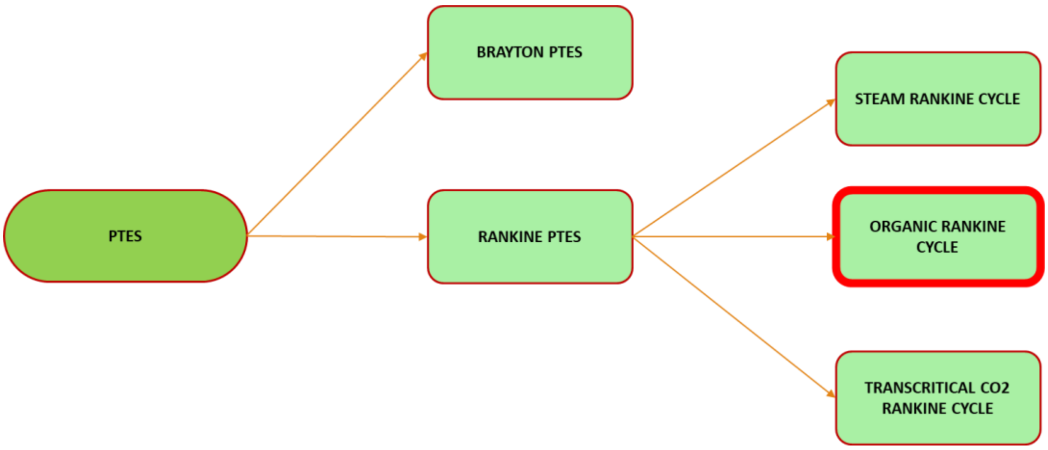


Figure 2-2. Classification of Pumped Thermal Energy Storage

2.1.1 Brayton Based Pumped Thermal Energy Storage:

In Brayton based PTES, the system comprises of a brayton heat pump and a brayton heat engine for the charging and discharging cycles. The brayton heat pump is based on an inverse brayton cycle. The working principle of a brayton based PTES is simple and it may have both a hot and cold temperature reservoir.

During the charging mode, the working fluid absorbs heat from the cold reservoir before it enters the compressor, and the hot reservoir is heated up to create a temperature difference between the two reservoirs. During the discharge phase, the working fluid is compressed to a high pressure by utilizing heat from the hot storage and then it is expanded to produce electricity. When compared to an expander, the compressor uses more energy which should be taken into consideration for utilizing these systems.

The Brayton based PTES layout consists of a compressor, an expander and two heat exchangers for both the charging and discharging cycles. Sensible thermal storage (STES) is the most used storage configuration for this system to minimize the temperature difference between the gas and the storage material and also because of no phase change. The advantage of brayton PTES is that both the hot and cold heat transfers are mostly sensible heat and packed bed STES is very common because of their low costs, large heat capacity and high cyclic temperature variations [14]. In literature, some authors propped to use recuperated brayton cycle and two storage tanks for better performances.

Liquid STES can store heat only up to 300˚ C thus it allows for a small temperature variation between compressor outlet and turbine inlet. Because of this small variation it may lead to a low back to work ratio which is defined as the ratio between the power delivered by turbine and power required by compressor which results in low efficiency. So, a solid STES is preferred to store heat at high temperature with packed bed configuration.

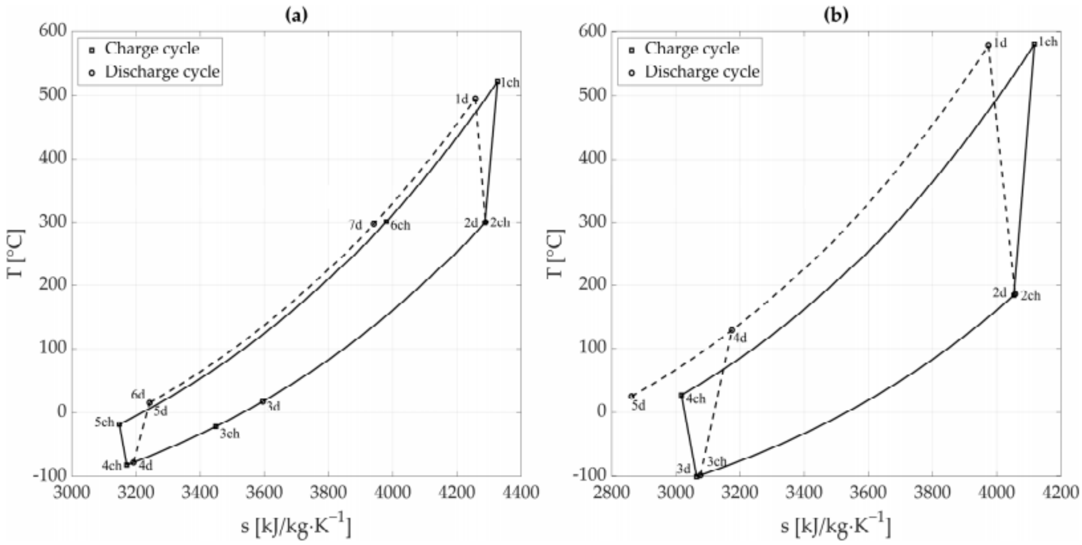


Figure 2-4 Brayton based PTES T-s diagram with charging and discharging cycles: a) Solid based thermal energy storage b)Liquid based thermal energy storage[15]

Liquid based storage requires internal regeneration to achieve high temperature and improve the performance whereas solid based storage doesn't need internal regeneration. But however solid based operated with high pressure ratio which increases the turbomachinery cost and liquid based storage uses a recuperated cycle to achieve same efficiency for lower pressure ratio[15].

For brayton PTES both dynamic and volumetric machines can be used. Reversible machine can be more easily adopted if a volumetric machine is used with low capital costs but with a decrease in performance [13]. Usually, Brayton PTES operate at a very high temperature range (above 200˚C) so it is not possible to integrate low temperature or waste heat sources.

The general values of roundtrip efficiency for brayton based PTES in the literature is reported in the range 60 – 70 % but since Brayton based systems are highly sensitive to the efficiency of the compressor and expander much lower values are obtained for efficiencies lower than 90 %and even less roundtrip efficiencies for systems considering compressor and expander efficiencies of 80% [16]. Despite low efficiency, the brayton PTES have high energy density with low capital costs and is an economically feasible application.

Recuperated Brayton cycle is also a possible configuration which can be integrated with a packed bed storage system. From the thermodynamic point of view high temperature heat transfers from the storage increases the efficiency of the cycle by reducing the required storage capacity so I that way packed bed storage system is advantageous. The major problem with a recuperated brayton cycle is the internal heat exchanger which not only increases the cost but also leads to additional exergy losses. To overcome this problem the pressure of working fluid can be increased to reduce the size of internal heat exchanger but still this increase the cost involved in the pressure vessel [14].

2.1.2 Rankine Based Pumped Thermal Energy Storage:

In Rankine based PTES, the systems comprise of a Vapor compression heat pump (VCHP) and rankine cycle as the charging and discharging cycles. Rankine cycles are characterized by phase change of the working fluid which can store both sensible and latent heat. With rankine cycles it is possible to achieve high energy densities and store it a much lower temperature (usually below 200˚C) [11][16] when compared to brayton cycle. In this type of configuration, the thermal losses are less when compared to brayton cycle and we can use latent heat energy storage (LTES) configuration with phase change materials (PCM) for storing heat which show large energy densities which have large storage densities. Rankine PTES also allows the use of pressurized water tanks which is a cost-effective storage system whose operation and is limited to temperatures under 140˚C[16][19]. Rankine based PTES can be classified into steam rankine cycles, Transcritical rankine cycles and organic rankine cycles.

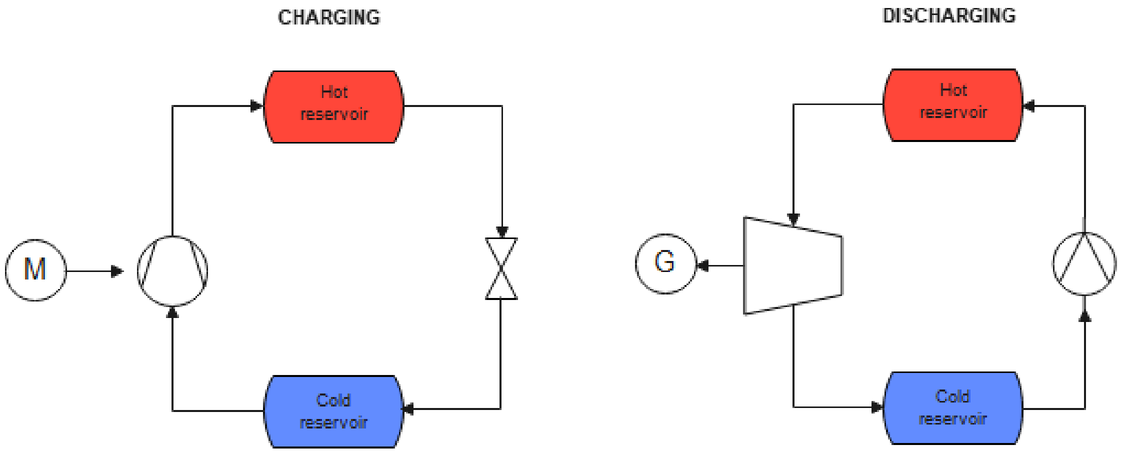


Figure 2-5. Classical Rankine based PTES with VCHP in the charging phase and Rankine cycle in the discharging phase.

Based on steam rankine cycle

Steam rankine based PTES systems involve complex design but it is capable of achieving roundtrip efficiencies around 72.8% [19]. As water is used as the working fluid and in the charging phase because of low saturation pressure at ambient temperature the heat pump cannot function with a single stage compression which decreases the performance. But however, by using multistage compression and by adopting binary cycle configuration using ammonia as bottom cycle to boost up the evaporation temperature water up to 80˚C. Then the water is compressed in multiple stages to attain high temperature and after each compression stage superheated steam is mixed with liquid to enter the next compression stage with same pressure. Multistage compression can also be adopted for ammonia bottom cycle. The latent storage unit is charged by the condensing steam and the condensate is subcooled in the sensible storage unit.

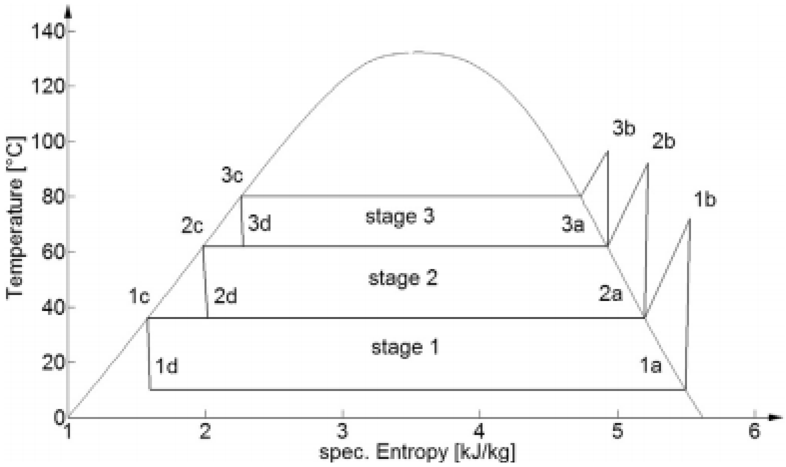


Figure 2-6 T-s diagram for ammonia compression cycle for conventional rankine cycle based PTES [19]

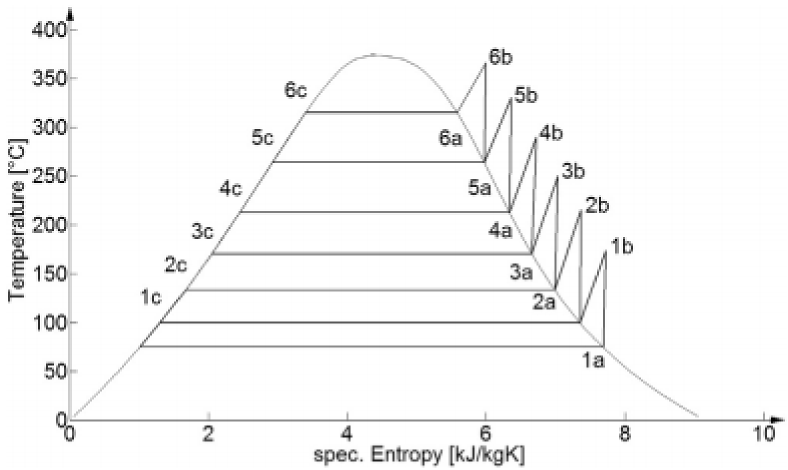


Figure 2-7 T-s diagram for steam compression cycle for conventional rankine based PTES [19]

In the discharge phase stored heat is utilized by the working fluid and since water is a working fluid wet expansion takes place, so superheating is adopted, and the expansion process is divided into two stages.

Multiple compression and expansion stages and binary cycle configuration the system configuration is very complex and because of this reason conventional steam rankine cycles are not so efficient for PTES systems despite good performances.

Transcritical Rankine cycles (CO2)

Transcritical rankine based PTES use working fluids with low temperature critical points and lower pressures. Transcritical PTES configuration uses carbon dioxide (CO2) as the working fluid. Working with CO2 and using Transcritical cycle eliminates the need of a latent heat storage unit since in this case there is no evaporation at constant temperature. So, configuration using pressurized liquid water storage on the hot storage can minimize the cost.

During charging, low temperature heat can be used to preheat the working fluid and then after compression heat is transferred from the CO2 to the pressurized liquid water on the hot side. After internal heat transfer CO2 expands in the turbine and heat from ice storage is used to evaporate the CO2. During discharge, heat transfer from storage is used to heat CO2 before expanding in the turbine and then it condensed in the ice storage.

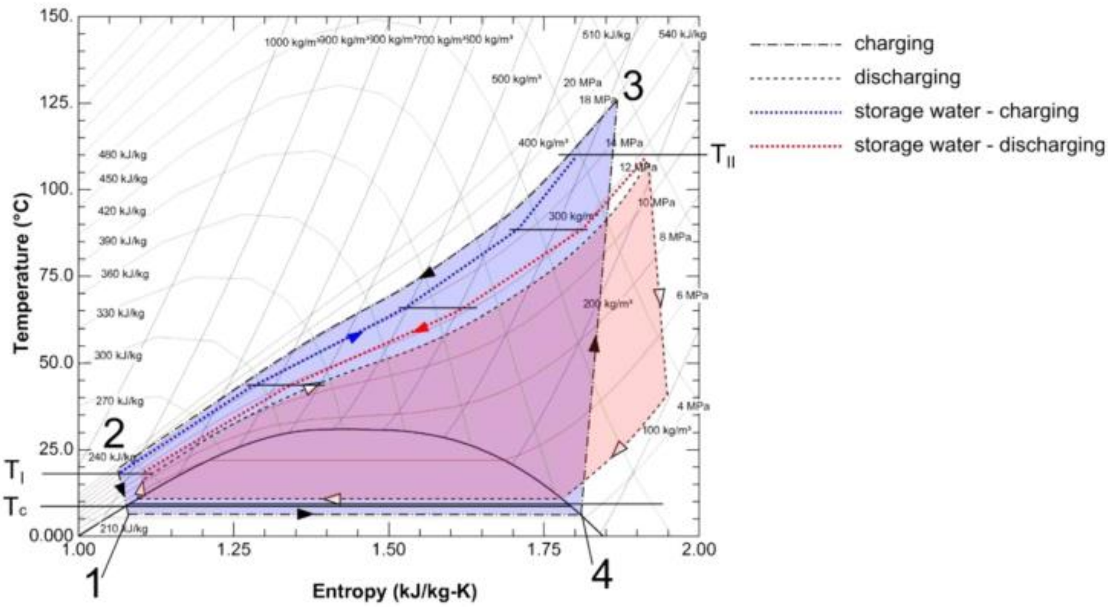


Figure 2-8 T-s diagram Transcritical CO2 rankine based PTES [20]

The advantages of this type of system are mainly related to the low temperature requirements, low cost, and good thermodynamic performance of the water as storage system, low environmental impact of the working and storage fluid. The main disadvantage is that during the Transcritical phase change of CO2 has a big chance of the specific heat and therefore particular attention must be paid to the design of the CO2 water hot heat exchanger. For low temperature range it is better to store heat in pressurized water tank with good heat transfer coefficients which reduces the temperature difference and the pressure losses.

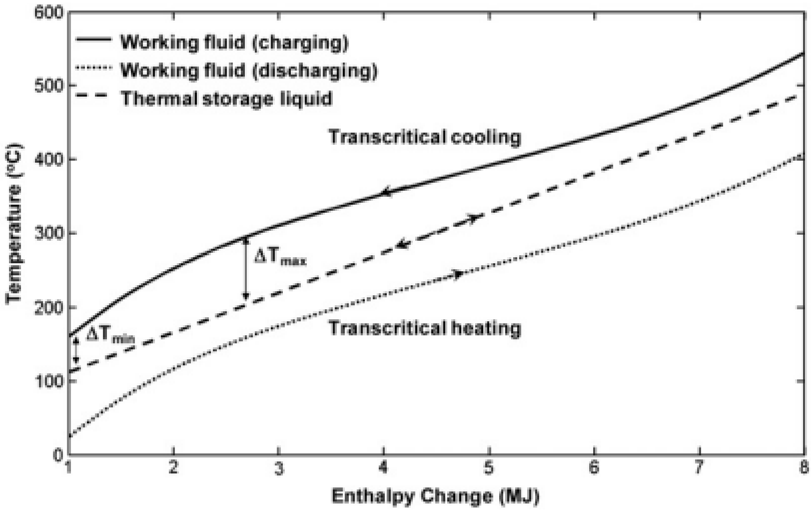


Figure 2-9 T-Q diagram Transcritical rankine based PTES [20]

The hot storage can be made up of one or more sensible heat storages which can work with temperature greater than 100˚C [21][22]. And also, as the specific heat of fluid strongly varies during the phase change the temperature difference in the heat exchanger varies which causes losses or irreversibilities. By adopting multiple storage tanks, the mass flow rate can be increased or decrease according to the variation of temperature during phase change process. This system is capable of reaching efficiencies up to 50-70 %

Organic Rankine Cycles (ORC)

Subcritical rankine based PTES use organic fluids as the working fluid as they can be used under pressure at ambient temperature avoiding the use of costly and bulky equipments. The saturation curve of the organic rankine cycles usually fit well the heat source and sink temperatures ranges used. In the charge phase, VCHP is used to compress the working fluid and store heat in high temperature reservoir and in the discharge phase organic rankine cycle is used to generate electricity by utilizing the heat from the hot reservoir.

Organic rankine cycles show poor performances without thermal integration as the roundtrip efficiencies are reported to be less than 50 % [23][24]–[28] except the cascaded subcritical rankine cycles which showed roundtrip efficiency of 70%.

Thermally integrated organic rankine PTES on the other hand show much better performances when used with low temperature heat sinks and satisfactory temperatures differences between the source and the sink. Roundtrip efficiency from 70% to 130% are reported in the literatures [13], [17], [29]–[31] for waste heat temperatures below 120˚C. Such an improvement in the performances of the ORC based PTES leading to very high roundtrip efficiencies is actually the main reason for the extensive research in this type of PTES systems.

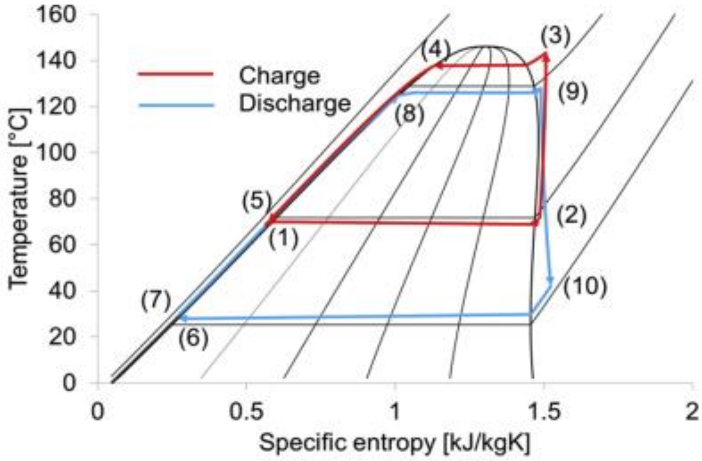


Figure 2-10 T-s diagram of ORC based PTES system [39]

The classical or base configuration are the most common in the literature but any studies considered regenerated or recuperated configuration of VCHP and ORC cycles which sub cools 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 [23]–[25], [32].

The internal regeneration for a cycle is expected to improve its performance at a reasonable cost increase and becomes more common for both high temperature VCHP and ORC. Also some authors suggest other configurations like 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 [33].

PTES	Power (MW)	Temperature range (˚C)	Energy capacity (MWh)	RTE (%)	Fluids	Reference
Brayton	5-100	-150-1000	16-600	50-80	Argon, Nitrogen, Air	[34], [35]
HP/ORC	1-10	15-180	up to 40	60-130	R1233zd(E), Butene	[13], [29]
HP/Transcritical	1-50	-25-150	10-200	50-75	CO2	[20], [36]
HP/Steam rankine	1.5	10-80(NH3) & 75-315(Water)	up to 10	72.8	NH3/Water	[19]

Table 2-1. Characteristics of different PTES configurations from literatures

Table 2-2 Commercial Development of Rankine Based PTES systems[37]

Company, System	Charging Method	TES	Discharging Method	Power Output	Storage Capacity/ Duration	Roundtrip Efficiency	State
Siemens Gamesa, ETES	Resistance heaters to air	Volcanic rock bed~600 °C	steam Rankine cycle	units to 100s MW	24 h	25% to >40%	Demo
RWE, Store2Power	Resistance heaters	molten salt	steam Rankine cycle	100s MW	hours	~40%	n.a.
E2S Power	Resistance heaters	Graphite-aluminium alloy at 700 °C	steam Rankine cycle	1–100s MW	hours	25–40%	lab proof of concept
Spilling	Steam compression & liquefaction	Saturated water (steam accumulators)	Steam expander (steam engine)	up to MW	hours	n.a.	n.a.
GE, AMSESS	CO_2 Brayton + el. heating	Molten salt, water tank	Steam cycle with extraction	20–100 MW	8 h	42–62%	Concept
Consortium CHESTER	Heat pump (organic fluid)	PCM and water	ORC	MW scale (8 kW exper.)	hours to days	n.a.	lab proof of concept
Climeon	Heat pump (organic fluid)	water (e.g., district heating system)	ORC	80 kW to MW	hours	25–60%	concept with existing ORC
TC Mach	Heat pump (organic fluid)	Stone dust TES	ORC	kW	hours	n.a.	construction of proof of concept
Future Bay	Heat pump (organic fluid)	water (hot) and PCM (cold)	ORC	10s kW	hours	n.a.	Demo
Highview	Air liquifaction	Liquid air + other TES	Vaporization, expansion turbine	50–350 MW	about 6	60–70%	Pilot, full scale construction
MAN/ABB, ETES	CO_2 heat pump	120 °C water + cold (ice) storage	CO_2 Rankine cycle	several MWe	~5 h	~45%	lab demo
Echogen, ETES	CO_2 heat pump, fluidized bed heat exchange	Sand (hot) and ice (cold)	CO_2 Rankine cycle	25 MW	250 MWh	~60%	design
Energydome, CO_2 battery	CO_2 compression & liquefaction	Liquid CO_2 + other TES	Vaporization, expansion turbine	10–80 MW modules	20–200 MWh	77%	Pilot construction

2.2 Roundtrip Efficiency

The power-to-power ratio or roundtrip efficiency is defined as the electrical energy output during discharge and the electrical energy input during the charge phase. It is the key parameter of energy storage plants which defines the overall efficiency of the system.

$$\eta_{RTE} = \frac{W_{el,discharge}}{W_{el,charge}}$$

It can be also expressed as: $\eta_{RTE} = \eta_{charge} \cdot \eta_{storage} \cdot \eta_{discharge}$

Where η_{charge} denotes the amount of heat stored in the system per unit of electricity consumed, $\eta_{storage}$ denotes the storage duration per unit of initial heat stored, . $\eta_{discharge}$ denotes the amount of electricity generated from the storage.

2.3 Thermally Integrated Pumped Thermal Energy Storage:

The roundtrip efficiency of the most storage systems is often bounded to unit factor in the ideal case. Consider a PTES system with two isolated reservoirs at different temperatures between which heat is transferred during the charging and the discharging cycles. The roundtrip efficiency is then equal to unity if there is no temperature difference ΔT between the hot and the cold sides during the heat exchange and asymptotically tends to one for ΔT > 0 [38].

However, if only one reservoir is used then an external heat source, or an external heat sink must be considered. In this case additional thermal energy is used to produce electricity thereby increasing the roundtrip efficiency. As the roundtrip efficiency does not consider the input additional thermal energy values higher than unity can be achieved [38][39].

Pumped thermal energy storage using waste heat sources as additional thermal energy at a high temperature then the heat sink is generally referred to as thermally integrated pumped thermal energy storage (TI-PTES)[11] [15][17]. This can be further categorized into "hot storage" configurations (using high temperature reservoir) and "cold storage" (using low temperature reservoir).

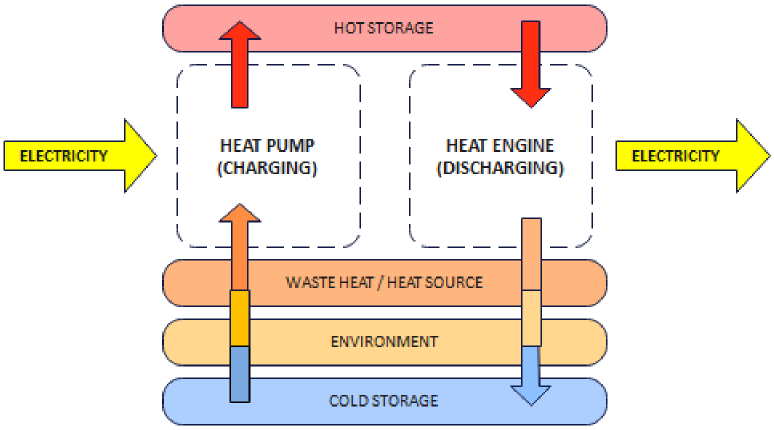


Figure 2-11. Working principle of PTES with different heat sink and heat source configurations.

Several thermally integrated PTES (TI-PTES) configurations have been proposed and studied in the literature. In PTES at least two reservoirs are required but in TI-PTES only one is generally used. In this case, high temperature or low temperature reservoir is considered.

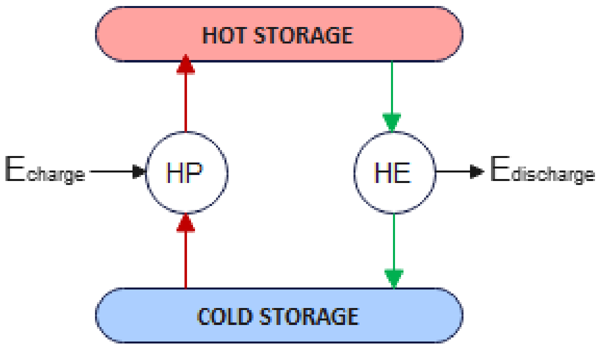


Figure 2-12. Standalone PTES configuration

In the hot storage configuration (hot TI-PTES), waste heat or additional thermal energy source is used in the charging phase to increase the temperature difference. This allows the power cycle to increase its roundtrip efficiency by working with a higher temperature difference. For the charging process electricity is utilized to run the compressor of HP which increases the temperature and pressure level of the working fluid. Thermal energy is provided to the hot storage through the condenser and then the working fluid flows through the expansion valve. The waste is utilized to heat up the working fluid in the evaporator. The major difference is in the charging phase with hot storage configuration and the discharge phase is the ideal ORC which uses the temperature difference between the hot storage and the air to run the turbine to produce electricity.

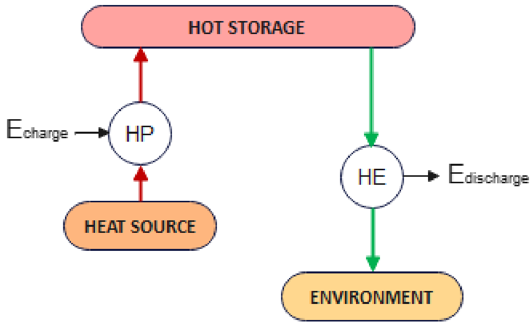


Figure 2-13. Hot Storage TI-PTES configuration

In the cold storage configuration, thermal energy is stored at a temperature lower than the ambient. So, this makes the power cycle work efficiently again with a higher temperature difference. The charging process utilizes electricity to run the compressor. The condenser releases the thermal energy to the air/ambient and then the working fluid enters the expansion valve which retrieves thermal energy from the cold storage. In the discharge phase the pump provides a flow of a certain amount to the evaporator and the waste heat is utilized to evaporate the working fluid and electricity is produced from the turbine.

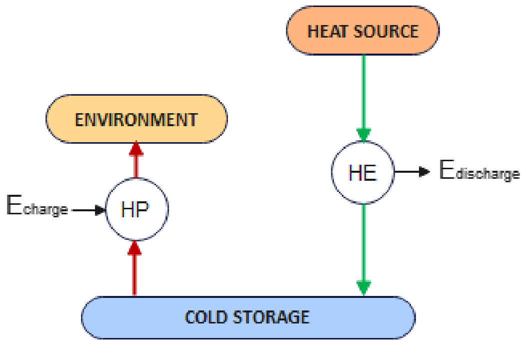


Figure 2-14. Cold storage PTES configuration.

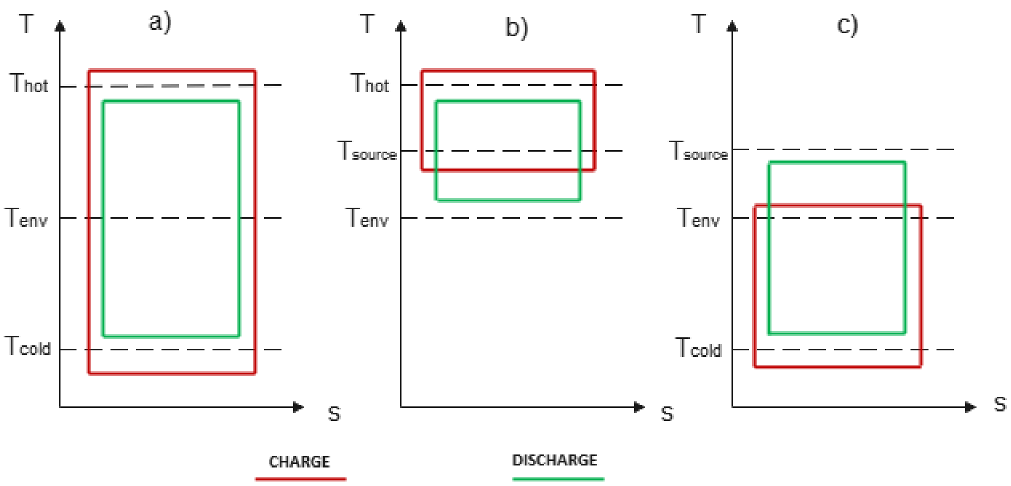


Figure 2-15. Representation of working of PTES configuration with Carnot cycles. a) Standalone PTES, b) Hot storage TI-PTES configuration, c) Cold storage TI-PTES configuration

In TI-PTES systems, the hot storage configurations are the most common in the literature because hot storage configurations are considered to be more effective than the cold storage option. However, for cold storage configurations there are practical considerations that can be convenient like access to cheap, stable, and reliable PCM for thermal energy storage.

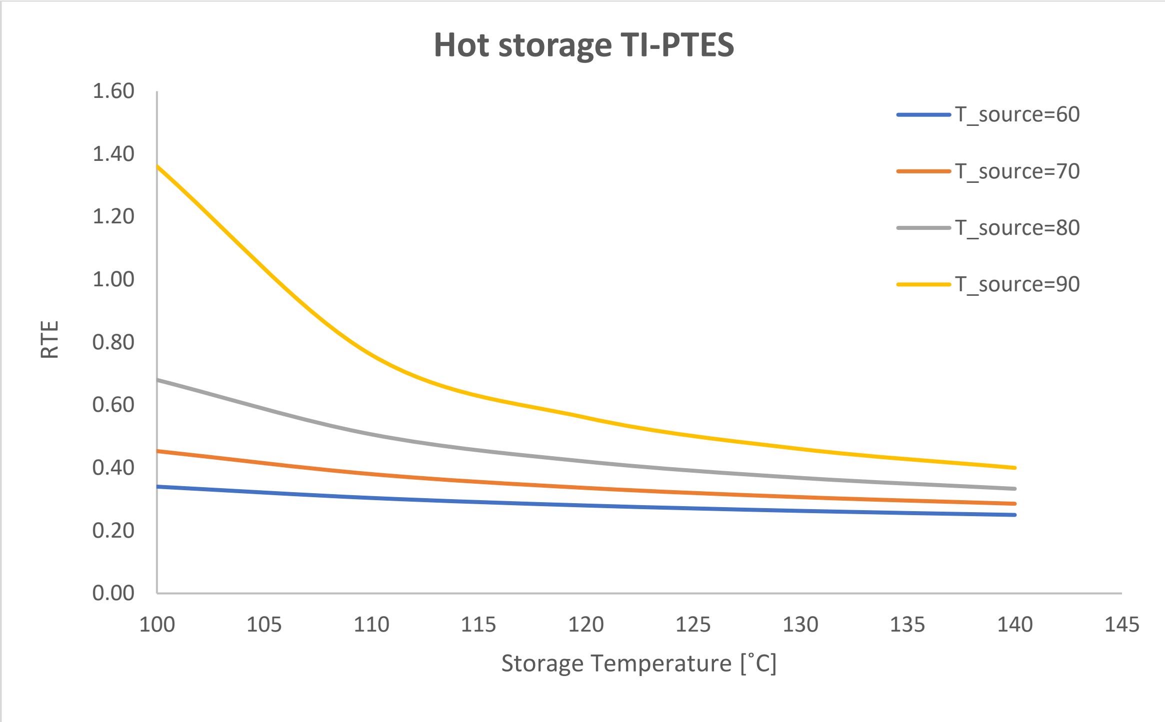


Figure 2-16. Roundtrip efficiency as a function of Storage Temperature for varying heat source (Hot storage TI-PTES)

The limitation of the roundtrip efficiency is derived from the Carnot efficiencies of the heat pump and the ORC process. Second law efficiency in the range of 50% to 70% can be assumed depending on the size of the plant and its complexity[18] to consider the exergy losses. By considering the second law efficiency the deviation between the real process and the ideal Carnot efficiency can be equaled.

The COP of the heat pump depends on the heat source temperature as it acts as the sink temperature for the heat pump and the hot storage. The ORC efficiency depends on the hot storage and ambient temperature as the heat is utilized from the hot storage and expansion takes place. The condition for hot storage TI-PTES is $T_{source} > T_{ambient}$ and $T_{storage} > T_{ambient}$.

$$COP_{HP,\,TI\,integration} = \eta_{\,\mathrm{II}} \left(\frac{T_{hot\ Storage}}{T_{hot\ storage} - T_{source}} \right)$$

$$\eta_{ORC} = \eta_{\,\mathrm{II}} \left(1 - \frac{T_{ambient}}{T_{hot\ storage}} \right)$$

For the Cold storage TI-PTES, the COP of the heat pump depends on the ambient temperature and the cold storage as the heat is released to the ambient and the thermal energy is stored in the cold storage. The ORC depends on the cold storage and the heat source as it utilized to evaporate the working fluid. The condition for hot storage TI-PTES is $T_{source} > T_{ambient}\ and\ T_{storage} < T_{ambient}$.

$$COP_{HP,\,TI\,integration} = \eta_{\,\mathrm{II}} \left(\frac{T_{ambient}}{T_{ambient} - T_{cold\ storage}} \right)$$

$$\eta_{ORC} = \eta_{\,\mathrm{II}} \left(1 - \frac{T_{cold\ storage}}{T_{source}} \right)$$

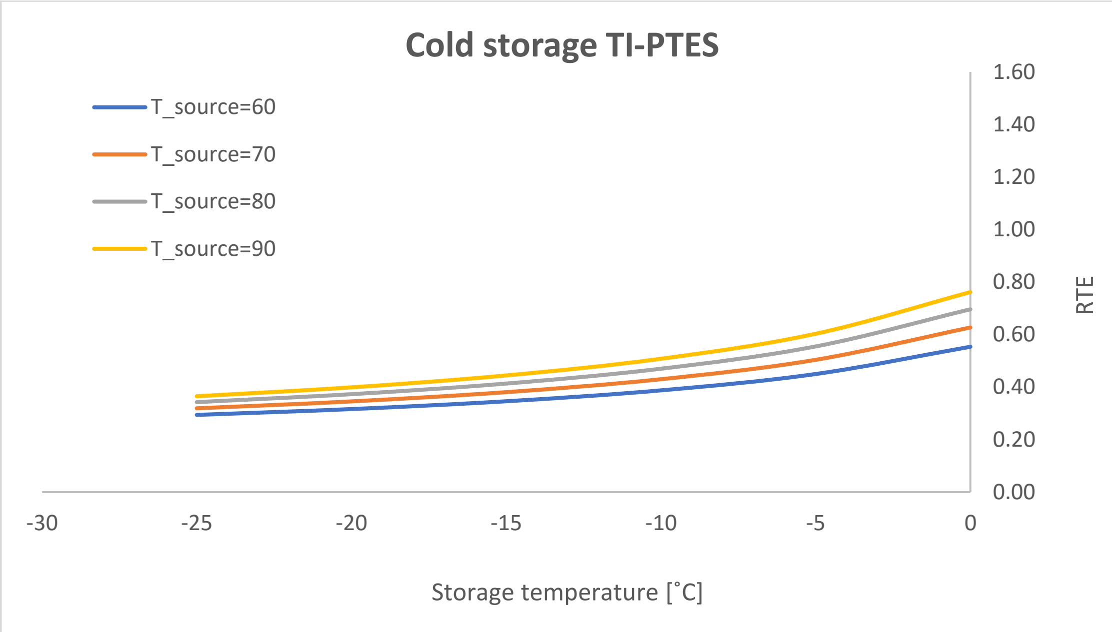


Figure 2-17. Roundtrip efficiency as a function of Storage Temperature for varying heat source (Cold storage TI-PTES)

The performance of TI-PTES is particularly remarkable for high temperature differences ΔT between the heat source and the heat sink for hot storage configurations [39]. For exploiting such waste heat or low temperature heat sources, the most suitable thermodynamic cycle is the rankine cycle as they can operate with better performance at temperatures lower than 200˚C. Also brayton and other hybrid systems can have better performances but for that they had to function at a very high temperature and such systems are not benefited by low temperature heat sources [17].

With the addition of thermal integration in PTES systems we can not only increase the performance but also effectively combine different energy sectors depending on the system integration and cost factors.

3.TYPES OF THERMAL ENERGY STORAGE SYSTEMS

The important part of the system is thermal storage and in this chapter the overview of different thermal energy storages (TES) will be summarized.

TES systems can be mainly classified into three types based on the principal mode of thermal energy storage:

- Sensible thermal energy storage
- Latent thermal energy storage
- Thermochemical energy storage

Besides the ability of suitable system integration, TES systems must fulfill the following technical requirements [40]:

- High volumetric energy density
- Good heat transfer performance between the working fluid and the TES
- Stability and safety over the lifetime of the application
- Low exergy losses due to heat transfer and heat loss with the environment
- Low cost per unit of exergy stored.

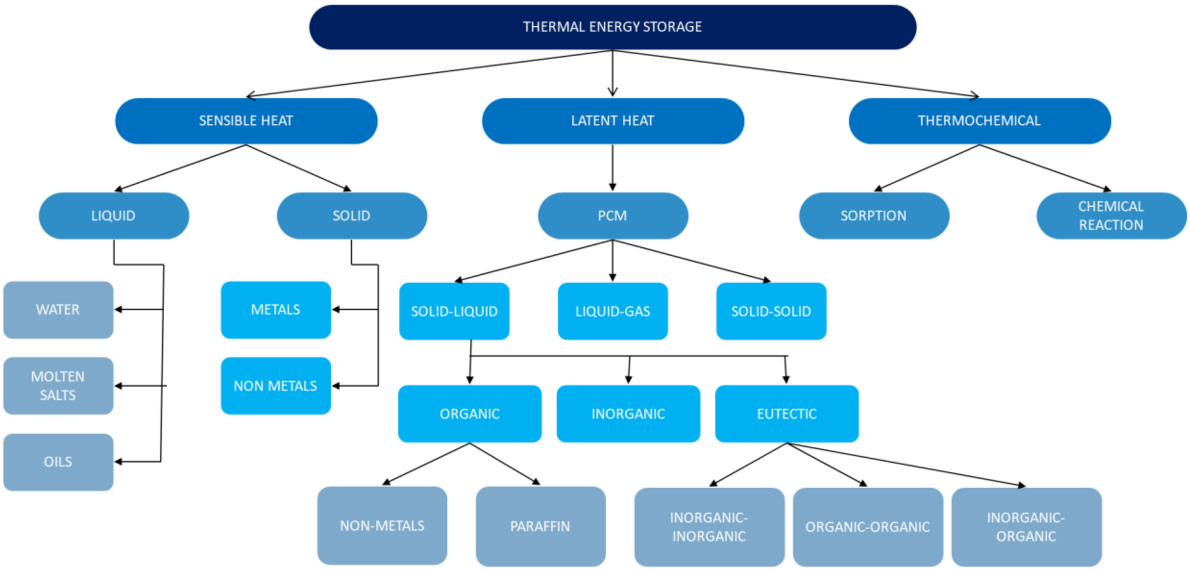


Figure 3-1. Classification of different Thermal energy storages.

3.1 Sensible Thermal Energy Storage:

Sensible thermal energy storage (STES) is the most known and developed method for storing heat. Energy is stored as a temperature difference based on specific heat capacity of the material. The temperature of the material varies and does not undergo any phase change during charging and discharging cycles either in solid or liquid. Heat is absorbed increasing the temperature of the medium depending on the mass of the material (m), specific heat capacity (cp) and the temperature change:

$$Q = \int_{T_i}^{T_f} mc_p \, dT$$

Where T_f and T_i are the final and initial temperature of the medium respectively. Therefore, to have high energy density the TES material should have high specific heat capacity and density. The major advantage of using this type of storage is the use of cheap and safe storage materials and the only disadvantage is the low energy density 20-200 kWh/m^3which differ depending on the temperature differences.

The storing media used for sensible heat storage applications can be subdivided into solid and liquid, but liquid storage medium has the advantage that the same medium can be used for heat transport and storage. This removes the need for a heat exchanger on the storage side.

Water is the most used application mainly because it is a cheap option and has a high specific heat capacity. The only limit is the temperature range where for non-pressurized cases the maximum storage temperature of 100 ˚C makes the usage of water as storage material not applicable for most PTES systems but until 200˚C [41].

Above 100 ˚C synthetic or diathermic oils and molten salts are used. Oils have a range of operation from 200-400 ˚C depending on the type and to increase the maximum temperature the storage system could be pressurized but this increases the overall costs. Molten salts are used because they are liquid at atmospheric pressure in the range of 300 to 600 ˚C while the upper limit is the decomposition limit of the salt, and the lower limit is the freezing temperature.

Typically, the molten salt consists of a mixture of 60% sodium and 40% potassium nitrate (NaNO3/KNO3) and is widely used in CSP sector. These salts are usually in solid phase at ambient temperatures so in order to keep in liquid form it is maintained above their melting points. The only drawback of molten salt is that they need to be kept at a temperature higher than the freezing point and therefore safety heating system are required[42].

Water tank thermal energy storage (WTTES) is the simplest and most widely used form of storage. For small scale applications the tank size is small with few hundred meters and for large scale applications big tanks of around thousand hundred meters. In underground thermal energy storage heat is stored underground in artificial pits or in aquifers which includes aquifer thermal energy storage (ATES), borehole thermal energy storage (BTES) and pit thermal energy storage. The main advantage is large amount of energy can be stored across seasons.

Solid STES has the advantage of being cheap and have a very high density and high thermal conductivity although they have a lower specific heat capacity compared to fluids. STES design generally have segmented and radial flow packed beds which is the packed bed of solid materials or concrete blocks inserted with heat transfer tubes[11]. Bed segmentation with smaller particles can help achieve higher storage efficiency and energy density. However, the capacity of charging and discharging solid media is limited by the heat transfer of the heat transport fluid and the solid. Therefore, higher heat exchange areas or higher fluid flow rates might be required for increasing the charging rate capacity which increases investment and operational costs.

The main features of sensible energy storage are its low cost and use of widely available non-hazardous materials. However, compared with other alternatives, it has low heat density, requires extensive insulation for high temperature applications.

3.2 Latent Thermal Energy Storage:

In latent thermal energy storage (LTES) phase of the storage material takes place and the energy is stored based on the latent heat of fusion. The LTES materials are commonly known as phase change materials (PCM). Although solid-solid and liquid-gas materials exist, solid-liquid are the most used PCM as they have lower volume variation and relatively high latent heat. LTES stores energy by absorbing and releasing heat with a change in the physical state of the material. This process is isothermal so there are no changes in temperature during the phase change.

$$Q = m \,.\, \Delta q$$

Where m is the mass of the PCM and Δq is the latent heat of fusion. PCM can be classified into organic, inorganic, and eutectic mixtures. Some examples of organic PCM are paraffins and non-metals, while examples of inorganic PCM are salt hydrates and metal alloys.

PCM materials usually have higher energy density than STES and their isothermal behavior allows for a more stable heat transfer and a more controlled charge and discharge process for accurate temperature regulation. One of the drawbacks of PCM is their low conductivity which makes them less suitable and there is a lack of commercially available PCM that could meet all the required characteristics.

Some of the required characteristics are:

- High phase change enthalpy
- Good thermal stability at low temperature
- Small volume change during the phase change process
- High thermal conductivity both in solid and liquid phase
- Low cost per unit of energy stored.

The PCM can be classified into Sub-zero PCM, Ice, Low-temperature PCM and high temperature PCM. The Sub-zero PCM are eutectic mixture which can be used for cold storage as the phase change takes place below 0˚C. The solid phase of water is ice which can be used for storing heat as it has high latent heat of fusion and good latent heat capacity. Low temperature PCM include paraffin waxes and inorganic salt hydrates which operate in the range of 0 ˚C to 120 ˚C. High temperature PCM can operate over 500 ˚C and the most commonly used materials are nitrate salts. High temperature PCM have high storage density but however low thermal conductivity and corrosive in nature.

1.3 Thermochemical Energy Storage:

In a thermochemical energy storage (TCES) a reversible endothermic chemical reaction is used to store energy in the form of chemical compounds and the heat is then released by recombining the two products (B and C) where exothermic reaction takes place. The following reaction is expressed as

$$A + \Delta Q \leftrightarrow B + C$$

The heat stored is than equal to the enthalpy of reaction ΔQ which is far greater than the enthalpy of fusion and higher than the heat capacity stored in sensible heat thus allowing to reach energy density up to 100-400 kWh/m^3. Good thermochemical storage allows for fully reversible reactions keeping high cycling storage efficiencies.

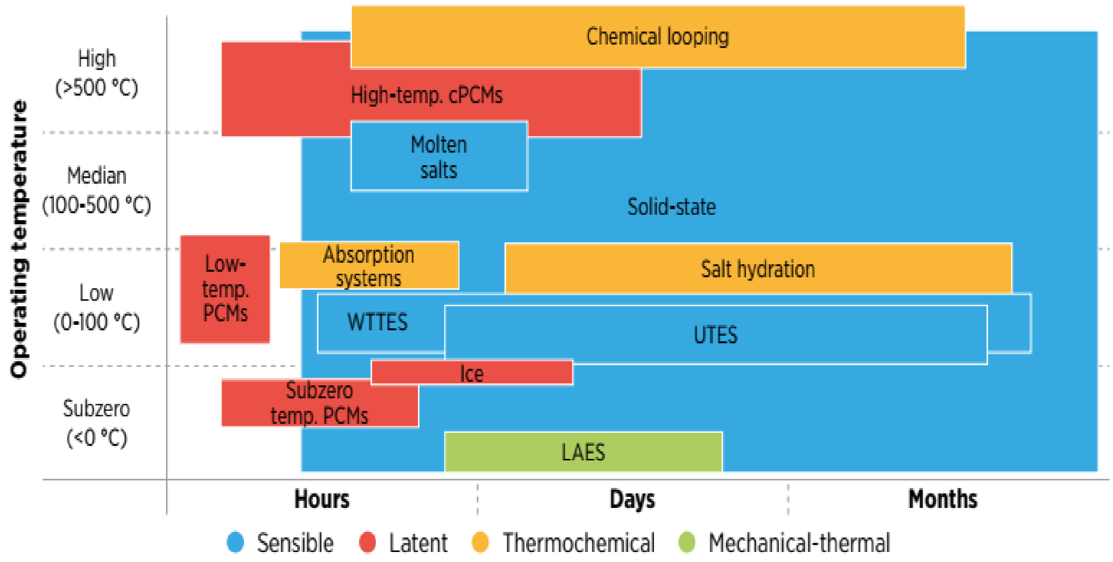
Thermochemical storage can be classified into reversible chemical reactions and absorption systems. In absorption based thermal storage heat is stored in the material itself by breaking the binding force between the sorbent and sorbate. Absorption systems can only work up to 350 ˚C while reversible chemical systems can operate at very high temperatures.

Chemical looping is a reversible chemical reaction between calcium oxide (Ca O) and carbon dioxide (CO_2) and the product formed is calcium carbonite ($CaCO_3$). Salt hydration is a sorption-based storage that absorbs and releases energy by alternate hydration and dehydration processes. When heat is supplied the salt dehydrates and release water molecules which are stored separately and when energy is required the stored water is supplied to the salt and heat is rejected. The absorption-based storage using concentrated refrigerant solution in which heat is stored by released water vapour form the refrigerant and stored separately and when energy is needed the water molecules area allowed to absorb by the refrigerant to generate heat.

For chemical reaction based thermochemical energy storage is based on carbonates, hydrates, and metal hydride. In which chemical reaction takes place and heat is stored in a chemical reaction and during the discharging process this heat is generated by an exothermic reaction which can be utilized by heat engines to generate electricity.

Some advantages of thermochemical energy storage are discharging temperature and very high energy density. Also, it has the ability to store heat with high efficiency for as long as the chemical reaction is not reversed which makes this technology a good candidate for seasonal storage. However, this technology is mainly at an R&D stage which translates into a high cost and creates uncertainty with regards to reliability and toxicity.

As it is possible to store energy for a long time in thermochemical energy storage systems, the stored energy can be used during off-peak hours for district heating purposes when there is demand.



WTTES- Water tank thermal energy storage; UTES- underground thermal energy storage CPCM- composite phase change material.

Figure 3-2. Operating temperatures and storage duration of TES technologies.

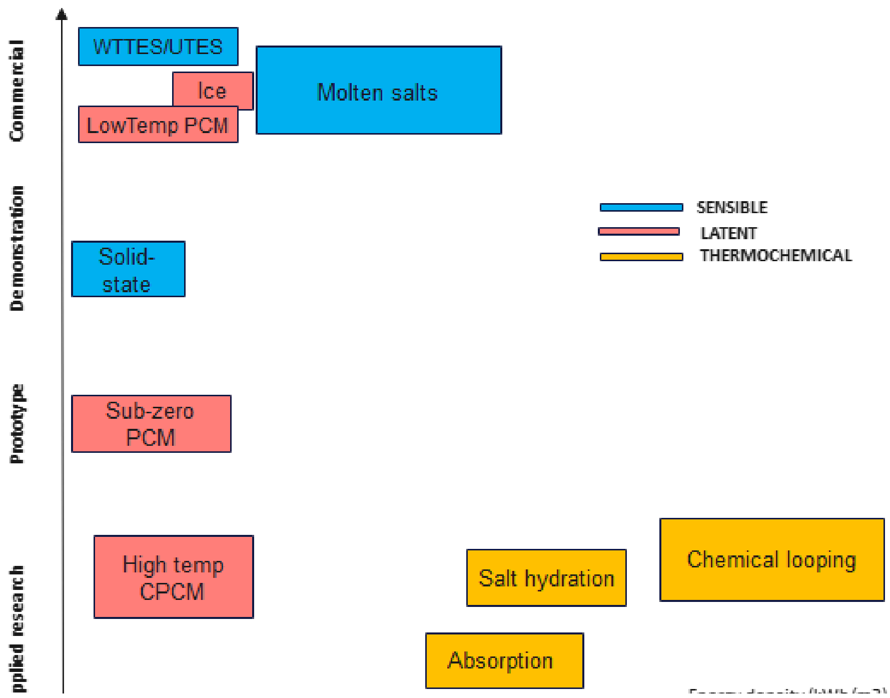


Figure 3-3. Classification of TES technologies based on technical maturity and energy density.

Table 3-1. Technical characteristics of different TES technologies[8].

TES	Type of storage	Power (W)	Capacity (Wh)	Energy density (kWh/m3)	Temperature range (˚C)	Storage duration	RTE (%)
Sensible	WTTES	kW - 10 MW	kWh - 1GWh	15-80	10-90	Hours-months	50-90
	UTES	MW - 100 MW	MWh - GWh	25-85	5-95	Weeks to months	>90
	Solid state	kW - 100 MW	10 kWh - GWh	25	-160 to 1300	Hours-months	>90
	Molten salts	100kW - 300 MW	MWh - 5 GWh	70-200	265-565	Hours-days	>98
Latent	Sub-zero PCM	kW - 10 kW	kWh - 100 kWh	30-85	Down to -114	Hours	>90
	Ice	kW - 10 MW	kWh - 100 MWh	92	-3 to 3	Hours-days	>95
	Low temperature PCM	kW - 10 kW	kWh - 100 kWh	56-60	Up to 120	Hours	>90
	High temperature CPCM	kW - 100 MW	10 kWh - GWh	30-85	Up to 1000	Hours-days	>90
Thermo-chemical	Chemical looping	10 kW - 1 MW	MWh - 100 MWh	800-1200	500-900	Months	45-63
	Salt hydration	-	10 kWh - 100 kWh	200-350	30-200	Months	50-60
	Absorption	10 kW - 1 MW	10 kWh - 100 kWh	180-310	5-165	Hours to days	COP - 0.7-1.7

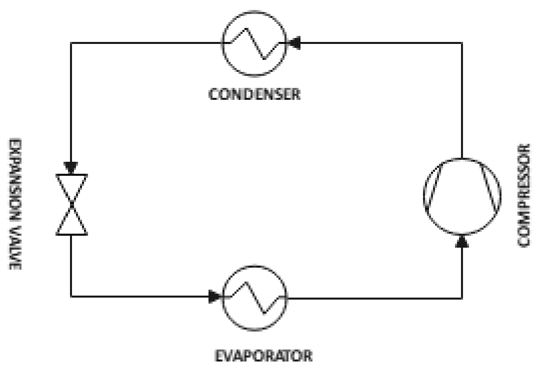
4. PTES BASED ON ORGANIC RANKINE CYCLE CONFIGURATIONS

4.1 PTES Configuration Based on Sub-Systems Model

Based on the sub-systems, PTES can be classified with different possible configurations. The regenerated or recuperated configuration for both charging and discharging cycles and a possible reversible PTES machine is of major interest among the different PTES configurations. So, the base and regenerative configuration for both the sub systems are discussed in this chapter. e

4.1.1 Vapour Compression Heat Pump (Base & Regenerative):

For high temperature Vapour compression heat pump (VCHP) different configurations can be adapted for utilizing waste heat which includes single stage cycle, single stage with internal heat exchanger, two stage cycle, two stage cycle with internal heat exchanger and a two stage with intermediate internal heat exchanger[43].



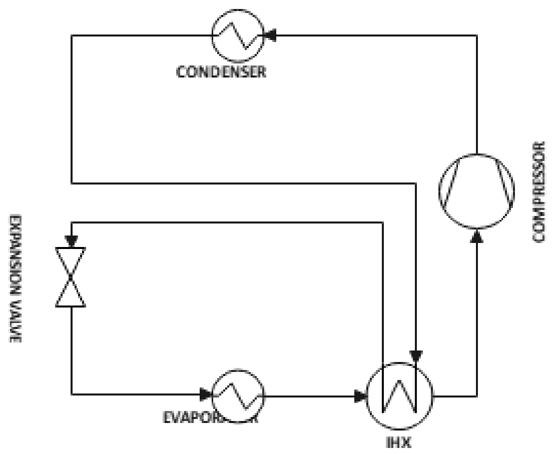


Figure 4-1. Basic and regenerated configuration of VCHP.

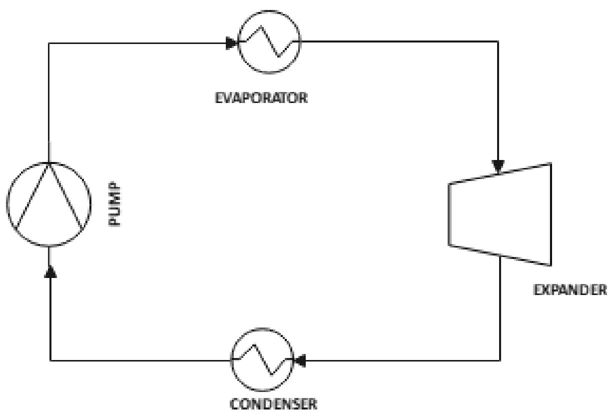
The basic single stage VCHP is composed of a compressor, condenser, expansion valve, and evaporator. By adding an internal heat exchanger between the suction and the liquid line to this basic configuration the recuperated or regenerated single stage configuration is achieved. The regenerator reduces the temperature of the working fluid by heating the working fluid on the suction side. Therefore, the internal heat exchanger is installed between the condenser and throttling valve / evaporator and compressor. After releasing heat in the condenser, the refrigerant enters the regenerator and releases the heat again before entering the throttling valve. In contrast to the refrigeration systems, the internal heat exchanger is needed more at the suction end to increase the discharge temperature. Higher discharge temperature increases the performance of the heat pump applications.

For temperature lift of around 40-60K, single stage with internal heat exchanger has higher performance than the other configurations. So, considering the performance of the heat pump the single stage with regeneration can be selected for the temperature lift of 40 – 60K and the two-stage cycle with internal regeneration may be suitable for temperature lift of 80-100K. The waste heat temperatures and the working fluid selection are very sensitive to this type of configuration. Therefore, temperature lift is the key parameter to select the proper configuration. High COP requires a high temperature range for the waste heat source and low storage temperatures for thermal storage systems.

For high temperature heat pump configurations, instead of the basic configuration the system with internal heat exchange shows better performance and COP values. So, depending on the waste heat temperature and the working fluid a single stage or two stage cycle with internal heat exchanger can be adopted. The only drawback of this type of configuration is the increase in cost of the equipments to have a better overall performance.

4.1.2 Organic Rankine Cycle (Base & Regenerative):

Different configurations for organic rankine cycle are considered for utilizing waste heat like basic ORC cycle, ORC with internal heat exchanger (single stage and two stage cycles), reheat ORC. In this chapter the basic ORC and ORC with internal heat exchangers are discussed for PTES system configuration.



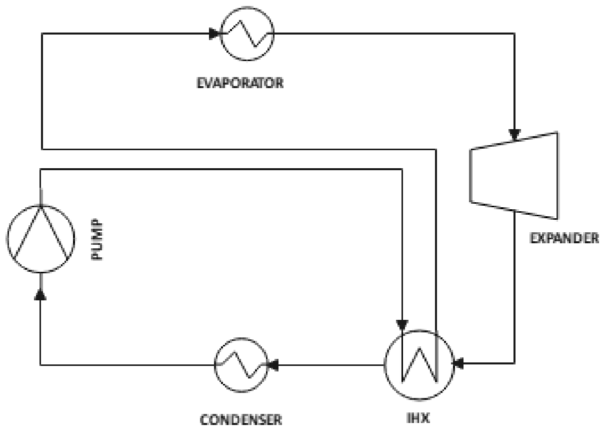


Figure 4-2. Basic and regenerated configuration of Organic rankine cycle.

The basic ORC comprises of evaporator, expander, pump, and condenser. For the ORC with internal heat exchanger configuration, a internal heat exchanger is added between the turbine and the condenser / pump and evaporator. After expanding in the turbine, the working fluid enters the internal heat exchanger and releases the heat before entering the condenser. With the help of internal heat exchanger, the working fluid is heated again before entering the evaporator. In this configuration, the wasted thermal energy rejected by the condenser would be extracted and supplied again to the working fluid.

By using an internal heat exchanger, it improves the overall thermodynamic performance of the system. There is no high change on the net power output by using an internal heat exchanger but with respect to thermal efficiency the ORC with internal heat exchanger has better performance[44][45]. However, the overall performance of the organic rankine cycle differs depending on the selection of working fluid which is the major factor to be considered.

The system with internal heat exchanger not only increases the thermodynamic performance of the system but also decreases the amount of heat required and system irreversibility to produce the same power output compared to basic ORC. The expansion of the ORC ends in the superheated state at a temperature higher than the condensation temperature[38]. So, in that case the ORC can be internally regenerated but unlike other ORC applications the internal regeneration is beneficial in this case as the thermal energy is stored before the ORC cycle and hence, we have a more compact thermal energy storage if we can well efficient ORC performance. Introduction of internal regenerators in the ORC sub system reduces the cost of turbine and compressor than the systems without internal heat exchanger.

Comparison between with and without the use of regenerators for the PTES system

In a study carried out by G. Frate [17], different system configuration was discussed and a comparison was made for the high temperature and low temperature configurations. In that study the advantage of utilizing a configuration with internal heat exchanger is discussed and the results showed that the internal regeneration helped to increase the exergy efficiency by 15% and energy density by 6%. Therefore, introducing an internal heat exchanger is a critical way to improve the thermodynamic performance of the system. Also in another study made by G.Frate[25], analyzed the PTES system with thermal integration and in that the introduction of regenerators improved the exergy efficiency by 15% with a roundtrip efficiency of 55% and energy storage density of 15 kWh/m^3 .

Introducing internal regeneration in the VCHP system show different effect when compared to the introduction in ORC system. This is studied by Ruoxuan Fan [23], which compared four different combination of sub-systems with and without regenerators. So, as a result the system with internal regeneration on both the charging and discharging cycles showed better economic performances than the systems with basic configurations. With respect to thermodynamic performance, the internal regeneration on VCHP sub system improved the roundtrip efficiency is obvious than the internal regeneration on the ORC sub system. In a study carried out by Liu [32], also the introduction of regenerators improved the LCS while the basic configuration is 11.5% higher. Therefore, using regenerators improved both the economic and thermodynamic performance. Without the use of regenerators, the cost of the compressor and turbine may increase by 2.3% & 1.5%, and it is concluded that the exergy destruction of heat pump is higher than the ORC sub systems so, without regenerator in both the sub systems it accounted for 19% overall increase.

4.1.3 Reversible Pumped Thermal Energy Storage:

In PTES systems, the equipments costs of both charging and discharging cycles can be significantly reduced by using a reversible PTES system. A clear separation between the heat pump and ORC guarantees high performance but for small scale applications a reversible cycle can be used by compromising the performance in terms of cost.

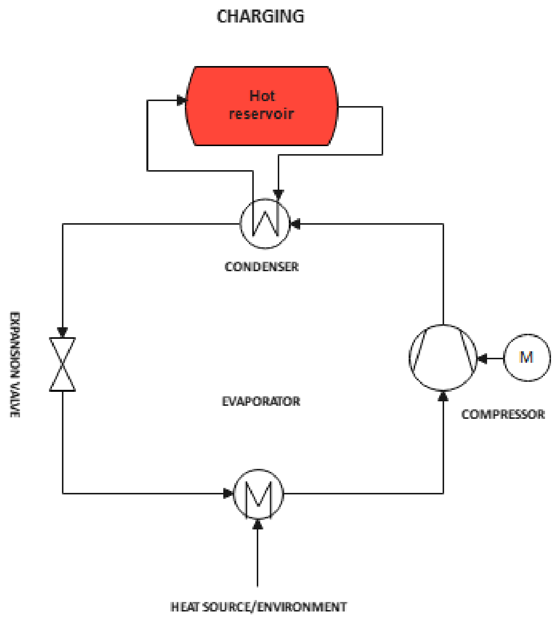


Figure 4-3. Charging cycle configuration of Reversible PTES using vapor compression heat pump (VCHP).

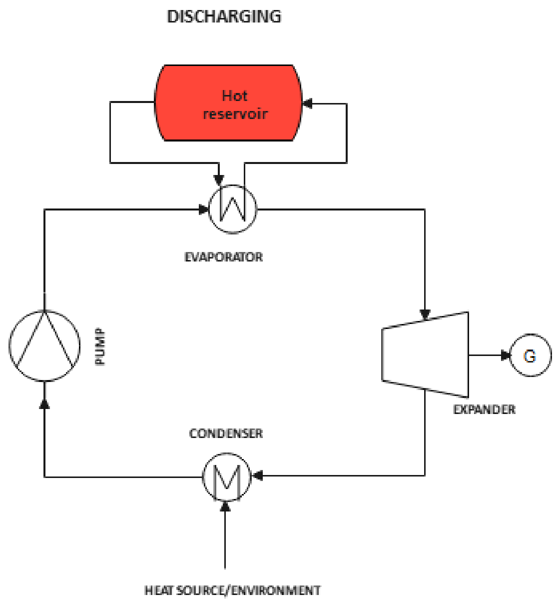


Figure 4-4. Discharging cycle configuration of Reversible PTES using Organic Rankine Cycle.

In that case, two types of reversible configurations can be adopted which includes a partly reversible configuration and fully reversible configuration[17].

In partly reversible configuration, both the VCHP and ORC may use identical heat exchangers but, in that case, both the charging and discharging cycles must operate with the same working fluid. Also, by adopting this configuration similar mass flow rates can be achieved by setting a nominal charge and discharge durations. It is only partly reversible because integrating the heat exchangers is just the minimal task of reversing the cycle while there are at least three machines pump, expander, and compressor. For smaller systems, the cost of the heat exchangers is significant and may be 40% of the total cost and when compared to sum of compressor and expander coat is slightly more than 20%. For larger systems the difference is even greater, so the partly reversible configuration is very promising and has a significant cost reduction with little sacrifice in the performance of the system. So, for medium to large scale systems the partly reversible PTES systems are more suitable.

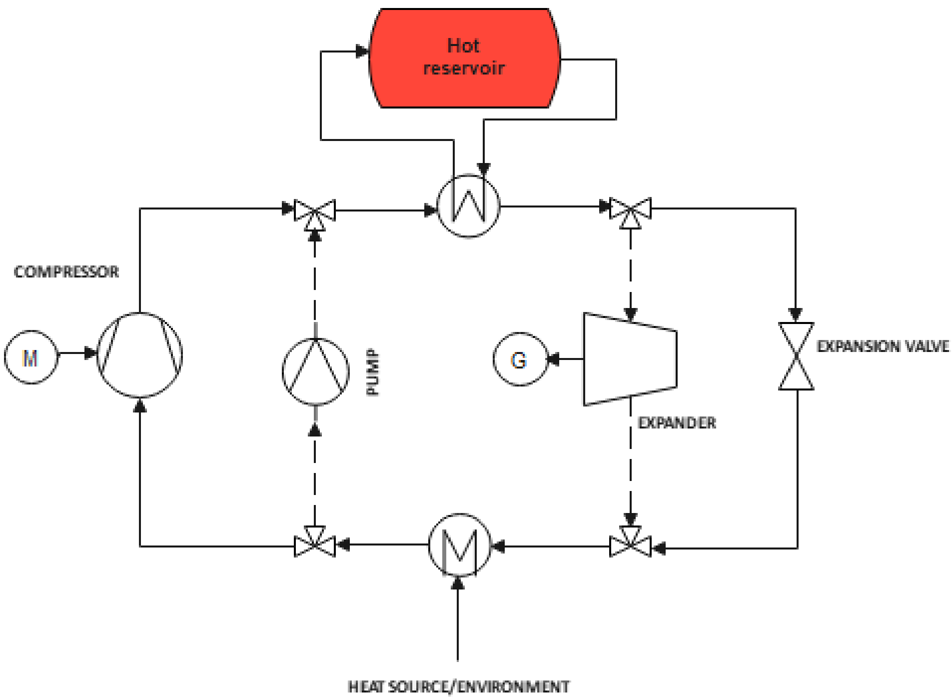


Figure 4-5. Partly reversible PTES configuration using VCHP and ORC.

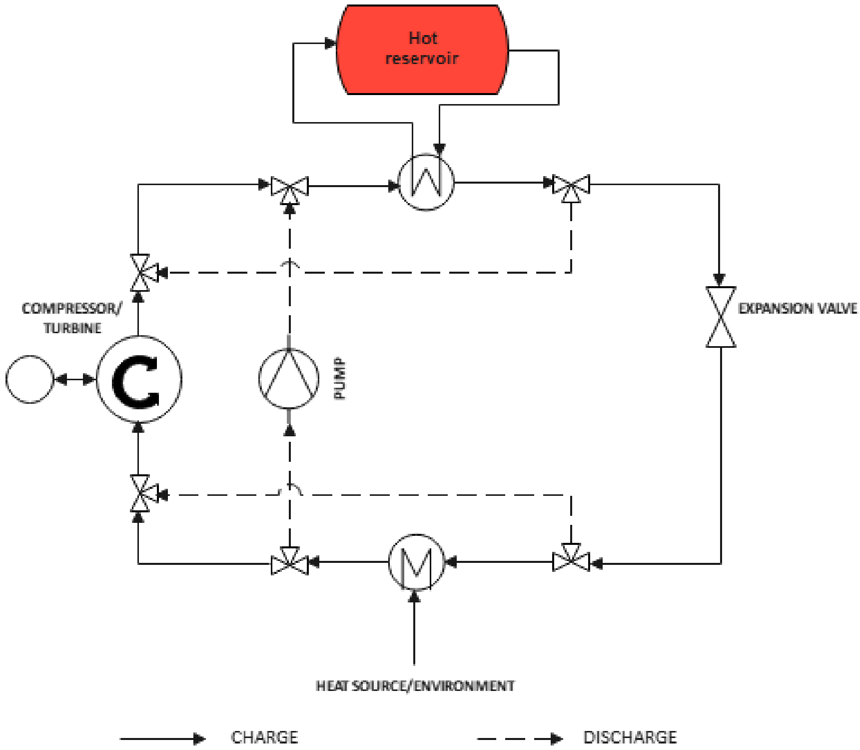


Figure 4-6. Fully Reversible PTES configuration using VCHP and ORC

For very small-scale systems, it is possible to adopt a fully reversible PTES system with the use of volumetric machines for both the VCHP compressor and ORC expander. So, in this case the machine is alternatively used for charging and discharging process. A fully reversible PTES configuration and ways of integrating it thermal source like waste heat are studied by different authors in the literature [18], [29], [46],[28],[47]. The most sensitive component in the fully reversible PTES configuration is the reversible compressor/expander. In a study [48], the experimental model for reversible PTES was made using a scroll compressor with minor modifications with a net power output of 3.5 kW. Therefore, the cost of the machine can be considered like the compressor one.

In general, the heat exchangers can comprise both the condenser and the evaporator in the heat pump as well as in the ORC and a volumetric machine can be used both as compressor and turbine for the charging and discharging cycles. Only the throttling valve of the heat pump must be replaced with a pump. This configuration can also be integrated with the waste heat or other thermal sources to improve the performance of the overall system.

Small scale systems (1-1000 kW) are often characterized by simple layout to reduce to cost which allows to design the systems only in the range of 10kW to 1MW. The major disadvantage of using reversible machines is the poor performance with roundtrip efficiencies often reported less than 50%. The performance can be improved by utilizing additional heat sources in the charging phase.

4.2 PTES Configuration Based on Thermal Storage

In this chapter all the PTES configuration based on different TES rely on Vapour compression heat pump (VCHP) for charging phase and Organic rankine cycle (ORC) for discharging phase. As we have seen in the previous chapter both the charging and discharging sub-systems can be modelled with a base configuration and a regenerated configuration. The aim of this chapter is to present and discuss the integration of different TES configurations with ORC based PTES.

Rankine based PTES works with a narrow temperature range and lower hot reservoir operating temperatures. So, for temperature range below 200˚C sensible heat storage with multiple pressurized tanks can be used but with a increase in cost because of the need of thicker walls at high pressures. In order to avoid the increasing costs in water tanks an alternate solution is using latent heat storage with ˚C PCM materials or molten salts with temperatures between 200 and 565 ˚C. But however, it is a tradeoff between the PTES performance and the economic scalability of the plant. So, with a good thermoeconomic analysis a better system can be developed.

4.2.1 Sensible Heat PTES Configuration

Sensible heat PTES configuration has low energy density values, but it is the simplest TES from design point of view because compared to latent heat storage it does not have any fixed temperature range. It is also capable of providing constant charge and discharge temperature profiles.

The sensible heat storage can be designed with single reservoir and double reservoir configuration. The double reservoir configuration is capable of providing constant charge and discharge rate but the investment cost will increase because of the extra reservoir to increase the storage volume[49]. With constant charge and discharge profiles it is possible to achieve better performance. The selection of single or double reservoir is the adjustment between the investment cost and the performance which can be done by thermo-economic optimization of the system.

The heat is stored in the storage tank with hot fluid on the top and the cold fluid at the bottom as a result of different densities of the fluid with respect to their temperature difference, so this forms a boundary between them which is called a thermocline. With the thermocline formation it is possible to store both hot and cold fluid in the charging and discharging process. In order to achieve a stable operation of the thermocline it is necessary to minimum low flow rates to maintain the balance.

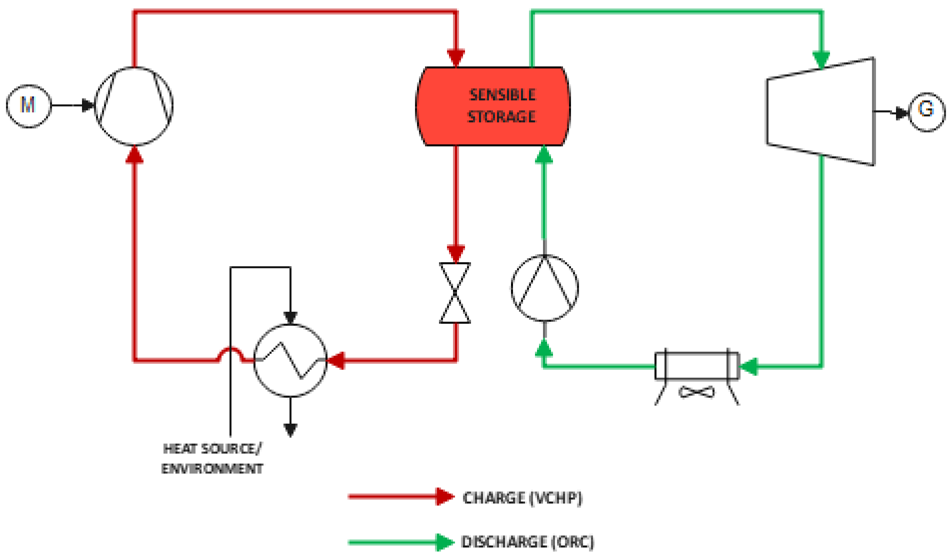


Figure 4-7. PTES layout integrated with Sensible thermal energy storage with basic configurations of VCHP and ORC.

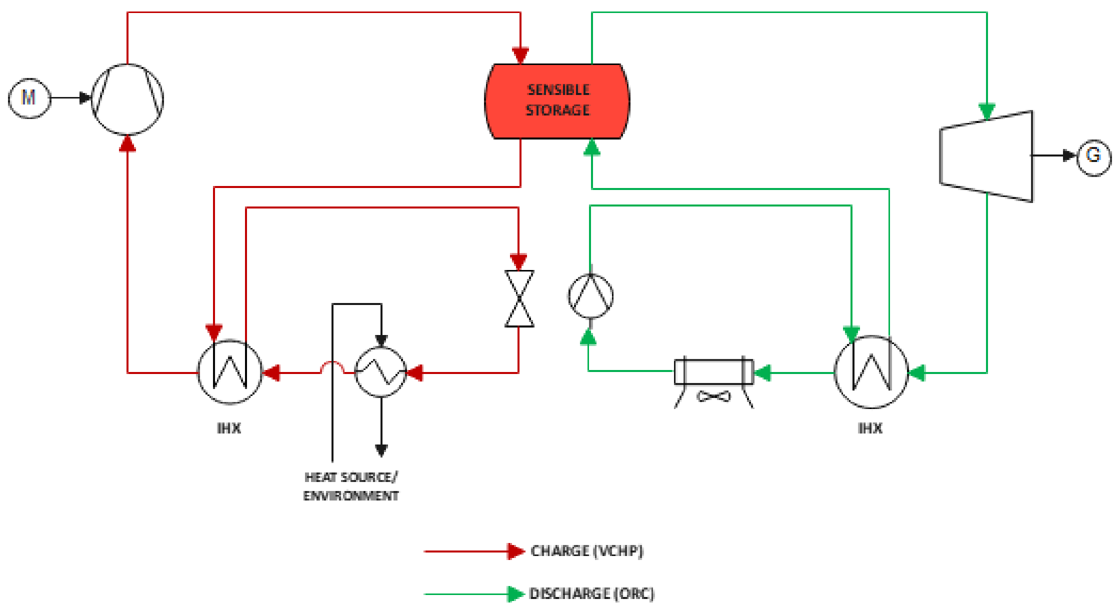


Figure 4-8. PTES layout integrated with Sensible thermal energy storage with regenerated configurations of VCHP and ORC.

In the charging phase the compressor increases the temperature of the fluid and through the condenser the thermal energy is stored in sensible heat storage. After the heat transfer the pressure and temperature of the fluid is decreased by expansion valve and evaporator superheats the working fluid before compression. In the discharging process the thermal energy from the hot storage is utilized to evaporate the working fluid and expand in the turbine. In both the sub-systems regeneration can be adopted, which lead to a better performance of the systems but the only disadvantage is the increase in cost.

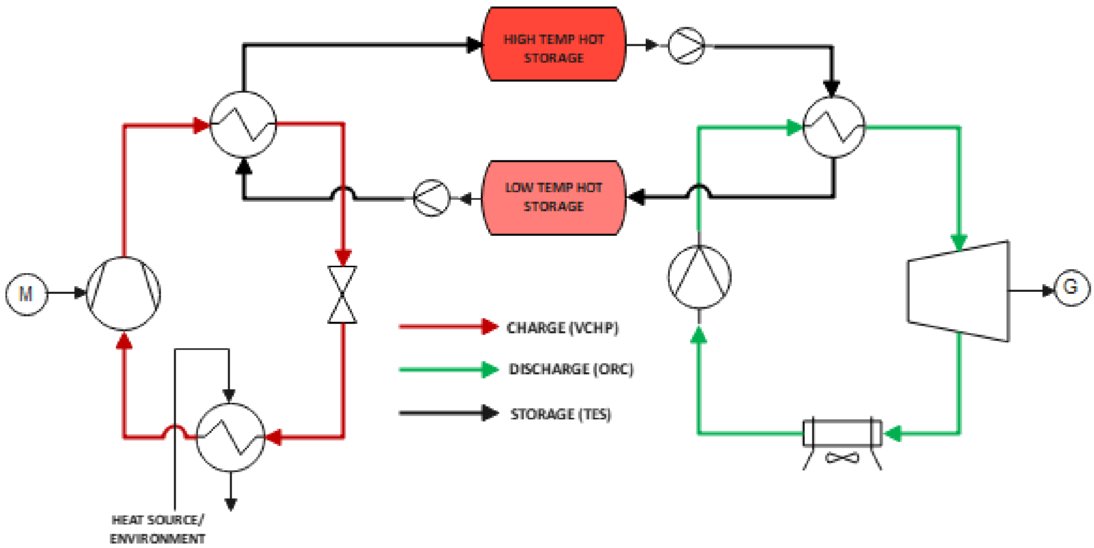


Figure 4-9 PTES layout integrated with Double reservoir sensible thermal energy storage using basic configurations of VCHP and ORC

4.2.2 Latent Heat PTES Configuration

Latent heat PTES configuration can provide larger energy storage densities and smaller volumes compared to sensible heat storage. The charging phase consists of a compressor which increases the pressure and temperature of the working fluid and then the thermal energy to the latent heat storage is provided by the condenser. The lamination valve is used to decrease the pressure of the fluid and the saturation temperature and then evaporator superheats the working fluid to evaporate from the lamination valve to provide high temperature for the compression. In the discharging phase, the working fluid from the condenser is pumped and using the stored latent heat the working fluid is evaporated and expanded in the turbine.

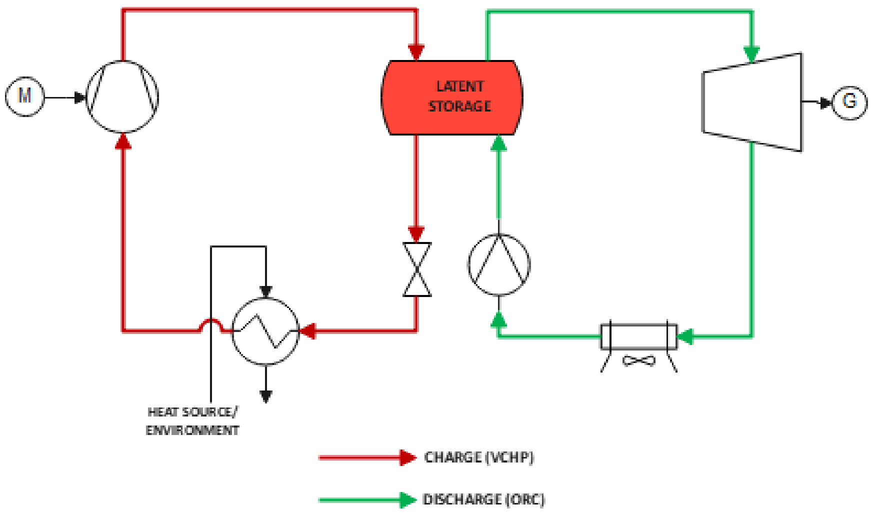


Figure 4-10 PTES layout integrated with latent thermal energy storage with basic configurations of VCHP and ORC.

In both the charging and discharging phase regeneration can be adopted to increase the COP and ORC efficiency. In charging phase, the heat from the condensed fluid can be used to superheat the steam after the evaporation process. In the discharging process if the fluid at the outlet of the turbine is sufficiently high it can be used to preheat the working fluid after the pump and increase the temperature at the inlet of the evaporator. Adopting a regenerated configuration in the charging phase is more obvious, which increases the COP of the heat pump cycle and overall performance.

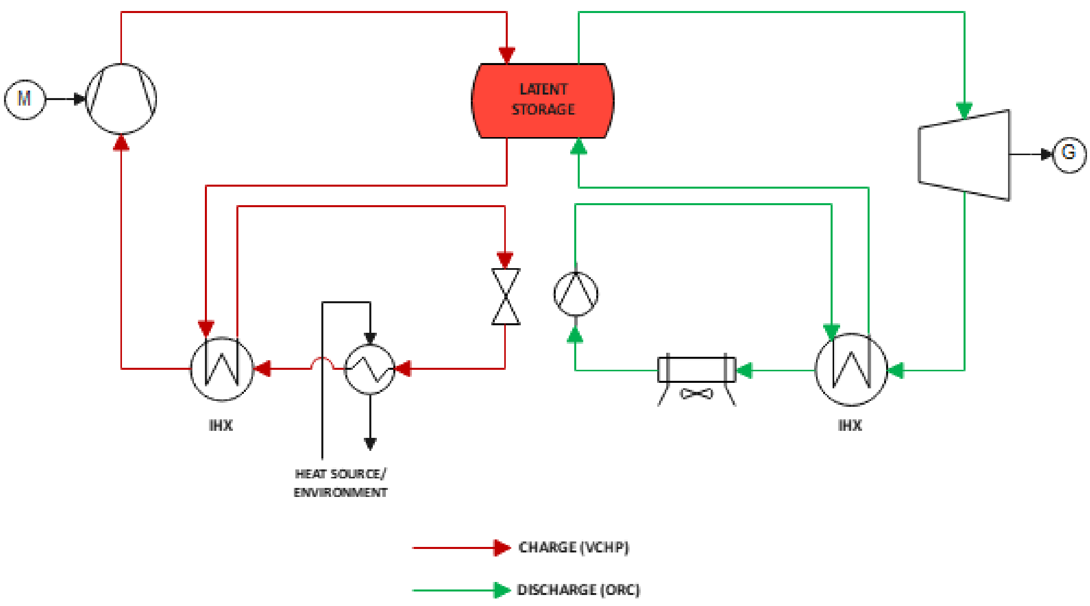


Figure 4-11 PTES layout integrated with latent thermal energy storage with regenerated configurations of VCHP and ORC.

There are no significant differences in the performances obtained by rankine based PTES with either STES or LTES configurations. In a study by Eppinger the overall roundtrip efficiency ranged between 30-90% and similar trends in sensitivity analysis we reported for both STES and LTES configuration like the heat source temperatures, isentropic efficiencies, and the pinch point differences. However, the LTES roundtrip efficiency is more sensitive to the choice of the storage temperature than the STES as the phase change temperature is very important for LTES configurations. In LTES configuration a decrease of roundtrip efficiency is observed from 65% at a storage temperature of 110 [˚C] down to 35% at storage temperature of 160 [˚C] when using R1233zd[E] whereas in the case of STES configurations roundtrip efficiency varies less than 10% around 52.2%.

4.2.3 Sensible and Latent Heat PTES Configuration

In this configuration both sensible heat and latent heat storage are used. In the charging phase, after the condensation process the thermal energy from the sub-cooling of the working fluid is stored in the sensible heat storage and during the discharging phase the thermal energy from the sensible heat storage is used to preheat the working fluid till the evaporation temperature.

The major advantage of integrating a sensible heat storage with the latent heat storage system is to reduce the irreversibilities caused during the heat transfer process by reducing the temperature difference between the source and the working fluid. But this configuration requires a high heat exchange area, so it leads to higher costs.

For the charging and discharging sub-systems both the basic and regenerative configuration can be adopted. By adopting the regenerative configuration, it is possible to use the saturated liquid from the condenser to store heat first in the sensible heat storage before moving into the regenerator. Because of this the mass flow rate of the working fluid is increased and obviously the COP of the heat pump also increases. Also, on the other side the temperature of the working fluid is increased using the sensible heat before entering the heat exchanger that comes in contact with the latent heat storage. So, by achieving this the thermal energy from the latent heat is completely utilized to evaporate the working fluid therefore a higher mass flow rate can be achieved.

As the power cycle efficiency and the COP of the heat pump are expected to grow, the overall roundtrip efficiency will also increase. The working fluid of the heat pump be subcooled until the temperature is equal to the inlet of the economizer on the power cycle because the aim is to increase the temperature at the outlet of the economizer as much as possible. But this is not possible because it must happen with a certain temperature difference. Therefore, after storing the heat in sensible heat storage the working fluid must be heated up using low temperature waste heat sources which will lead to high roundtrip efficiency.

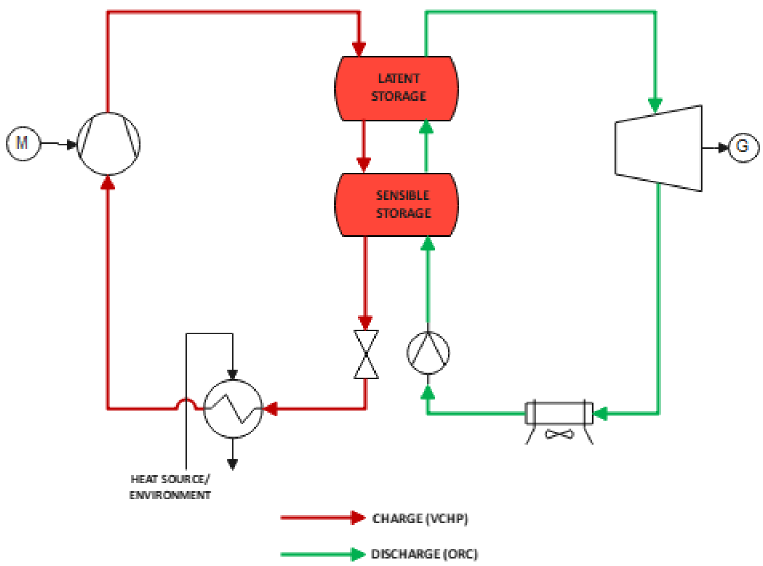


Figure 4-12 PTES layout integrated with sensible & latent thermal energy storage with basic configurations of VCHP and ORC.

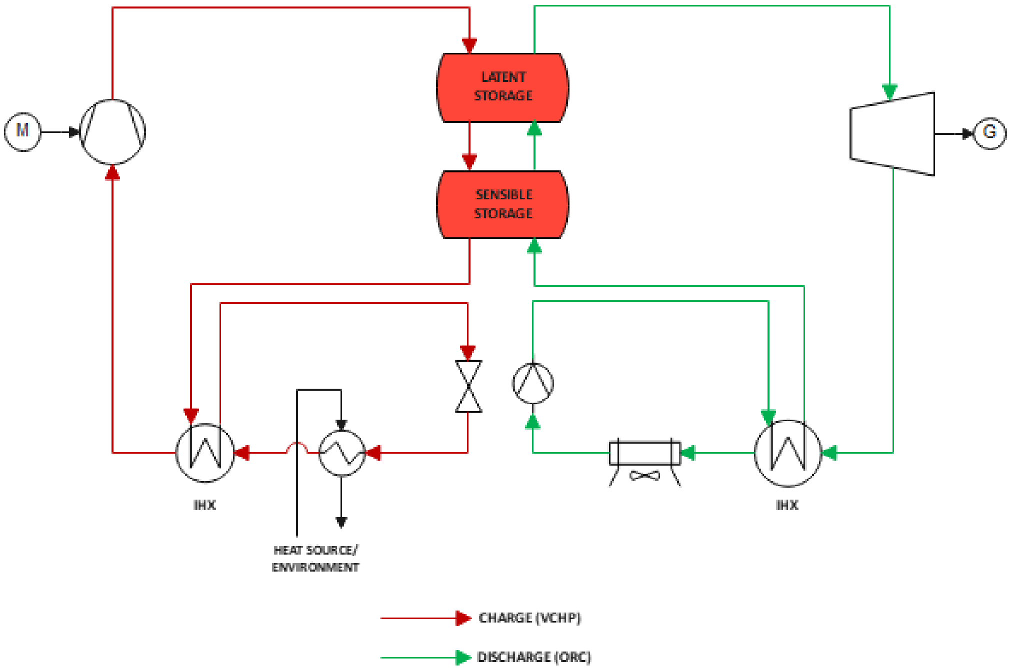


Figure 4-13 PTES layout integrated with sensible & latent thermal energy storage with regenerated configurations of VCHP and ORC.

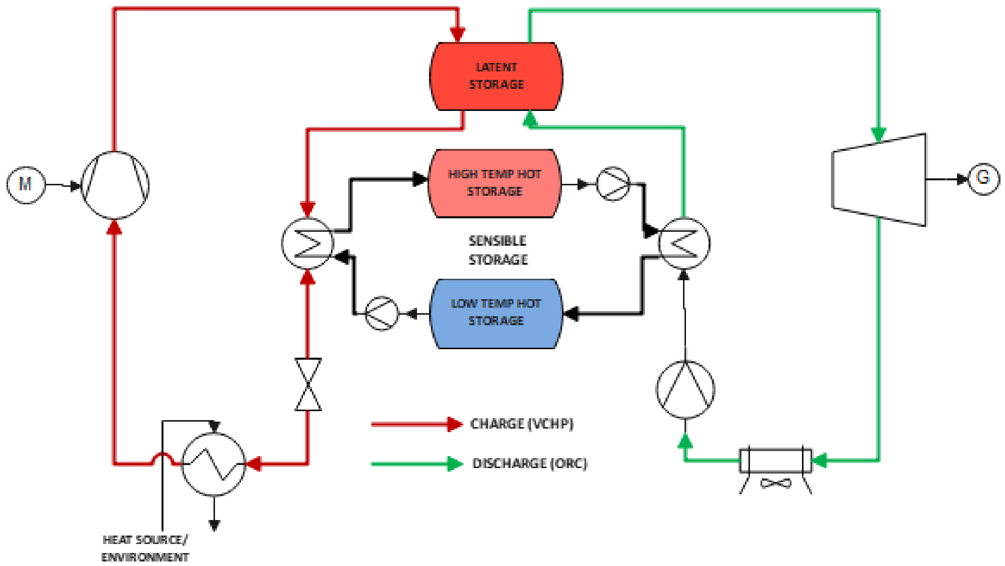


Figure 4-14 PTES layout integrated with double reservoir sensible & latent thermal energy storage with basic configurations of VCHP and ORC.

The combination of STES and LTES is considered in the studies conducted by CHESTER H2020 European project framework [50]. The LTES with a eutectic mixture of potassium nitrate and lithium nitrate 133 [˚C] is used to exchange thermal energy while a pressurized water storage is used as the STES to match the heat pump subcooling and the ORC preheating. With this storage configuration roundtrip efficiencies ranging from 40-120 % are obtained with the heat source temperature ranging from 40-100 [˚C] and sink temperature 15 [˚C]. A pinch point difference of 5 [˚C] and an additional heat exchanger is required at low temperatures as the heat ratio between the heat exchange with the STES and LTES must remain the same for the charge and the discharge.

4.2.4 Thermochemical PTES Configuration

Thermochemical storages have 5-10 times higher energy density than sensible and latent heat storage systems, but this technology is at a very early stage of development. With this type of configuration, it is possible to achieve more compact systems and very little irreversibilities. The performance of thermochemical energy storage is defined as the ratio between thermal energy input and thermal energy output during a complete charge and discharge. Since the energy is stored as chemical energy it is possible to store the reactants at ambient temperature, there is less energy loss with the charging and discharging cycles. Compared to sensible and latent heat storage they have very energy densities so they cannot be combined with other storages and thermochemical storage efficiency is close to unity.

In the charging phase, the compressor is used to increase the temperature of the working fluid and then condenser transfers thermal energy to the thermochemical reactor as a result an endothermic reaction takes place inside the reactor. By integrating the low temperature waste heat sources, it is possible to achieve a very high temperature.

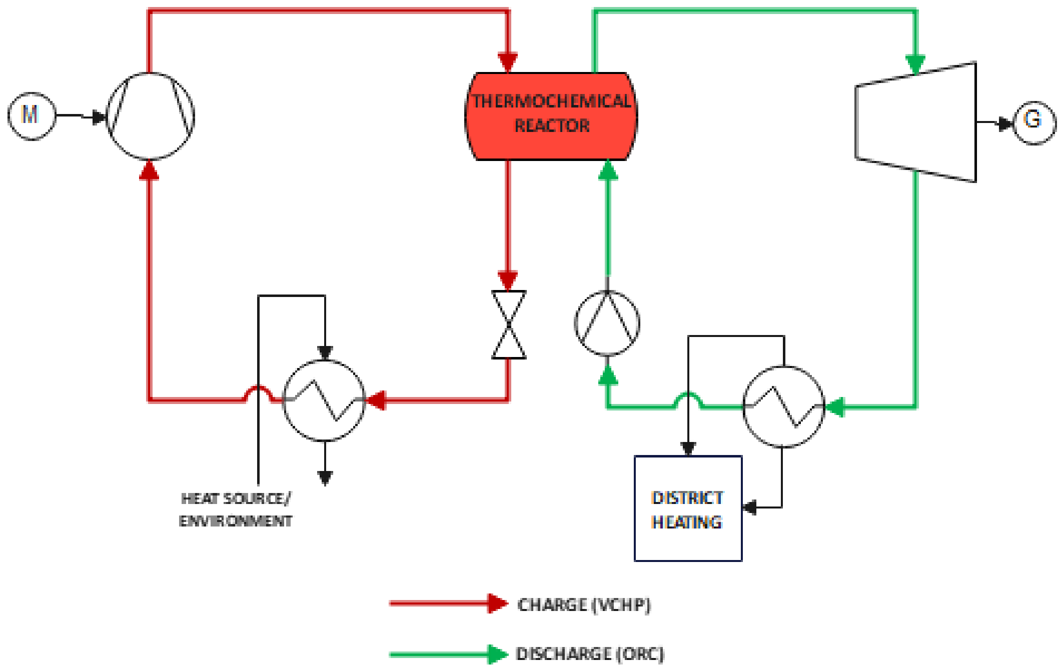


Figure 4-15 PTES layout integrated with Thermochemical energy storage with basic configurations of VCHP and ORC.

In the discharge phase, the reaction is reversed with an exothermic reaction as a result heat is generated and this generated thermal energy is used in the power cycle to evaporate the working fluid and expand it in the turbine. Also, the heat released by the power cycle can be used for district heating purposes. As it is possible to store energy for a long time in thermochemical energy storage systems, the stored energy can be used during off-peak hours for district heating.

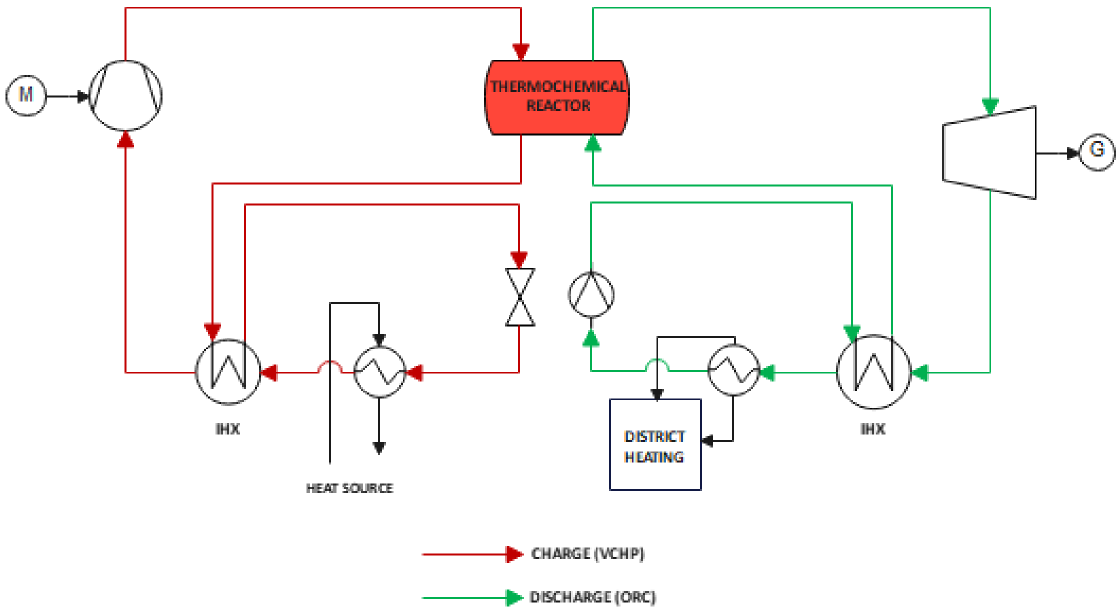


Figure 4-16 PTES layout integrated with thermochemical energy storage with regenerated configurations of VCHP and ORC.

4.3 Optimal Cycle Configuration

To achieve high round trip efficiencies for PTES based on ORC cycles certain technical constraints should be considered. The choice of machinery should be well established technology in order to reduce the technological risks. The PTES systems with simple configuration with limited components are necessary for reducing both the exergy generation and the cost of the plant. But however, adopting a regenerative configuration improves the overall efficiency of the system. The integration of regenerators is a tradeoff between the cost and performance of the system. By using the regenerative configuration, the entropy generation of the heat exchangers are reduced which increases the roundtrip efficiency.

Considering the technological limitation of the heat pump and the heat engine, the maximum temperature reached by these systems can be set in the range of 100˚C to 150 ˚C. But however, the maximum temperature is also determined by the type of working fluid and the type of storage medium used. At high very high temperature and pressure the heat pump technology is not commercially available and at very low temperatures the ORC operates at very low efficiency.

The environment or ambient acts as the heat sink 15 ˚C for the ORC cycle and low temperature heat source or waste heat can be used in the heat pump for charging process which is generally called as Thermally integrated PTES systems. The low temperature heat source or waste heat is in the range of 80˚C to 100 ˚C.

The configuration with regenerative layout reduces the exergy of the working fluid before entering the throttling valve as the working fluid after storing heat in hot storage can be used for superheating the working fluid before entering compressor. And similarly in the case of ORC discharging cycle working fluid after turbine can be used to preheat the working fluid before entering the reservoir. Regenerated configuration reduces the exergy losses but increases the complexity of the system and additional equipment costs. Also, the regenerators cause variation in the compressor inlet temperature which may result in higher exergy losses as the compressor is operated far from its design point.

The configuration with a simple layout involves lower equipment costs and but depending on the working fluid and operating temperature the systems may require an additional component in order to superheat the working fluid before entering the compressor. In general, with a simple configuration the compressor works close to its design point with constant inlet temperatures. The absence of regenerators increases the exergy destruction of the heat exchangers.

The PTES systems based subcritical ORC use pure working fluids and therefore the organic fluid experiences isothermal, liquid-vapor phase change during the heat transfer. As a consequence, the ORC based PTES systems can be combined with latent heat storage because the latent heat storage is also subject to isothermal, liquid-solid phase change during heat transfer. Therefore, the exergy loss can be minimized with small temperature differences. With a good combination of working fluid and phase change materials for latent storage it is possible to achieve high roundtrip efficiencies.

5. POWER PLANT COMPONENTS & SPECIFICATIONS

5.1 Working Fluid

The choice of working fluid for pumped thermal energy storage systems depends on several factors including the operating temperature range, heat transfer properties and environmental impacts like Global warming potential (GWP) and Ozone depletion potential (ODP). Lower environmental impact can be achieved by choosing working fluids with ozone depletion potential larger than 1 and lower GWP should be considered. For an ideal working fluid selection, it should have high specific heat capacity, low viscosity, and good thermal stability to efficiently transfer thermal energy. Working fluid selection is a key aspect because it directly influences the size and design of the components and which in turn affects overall efficiency.

In general Rankine based PTES uses pure working fluids and mixtures. Pure fluids can be further divided into dry (retrograde), wet, and isentropic fluids based on the slope of the saturated vapour curve on the T-s diagram.

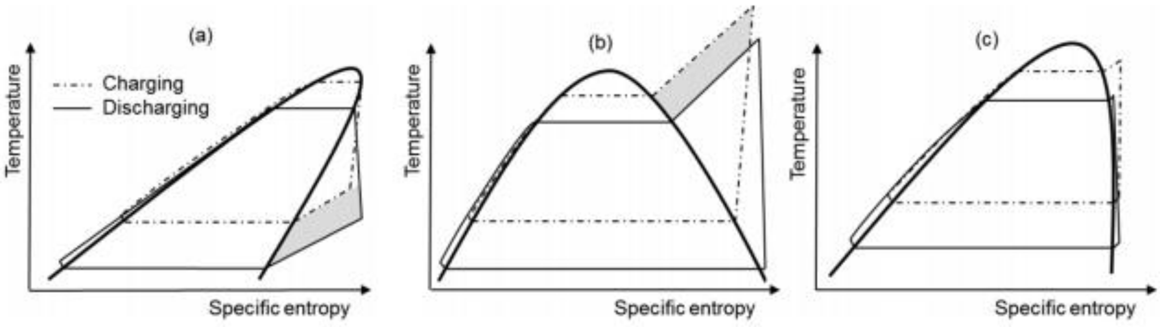


Figure 5-1 T-s diagram of different working fluids: a) Dry fluid b) Wet fluid c) Isentropic fluid.

Retrograde or dry fluids exhibit retrograde condensation or boiling, this occurs when the pressure of the fluid drops below a certain critical point causing the vapor to condense into liquid at a higher temperature than expected with the normal boiling point. Because of this phenomenon the saturation vapor curve exhibits a negative slope indicating the non-ideal behavior. From the Fig. the shaded area represents the mismatch between the charging and discharging cycles and this mismatch can be avoided by preheating the fluid before entering the compressor and cooling the fluid after the turbine. Therefore, it reduces the final compression temperature which increases the COP, and an extra condenser will be required to cool down the superheated fluid at the turbine outlet. This mismatch may require additional thermal storage which again increases entropy production. Similarly, in the case of wet fluids this mismatch requires extra storage to cool the vapor from compressor outlet and superheating the vapor before entering the turbine. Also, a reheater may be required in the case of wet fluids.

Dry fluids require very high degree of superheating to prevent wet compression because wet compression could decrease the evaporation temperature and affect the system performance and if this superheating is done inside the evaporator it reduces the COP and the global system performance. But however the superheating of dry fluids requires less exergy. Wet fluids do not require any superheating or may be very less value[51].

Isentropic fluids exhibit very little differences between the charging and the discharging cycles so no additional storages or heat exchangers are required which minimizes the entropy generation. So, the fluid selection depends on the different combination of source and sink temperatures. Also, by choosing high quality compressors and expanders with good isentropic efficiencies the mismatch in the temperature differences with the heat transfer can be optimized.

Several fluids can be used for rankine cycles including organic and inorganic refrigerants, water, ammonia, and ethanol. As we are investigating ORC based PTES our subject of interest here is organic fluids. Some of the commonly used organic working fluids are R134a, R245fa, pentane and cyclohexane. Due to operational temperatures, it is better to work with low pressures to reduce the mechanical and thermal stress. So, for the selection of working fluid lower operational pressures should be considered. R1233zd(E) stated to have better efficiency and lower operational pressure values which makes them suitable for PTES systems.

For subcritical operation of PTES systems and maximum temperature range of 100˚C to 150 ˚C, the critical temperature of the working fluid should be chosen greater than 120 ˚C because critical temperature lower than that impose a very low upper temperature limit for the PTES system. The lowest pressure should be higher than the ambient pressure to prevent air or water infiltration and the higher-pressure range should be lower than 30 bar as the components for hat engine are commercially available in that range and average equipment costs. Also, high density of liquid or vapour phase results in lower volume floe rates which results in smaller components and reduced equipment costs.

For a simple system design, the heat pump and the organic rankine cycle can be designed with the same working fluid. In a study conducted by Dumont and Lemort [29], 16 different organic fluids were examined with ORC based PTES to determine the optimal roundtrip efficiency. The study investigated different temperature lifts in the VCHP and different heat source and sink temperatures and R1233zd(E), R1234Y, R236EA, R11 and R245FA reported the best performances. But considering the environmental hazards such as GWP and ODP and other factors such as flammability or toxicity[52], only R1233zd(E) and R1234Y are the safe fluids from the previous study. In an extensive study carried out by Frate [53] which investigated different working fluids for VCHP for temperature ranges from 50-150 [˚C], R1233zd(E) show better performance with the compromise between the COP and volumetric heating capacity.

The impact of fluid selection with respect to heat source temperature is also studied by Frate [38] for 16 fluids and it is noted that the performance of the system globally increased with the increase in the heat source temperature Tsource but the maximum performance was achieved for the heat source temperature Tsource close to the critical temperature. The best results were obtained with R1233zd(E) achieving roundtrip efficiency of 130% for heat source temperature of 110 [˚C]. In the thermodynamic analysis carried of high temperature PTES systems[22], highest roundtrip efficiencies were observed for R1233zd(E) compared to R1234ze(Z) and butene for heat source temperature higher than 80 ˚C. Maximum roundtrip efficiency of 1.61 was achieved at a heat source temperature of 100 ˚C and sink temperature 10 ˚C. Disadvantage of R1233zd(E) is the high boiling point and R1234ze(Z) and butene are flammable fluids unlike R1233zd(E).

Two different fluids are also considered for charging and discharging cycles in some literatures. R1233zd(E) is considered as working fluid for VCHP and with butene in the ORC showed the best performance[22].

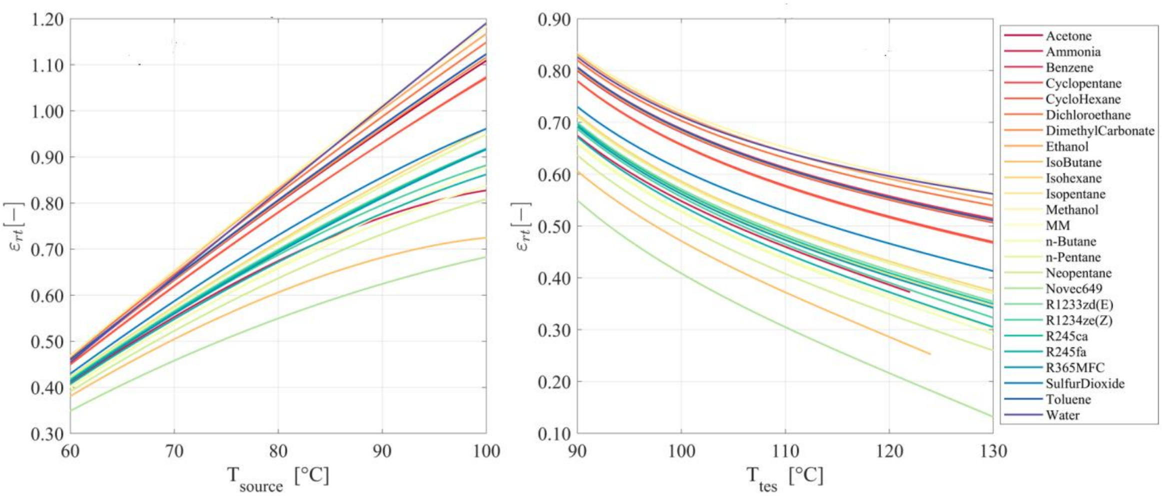


Figure 5-2 Performance of different working fluids for TI-PTES configuration. a) RTE for different source temperature with heat pump lift of 10 K b) RTE for different storage temperature with constant heat source of 80˚C. [17]

A multi-objective analysis was also conducted by Frate [49] for different working fluid pairs considering the roundtrip efficiency, exergetic efficiency and energy density. The best performance was achieved by using pentane for the VCHP and R245fa for the ORC. In a study conducted by Ruoxuan and Huan, for a regenerated configuration of ORC based PTES R245fa-HFO-1336mzz(Z) working fluid pair showed better performance with an optimum roundtrip efficiency of 42.63%[24].

For the latent storage PTES configuration, the working fluid with high enthalpy of vaporization must be chosen. PTES roundtrip efficiency increases with the increase in latent heat of vaporization because high heat can be transferred with a small temperature difference between the working fluid and TES. Butene is considered as a suitable fluid for LTES configurations because PCM are compatible with the isentropic saturation line and critical temperature. The candidates with the most potential for the combination of STES and LTES configurations are the refrigerants related to the isentropic fluids which include R1233zd(E), R-1234ze(Z), R245fa, Butene and R141b[51]. Among these fluids R1233zd(E) is considered to be an environmentally friendly replacement to R245fa.

Group	Refrigerant	T_crit (˚C)	P_crit (MPa)	NBP (˚C)	ODP (-)	GWP (-)
Dry	R-1336mzz (Z)	171.4	2.9	33.4	0	2
	R-365mfc	186.85	3.266	40.2	0	804
	R113	214.06	3.392	47.59	1	6130
	R123	183.68	3.662	27.82	0.02	77
	R245ca	174.42	3.925	25.13	0	693
	Isobutane	134.7	3.64	-11.61	0	20
	Pentane	196.55	3.37	36.06	0	4
	Cyclopentane	238.57	4.57	49	0	11
	Cyclohexane	280.49	4.07	81	-	-
Wet	R-718	373.95	2.206	100	0	0
	Acetone	235	4.7	56	0	<10
	R134a	101.06	4.059	-26.07	0	1430
	Propane	96.7	4.248	-42.09	0	20
	Ethanol	240.75	6.148	78.4	-	-
	Ammonia	132.35	11.33	-33.34	0	0.01
Isentropic	R-1233zd(E)	166.5	3.62	18.3	0.0003	<1
	R-1234ze(Z)	150.1	3.53	9.7	0	<1
	R-245fa	154	3.65	15	0	858
	Butene	146.15	4	-6.3	0	<10
	R-141b	204.35	4.212	32.05	0.11	782
	Toluene	318.6	4.12	110.6	-	-
	R114	145.68	3.289	3.6	1	10040
	R142b	137.11	4.15	32.05	0.12	717
	R236fa	124.9	3.2		0	9810
	R1234yf	94.65	3.381	-29.55	0	<4.4

Table 5-1 List of Working Fluids for ORC based PTES systems [52],[53],[54],[55],[56]

5.2 Turbomachinery

The turbomachinery components play a crucial role in the operation of PTES systems as they are responsible for converting the stored thermal energy to generate electricity. The design and optimization of turbomachinery can significantly impact the performance and efficiency of the overall system. The operating conditions, type of fluid and the desired efficiency are some of the factors to be considered for designing these systems. In addition, the high temperatures and pressures involved in PTES systems require robust and durable turbomachinery.

5.2.1 Compressor & Expander Isentropic Efficiency

Isentropic efficiencies of compressor and expander are often characterized by their ability to produce or extract work compared to an isentropic transformation. Ideally, this efficiency should be closer to one and much lower value is because of the irreversibilities. In most literature constant isentropic efficiencies are considered and this is highly dependent on the roundtrip efficiency. A quasi-linear dependency of the roundtrip efficiency on the isentropic efficiency of the compressor and expander was investigated [30] and for STES or LTES layouts a variation of $\eta_{is,comp/exp}$ from 50% to 90% led to an increase of the roundtrip efficiency about 25%. The levelized cost of storge (LCOS) and the roundtrip efficiencies are more sensitive for the turbine efficiency than the compressor efficiency [44]. For a reversible PTES system[18], small changes in the isentropic efficiencies has a bigger impact in roundtrip efficiency and it is observed that for an isentropic efficiency of 65% to 85% the roundtrip efficiencies increased from 45% to over 70%.

The isentropic efficiency of the machine is directly dependent on the operating conditions such as the pressure, temperature so using a constant efficiency may lead to over emphasize the performances when used for wide range of operating conditions. In the experimental investigation of a small scale ORC based PTES conducted by Dumont [31], a reversible configuration is considered and scroll expander is used indeed the isentropic efficiency varies from 45-75% for the VCHP compressor and from 35-65% for the ORC expander and the reported roundtrip efficiency is 72.5% with a heat source temperature of 75 ˚C and sink temperature 15 ˚C considering a temperature lift of $\Delta T_{storage\ glide,HP}$ = 8 ˚C in the VCHP mode and $\Delta T_{storage\ glide,ORC}$ = 49 ˚C in the ORC mode. Large variations in the isentropic efficiencies of the compressor were also observed for high temperature heat pumps in some literatures.

The selection of turbomachinery components for PTES systems are based on the required power capacity and performance specifications and by utilizing a simplified model can aid in the design and optimization of these components. Lemort [54] proposed a semi-empirical model for scroll compressors and expander, which was later extended to other volumetric machines. This type of machine offers relatively high-pressure ratios for low rotational speeds and low flow rates. The isentropic efficiency of a compressor or expander is dependent on the pressure ratio and temperature at which they are operated. Volumetric machines have been found to have good performance in small scale PTES applications. Scroll and piston expanders are typically used for systems up to kW_{el} capacity, while screw expanders can be used for larger systems up to 200 kW_{el} and for high capacity radial inflow turbines are recommended for operation up to 500 kW_{el} but the use of parallel expanders could be considered to increase the power capacity. Frate also concluded that different fluid inlet qualities led to different volume ratio so multistage expanders are often used when the inlet quality is low whereas a single volumetric expander can be used with high inlet quality. So volumetric machines can handle up to MW range but beyond that threshold turbomachines are more suitable (large scale PTES).

The parameters used to model volumetric machine are the maximum efficiency (η_{mech}), volumetric ratio (r_v), displacement volume V_s and rotational speed (N_s). The built-in volumetric ratio is the ratio between the maximum and minimum volumes of the working chamber in a machine. For a small built-in volumetric ratios high performance is only possible for a narrow range of pressure ratios close to the optimal one which means if the pressure ratio slightly deviated from the optimal pressure ratio the performance tends to decrease. For a machine with a larger built-in volumetric ratio better performance is achieved for a wide range of pressure ratios. Small built-in volumetric ratios are suitable for machines that operate at constant pressure ratio with a relatively stable operating condition. Scroll compressors and expanders are more suitable for small volumetric ratios. Whereas reciprocating compressors and expanders are suitable with larger built-in volumetric ratios. The maximum efficiency only depends on the volumetric machine considered but it's mostly in the range of 98-99%.

In a thermally integrated reversible model proposed by Dumont, it is observed that for the heat pump the thermal power increases with the rotational speed and the ORC electrical power increases with the mass flow rate so the system should be optimized with a tradeoff between the maximizing the roundtrip efficiency and maximizing the electrical power. At high rotational speed, the mechanical losses and pressure drop decreases the performance. It also shows that optimizing the heat pump is more sensitive than the ORC.

In general, a higher volumetric ratio is chosen for PTES systems for better performance and efficiency. Studies have shown that a volumetric ratio of 2-3 is suitable for most PTES applications while higher ratios may be required for systems operating at lower temperatures. So volumetric ratio in the range of 2-6 can be considered for a maximal isentropic efficiency up to 80%.

Table 5-2 Performance parameters of volumetric machines and turbomachinery (C-compressor, E-expander).

Parameters	Volumetric machines				Turbomachinery	
	Screw	Scroll	Vane	Piston	Centrifugal	Axial
Power (MW)	1.5-6	0.05	0.5	15	0.2-50	10-300
Efficiency	50-70 [C];20-80[E]	55-60[C];10-87[E]	17-55[E]	70-90[C];75[E]	70-85[C];40-85[E]	87-92[C];50-90[E]
Flow rate (m3/hr)	100-100000	25-1200	<6000	20-34000	170-850000	50000-850000
Speed (rpm)	1000-20000	300-8000	<6000	200-2000	1800-50000	1500-10000
Pressure ratio1.2	2-4.5	2.5-5	2-3.5	3.5-4.5	1.2	1.1-1.6

5.2.2 Feed Pump Isentropic Efficiency

The Back work ratio is used to express the ratio between the work done by the feed pump and the work provided by the expander in Rankine cycles. For steam rankine cycles, the back work ratio is usually very small so that the feed pump thermodynamic performance has negligible impact on the overall thermal efficiency but in the case of organic rankine cycles with low critical pressures and low heat source temperature it becomes very important[55]. Another major problem with the pump is the cavitation of the fluid [56] which is more common for organic fluids than for water as their latent heat of vaporization and their evaporation temperature are much lower than water. Also, the value of subcooling has an impact on the back work ratio and thermal efficiency of the ORC which should ideally be minimized.

But however, in most of the literatures these parameters are not considered in the sensitivity analysis and a fixed value is used for the feed pump efficiency around 50-80 % while the subcooling are fixed in the range of $\Delta T_{sub} = 3 - 5K$.

5.3 Heat Exchangers

During a heat exchange, the hot fluid is always at a higher temperature than the cold fluid and the temperature difference between them is referred as Pinch point temperature or the pinch. Heat exchangers are the key components for PTES systems as the pinch point temperature difference (ΔT_{pp}) across the heat exchangers should be minimized as much as possible for better performance[30], [39]. In addition to pinch point temperature difference, the fluids on both the sides of the heat exchanges undergo a pressure drop which is linked to the heat exchange. The pressure drop (Δp) may be small but not negligible as it impacts at the end of phase change, and it also affects the saturation temperature of the fluid. Both the pinch point difference and the pressure drop must be considered for an efficient and reliable design. High heat transfer effectiveness and specific heat transfer area per unit volume are desirable for minimizing pinch point difference and compactness.

Minimizing the pinch point temperature difference can be sone by oversizing the heat exchanger. Optimizing the heat exchangers (HE) can be done iteratively as the area and the pinch are calculated based on the inlet temperatures and mass flow rates of the two fluids in the heat exchanger and the heat transfer coefficients. The heat transfer coefficients calculations depend on many parameters such as material properties, state properties and geometry. The pressure drop can be minimized by using a large heat transfer area or by reducing the flow rate. In practice, the pressure drop can be compensated by adjusting the temperature difference.

In an experimental study conducted by Steger [57] , a heat exchanger model was developed for ORC based PTES system with 100 kW reversible configuration and it is found that pinch point temperature is very narrow at the condensation temperature in HP mode and vapour outlet temperature in the ORC mode. Pinch point temperature of 2 K showed better results which can reduce the heat exchanger area of about 46.1% in HP mode and 43.5% in the ORC mode and pressure drop of 0.13 bar were observed for the HP mode whereas 0.33 bar was noted in the ORC mode. In modelling a PTES systems, Dumont [46] also considered a maximum of 2K pinch point difference and evaluated the pressure drops.

The choice of heat exchanger for the PTES systems is very important for an optimal design as it is difficult to choose the size of the other components of the PTES systems. The selection of heat exchanger can be done based on thermal capacity but for an optimal design they require use of computational fluid dynamics tools which are very intensive.

Heat exchangers can be classified based on their structure and design. Some of the heat exchanger models for PTES systems are Shell and tube HE (STHE),Plate HE (PHE), Plate fin HE (PFHE), Printed circuit HE (PCHE), Spiral HE (SHE), coil finned HE (CFHE)[10]. For cost effectiveness and technical maturity STHE are widely used but with a high heat transfer area and they have an optimum pinch point difference of 3-5K. Whereas PCHE are developing technologies which have high thermal effectiveness and low leakages, and they have minimum pinch point difference of 1.5 which boosts the roundtrip efficiency.

In most of the studies, for designing or modelling a PTES system a fixed pinch point difference and fixed pressure drops are used to model the heat exchangers for simplicity. By assuming these fixed parameters, it is possible to evaluate the performance without having to model the system detailed behavior in real time. These assumptions can be used to quickly estimate the system performance and evaluate different configurations. However, the full complexity of the heat exchange cannot be determined with these assumptions like the performance of the heat exchanger can degrade at part load conditions so a more detailed may be required especially in dynamic simulations.

5.4 Cost Distribution and LCOS

For a PTES system, the cost distribution includes several components and levelized cost of storage (LCOS) is the measure of lifetime cost of the PTES system. The LCOS for PTES systems can be compared to other energy storage technologies to determine the most cost-effective solution for a particular application. The LCOS is calculated by determining the total cost of the system and dividing it with the total amount of energy stored and retrieved during their lifetime which provides a values of cost per unit energy stored and retrieved.

In an exergoeconomic study[23], the LCOS decreases with the increase in storage temperature, and it is observed that the system with regenerated configuration in both the sub systems (Heat pump and ORC) showed the lowest LCOS and it is reported as 0.42 $/kWh at T_storage = 90 and 0.29 $/kWh at T_storage = 130. And the highest LCOS was reported for basic configuration in both the sub systems with 0.31$/kWh at T_storage = 130. This shows that adopted a regenerative layout improves the efficiency of the system and LCOS. And, as the roundtrip efficiency increases with a decrease in LCOS. With respect to the increase storage temperature the LCOS decreases and on the other hand increase in storage temperature decreases the roundtrip efficiency. So, with the increase in efficiency LCOS tends to increase so proper thermoeconomic optimization is required to model these systems.

Different configuration of basic and regenerative configurations were compared in a study [24] and it shows that Regenerative configuration in heat pump is more obvious in terms of LCOS but implementing regenerative layout in both the sub-systems reduced the LCOS much lower. Also, the trend of LCOS decreases with the increase in storage temperature from 90 ˚C to 130 ˚C.

High power and working pressure of compressor and turbine lead to high investment cost. And in this study using different working fluids or fluids pairs didn't show much impact on the reduction of cost. So, optimizing the operating conditions like working pressure and eliminating the heat exchange areas can improve economic performance. Also, by adopting the regenerated layout both economic and thermodynamic performance can be improved.

Among the cost distribution of all the components the turbine and compressor costs are the maximum because of the high power and operating pressure ranges. So, reducing power consumption and reducing working pressure has a significant impact on economic performance. In the economic analysis reported in [32], it is reported that without the regenerators, the compressor costs increase by 1.5% from the basic configuration and the turbine cost is almost the same for both the basic and regenerated configurations. A possible optimization to decrease the investment cost is increasing the evaporation temperature of heat pump decrease the size of the storage system because as the evaporation temperature increases the COP increases. This reduces the heat pump power consumption but increasing the evaporation temperature increases the LCOS and worsens the economic performance. It is also observed that if regenerators are not adopted the LCOS increase by 11.5%.

6. Performance Analysis

6.1 Sensitivity Analysis

The operating temperature range of PTES systems has a significant impact on the performance and efficiency of the system. Therefore, it is important to perform a sensitivity analysis to understand the effect of temperature limits on the performance of the PTES system. As a result of the sensitivity analysis, it helps to identify the optimal temperature range of the system to achieve high performance and efficiency.

6.1.1 Roundtrip Efficiency

In general, the roundtrip efficiency of the PTES system is defined as the ratio between the electrical energy consumed during the charging process and the electrical energy generated during the discharge process. This definition is easily comprehensible because electrical energy is pure exergy, a distinction between energetic efficiency and exergetic efficiency is not necessary.

$$\eta_{RTE} = \frac{Electrical\ energy\ generated\ during\ the\ discharge\ process}{Electrical\ energy\ consumed\ during\ the\ charging\ process} = \frac{E_{el,out}}{E_{el,in}}$$

In order to analyze the detailed performance of the PTES system, the individual efficiencies of the sub-systems must be taken into consideration. The PTES systems includes three sub-systems heat pump, thermal energy storage and organic rankine cycle. The three main parameters to calculate the roundtrip efficiency are Coefficient of performance (COP), ORC efficiency and storage efficiency. As a consequence, the product of the subsystem efficiencies must be equal to the roundtrip efficiency of the entire system.

$$\eta_{RTE} = COP_{HP} \cdot \eta_{storage} \cdot \eta_{ORC}$$

The limitation of the roundtrip efficiency is derived from the Carnot efficiencies of the heat pump and the ORC process. Second law efficiency in the range of 50% to 70% can be assumed depending on the size of the plant and its complexity[18] to consider the exergy losses. By considering the second law efficiency the deviation between the real process and the ideal Carnot efficiency can be equaled.

$$COP_{HP,ideal} = \eta_{\text{II}} \left(\frac{T_{Storage}}{T_{storage} - T_{sink}} \right)$$

$$\eta_{ORC} = \eta_{\text{II}} \left(1 - \frac{T_{sink}}{T_{storage}} \right)$$

As discussed in the previous chapters the PTES system can be integrated with low temperature heat sources to increase the roundtrip efficiency of the system. The low temperatures heat sources are waste heat from industrial processes, solar thermal or geothermal systems. The PTES systems using additional heat source are commonly known as thermally integrated PTES systems (TI-PTES). By utilizing low temperature heat source, the compression work of the heat pump is reduced which improves the COP of the heat pump and on the other hand as the temperature difference between the hot and the cold reservoir (environment) is large the ORC efficiency also increases which together improves the overall roundtrip efficiency of the system. If the temperature difference between the heat source and the heat sink is sufficiently large, the work delivered during the discharging process exceeds the work required by the compressor in the charging process by reducing the losses and as a result the roundtrip efficiency exceeds 100%.

The roundtrip efficiency of the thermally integrated PTES systems can be represented as:

$$\eta_{RTE,TI-PTES} = \frac{Electricla\ energy\ generated\ during\ discharging\ process}{Electrical\ enrgy\ consumed\ \ during\ charging\ process\ with\ integrated\ heat\ source}$$

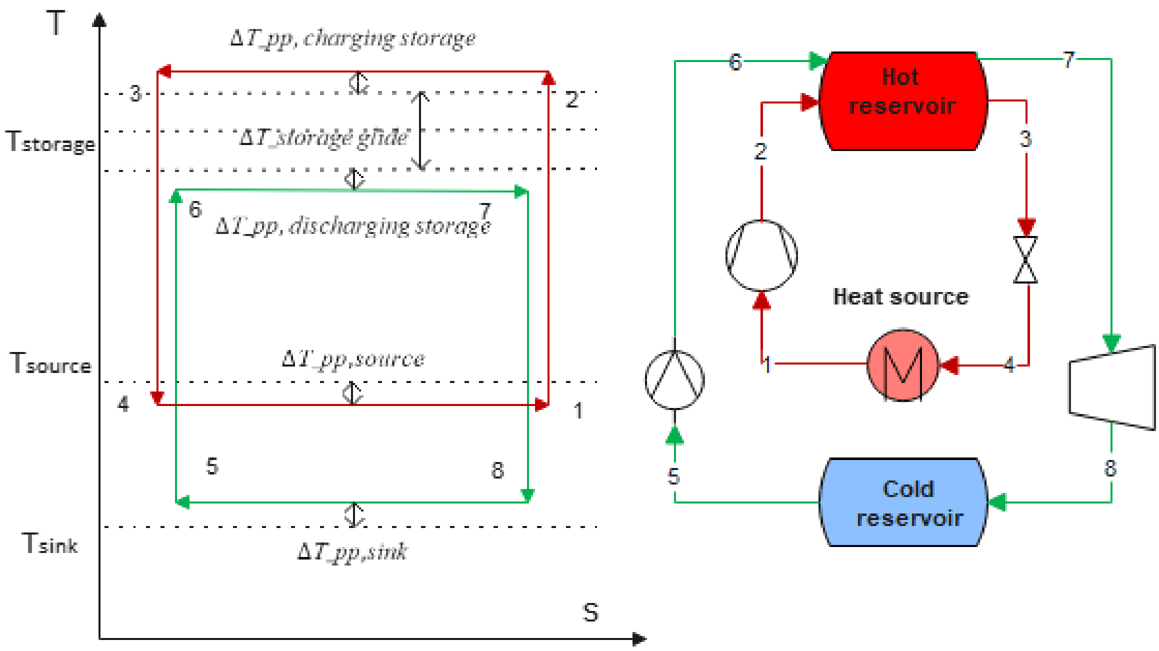


Figure 6-1 Carnot based Thermally integrated PTES cycle.

From an exergy point of view, the exergy losses of the heat pump and ORC can by compensated by the exergy added by the low temperature heat source and this exergy is added to exergy of the stored electricity. By compensating the exergy losses, the roundtrip efficiency increases. The COP and ORC efficiency of thermally integrated PTES systems can be described in terms of Carnot efficiency with a second law efficiency to compensate for the irreversibilities in the system. But however, the heat source is only integrated with the charging cycle in the heat pump, so the COP depends on the heat source and the ORC efficiency remains the same for both ideal and thermally integrated systems.

$$COP_{HP,\,TI-PTES} = \eta_{\mathrm{II}}\left(\frac{T_{Storage}}{T_{storage} - T_{source}}\right)$$

$$\eta_{ORC} = \eta_{\mathrm{II}}\left(1 - \frac{T_{sink}}{T_{storage}}\right)$$

Furthermore, the concept of Thermally integrated system is described with the pinch point differences and storage temperature glide to analyze the system in detail. The pinch point temperature difference represents the heat transfer between the working fluid where $\Delta T_{pp,charging}$ represents the heat transfer between the heat pump and the hot storage whereas $\Delta T_{pp,discharging}$ represents the heat transfer between the hot storage and the ORC system and $\Delta T_{pp,sink}$ represents the heat transfer between the ORC system and the environment.

The storage glide $\Delta T_{storage\ glide}$ corresponds to the temperature difference between the hot and cold parts of the thermal storages. The temperature difference between the heat source or waste heat and the storage temperature is defined as the lift of the heat pump which should be minimized to increase the COP of the heat pump. With the pinch point temperature difference s and the storage glides, the COP of the heat pump and the ORC efficiency can be represented as:

$$COP_{HP,\,TI-PTES} = \eta_{\text{II}}\left[\frac{\left(T_{storage} + \Delta T_{pp,charging} + {}^{\Delta T_{storage\ glide}}/_{2}\right)}{\left(T_{storage} + \Delta T_{pp,charging} + {}^{\Delta T_{storage\ glide}}/_{2}\right) - (T_{source} - \Delta T)}\right]$$

$$\eta_{ORC} = \eta_{\text{II}}\left[\frac{\left(T_{storage} - \Delta T_{pp,discharging} - {}^{\Delta T_{storage\ glide}}/_{2}\right) - (T_{sink} + \Delta T)}{(T_{storage} - \Delta T_{pp,discharging} - {}^{\Delta T_{storage\ glide}}/_{2})}\right]$$

$$\eta_{RTE,TI-PTES} = COP_{HP,\,TI-PTES} * \eta_{storage} * \eta_{ORC}$$

The sensitivity analysis is carried out to understand the different parameters to determine the optimal working conditions in order to improve the overall system efficiency. By understanding these parameters and optimizing the system according to it is possible to improve the design of the PTES system and reduce the purchase costs and even avoid the non-idealities that degrade the overall performance. For the sensitivity analysis both the sensible and latent heat storage are considered, and the performance of the corresponding storage systems are observed.

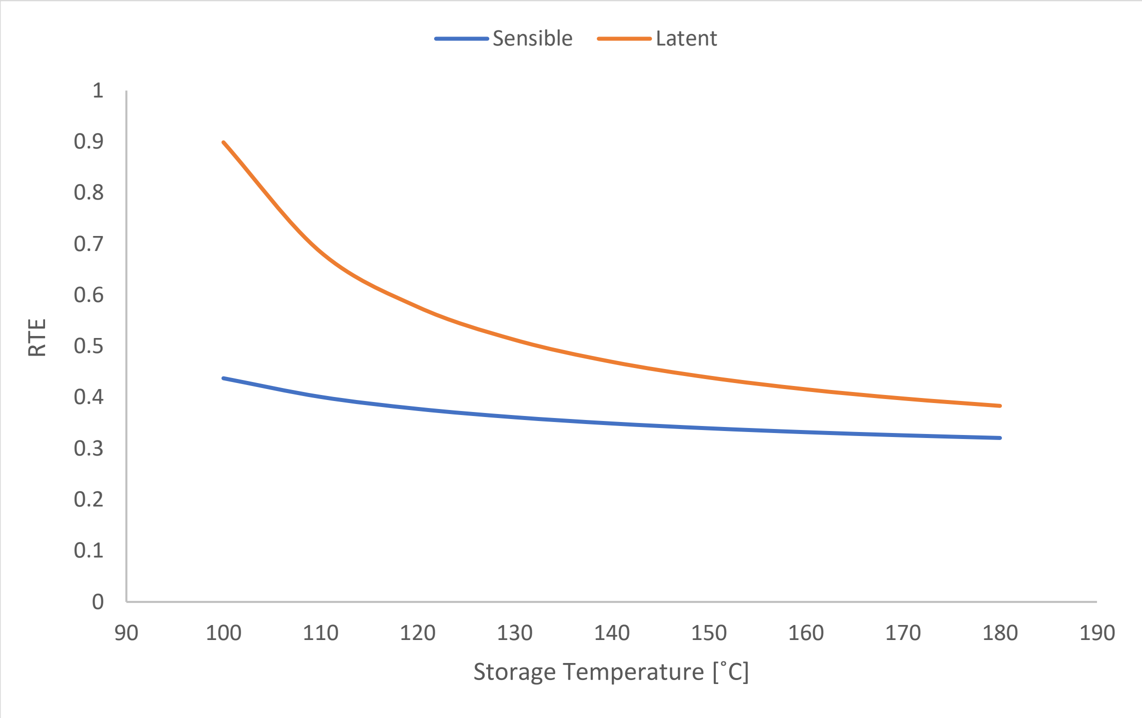


Figure 6-2 Roundtrip efficiencies as a function of Storage temperature with constant heat source of 90˚C and sink temperature 15˚C. Pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K latent (latent) was assumed.

The roundtrip efficiency is determined by the ORC efficiency and COP of the heat pump. The COP and ORC efficiency is obtained for different storage temperature with a constant heat source of 90˚C and sink temperature 15 ˚C and a constant pinch point temperature difference of 5K.

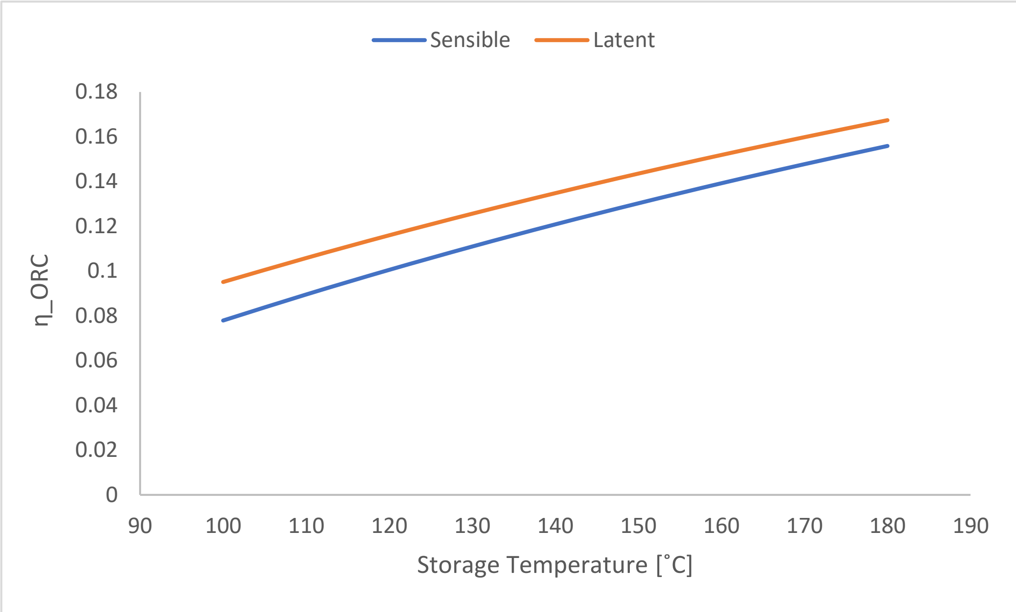


Figure 6-3 ORC efficiency as a function of Storage temperature with constant heat source of 90˚C and sink temperature 15˚C. Pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K latent (latent) was assumed.

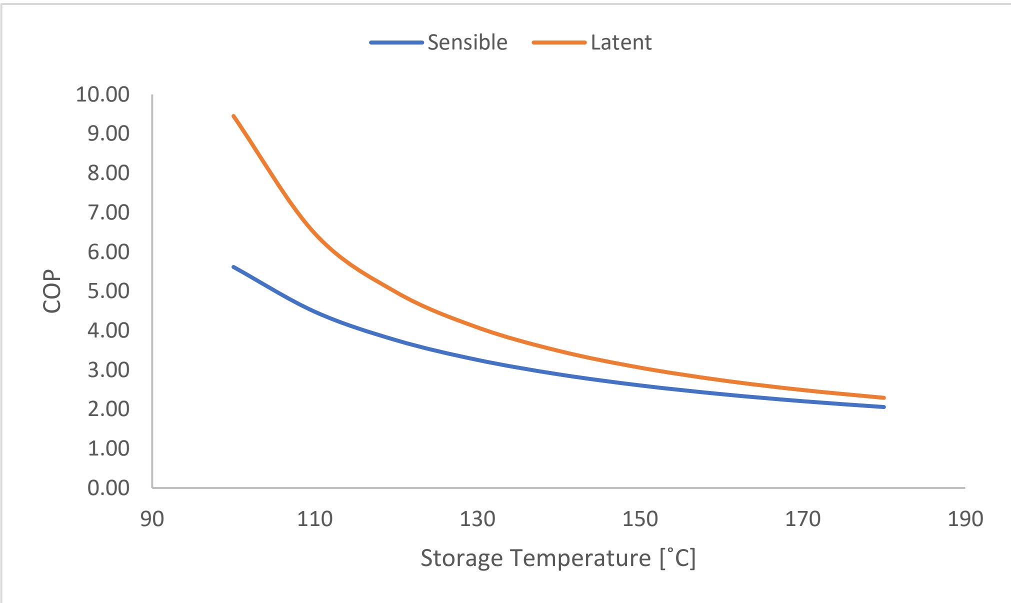


Figure 6-4 COP as a function of Storage temperature with constant heat source of 90˚C and sink temperature 15˚C. Pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K latent (latent) was assumed.

It can be observed that the increase in the storage temperature decreases the trend of COP of the heat pump and in contrast the ORC efficiency increases with the increase in storage temperature. The maximum to minimum ratio of COP is observed to 4.18(latent) and 2.44 (sensible) and whereas the maximum to minimum ratio of ORC efficiency is about 1.7 (latent) and 2 (sensible). As a consequence, the impact of COP is higher than the ORC efficiency and therefore the roundtrip efficiency decreases with the increase in the storage temperature.

It can be noted that the PTES system with latent heat storage shows better performance because of the storing of heat as an isothermal process in a phase change material. The roundtrip efficiency increases with the increase in latent heat of vaporization because high heat can be transferred with a small temperature difference between the working fluid and TES. The exergy losses can be minimized as the heat is transferred with a small temperature difference in the case of latent storage. While the storage glide is higher in the case of sensible storage with a assumed value of 30K.

In a study by Liu [32] It can be observed that increasing the storage temperature decreases the roundtrip efficiency and the highest roundtrip efficiency is observed at lower storage temperature. Increase in storage temperature increases the ORC efficiency but conversely on the other hand the COP of the heat pump decreases with the increases in storage temperature. But however, it is observed that the impact of COP is higher than the ORC efficiency which leads to decrease in roundtrip efficiency. In comparison of basic and regenerative sub-systems for PTES, it is also noted that the roundtrip efficiency decreases with the increase in storage temperature and a maximum roundtrip of at storage temperature of 90 °C[23].

6.1.2 Heat Source Temperature

A sensitivity analysis is conducted to analyze the impact of the heat source temperature (T_{source}) on the COP of the heat pump and the roundtrip efficiency of the PTES system. The sink temperature does not impact the COP of the heat pump so a constant temperature of 15 ˚C is assumed. The pinch point difference for associated with the storage ($\Delta T_{pp,charging}, \Delta T_{pp,discharging}$), sink ($\Delta T_{pp,sink}$) and the source ($\Delta T_{pp,source}$) is assumed to be 5K and the flowing plots are obtained. The storage glide ($\Delta T_{storage\ glide}$) for the sensible storage is assumed to 30K and for latent heat storage 0K as the heat is stored with phase change in isothermal conditions.

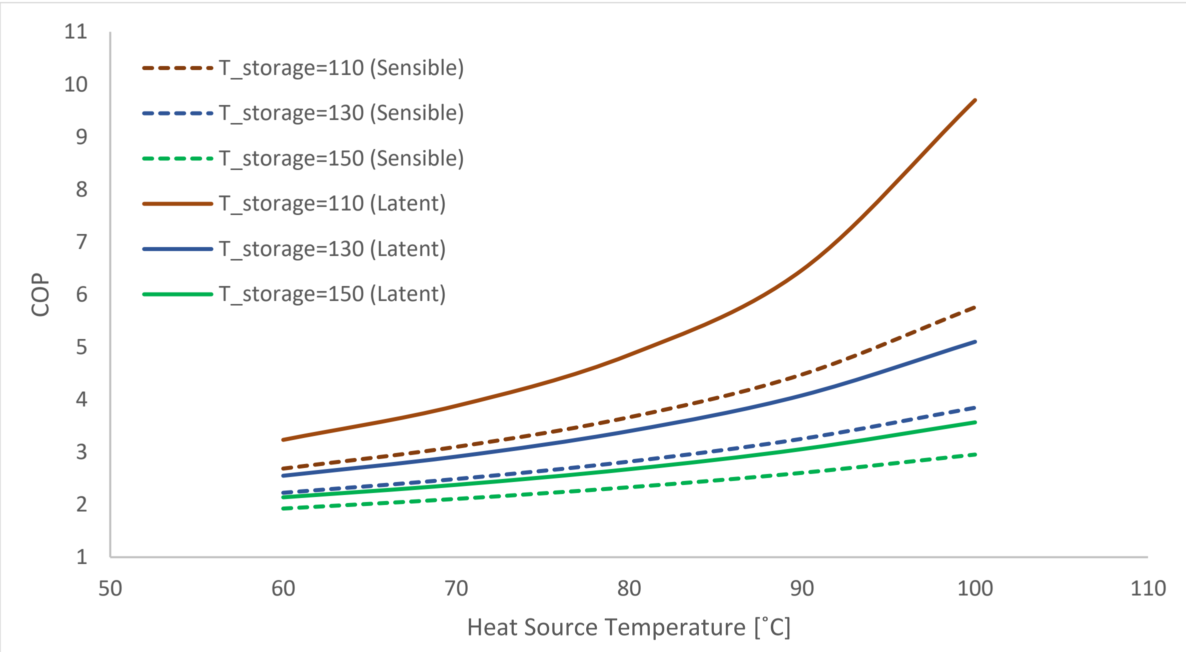


Figure 6-5 COP as a function of heat source temperature for varying storage temperatures with constant sink temperature 15 ˚C with pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K (latent).

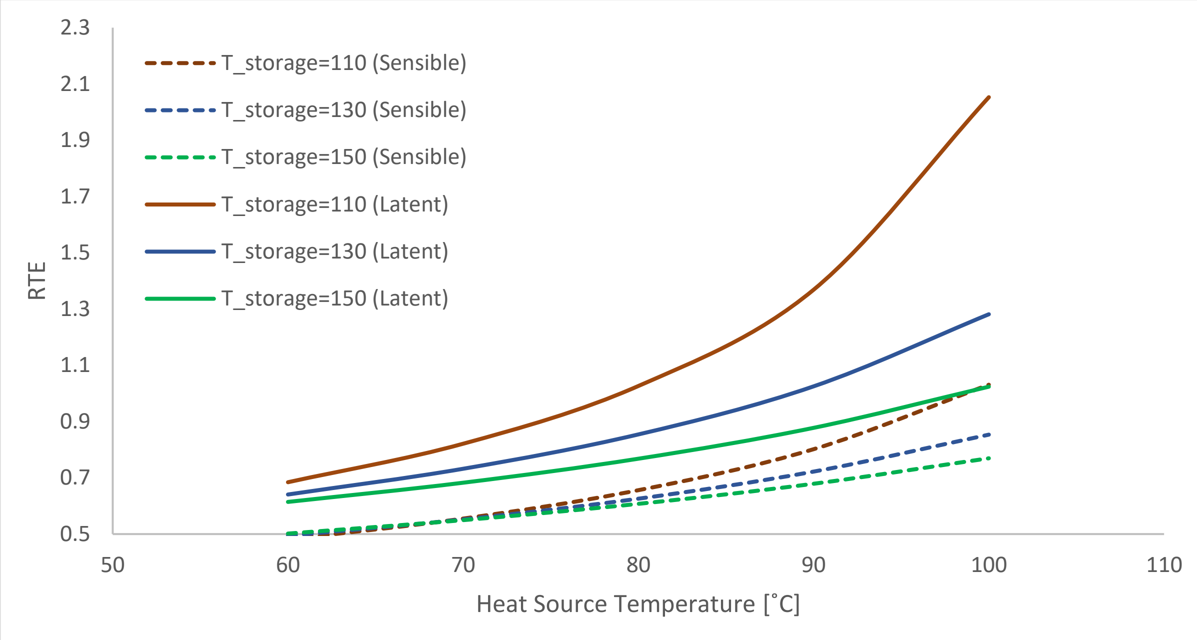


Figure 6-6 RTE as a function of heat source temperature for varying storage temperatures with constant sink temperature 15 ˚C with pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K (latent).

From the sensitivity analysis it can be observed that the maximum COP is obtained for lower storage temperature for the highest heat source temperature and similarly the roundtrip efficiency increases with the increase in heat source temperature. As a consequence, it is clear that the heat pump lift (temperature difference between the heat source and storage temperature) should be minimum which minimizes the compressor work to achieve high COP and roundtrip efficiencies. With a constant sink temperature and increasing the heat source temperature as the difference is sufficiently large the work output of ORC exceeds the work done by the compressor of the heat pump and therefore very high roundtrip efficiencies can be achieved. The PTES system based on latent heat storage shows better performance as the heat is stored with a phase change in a isothermal process.

In thermally integrated PTES, low temperature heat sources are used to boost up the performance of the heat pump thereby improving the efficiency of the ORC cycle to have high roundtrip efficiency. The RTE increases with the increase in the heat source temperature but however, In a study conducted by Frate [38] different organic working fluids were investigated and it is observed increase in RTE drops after the critical temperature of the working fluid. This effect cannot be observed with Carnot efficiency as it does not depend on the nature of the working fluid.

With a constant heat source, we can observe that the RTE increases as the storage temperature tends to heat source temperature which emphasize that the heat pump working should be designed the minimum operating temperature difference. Heat transfer between the HP and heat source temperature is limited by a minimum pinch point temperature difference. The temperature difference between the heat source temperature and the storage temperature should be as close as possible with a minimum operating temperature difference to achieve high COP and roundtrip efficiency.

The charging cycle with heat pump has a higher impact of variation of the heat source temperature as it is directly related to the COP so variations in heat source temperature have a higher impact on the roundtrip efficiency. In a parametric study of maximum roundtrip efficiency of 125% was achieved at source temperature of 100 ˚C and 15 ˚C sink temperature[39].

In the multi-criteria thermodynamic analysis [58], the relationship between the heat source temperature, pressure ratio of compressor and efficiencies were studied. It is observed that at 110 ˚C the heat pump achieved a COP of 9.91 with a compression ratio of 1.85 and ORC efficiency of 1.26 for Butene with a constant sink temperature of 20˚C.

In a thermally integrated reversible model proposed by Dumont[46], considered a minimum operating temperature difference (difference between the heat source and the storage temperature) as 10K for a storage capacity of 10 kWh.

6.1.3 Sink Temperature

The impact of the sink temperature should also be assessed with respect to the ORC efficiency, and it is not considered for the COP of the heat pump as the waste heat source is utilized in the heat pump. But however, the difference between the sink temperature and the heat source temperature has an impact on the overall roundtrip efficiency as the ORC performance may dominate the work done by the heat pump if the temperature difference between them is sufficiently large. So, the sensitivity analysis of the change in sink temperature is also observed with respect to the roundtrip efficiency.

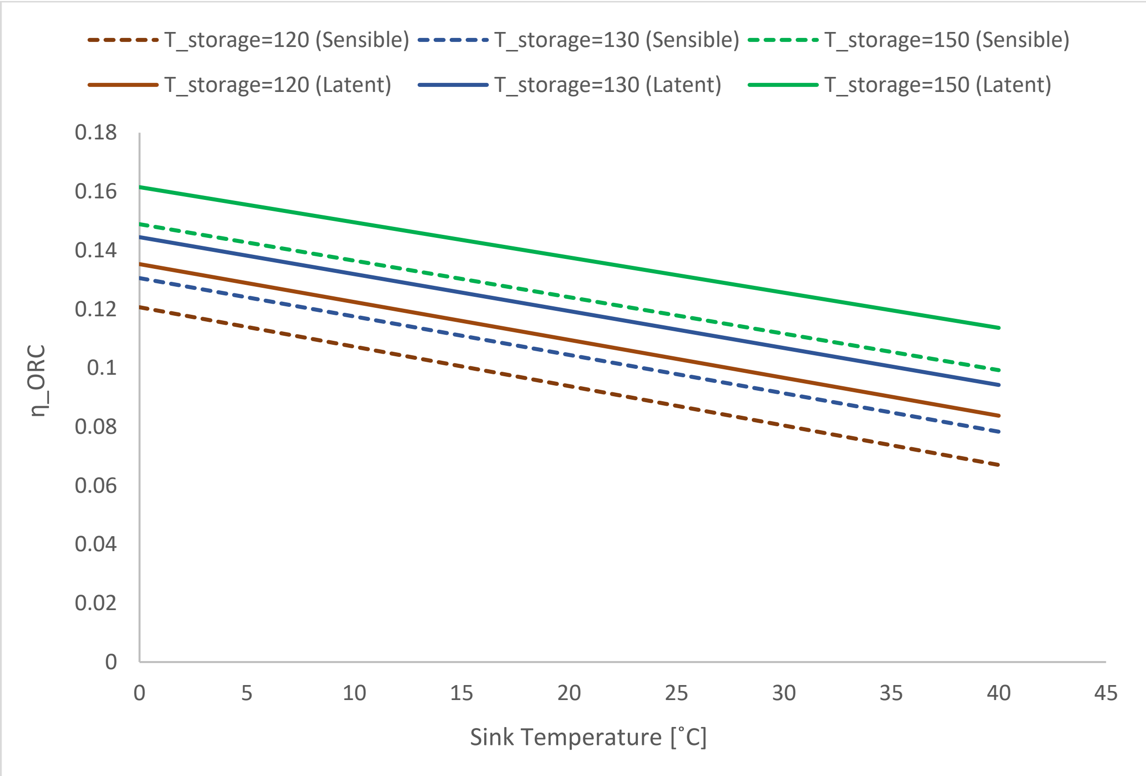


Figure 6-7 ORC efficiency as a function of sink temperature for varying storage temperatures with constant heat source temperature 90 ˚C with pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K (latent).

The impact of sink temperature is observed for variation of storage temperature from 120 ˚C to 150 ˚C with a constant heat source temperature of 90 ˚C and pinch point difference of 5K but however the heat pump lift increases with the increase in storage temperature to consider the operation of heat pump for a wide range of storage temperatures. The storage glide for sensible storage is assumed to be 30K and the latent storage 0K. It can be observed that the ORC efficiency is a linear function of sink temperature, and the ORC efficiency tends to decrease with the increase in sink temperature. The maximum ORC efficiency is achieved at higher storage temperatures as the temperature difference between the storage and sink is sufficiently large.

For a constant heat source temperature of 90 ˚C, the roundtrip efficiency decreases with an increase in sink temperature but however average sink temperature of 15 ˚C to 25 ˚C is assumed for efficient working of the ORC with over expansion of the expander. Also, higher the temperature difference between the heat source and the sink temperature in the case thermally integrated PTES yields high roundtrip efficiency.

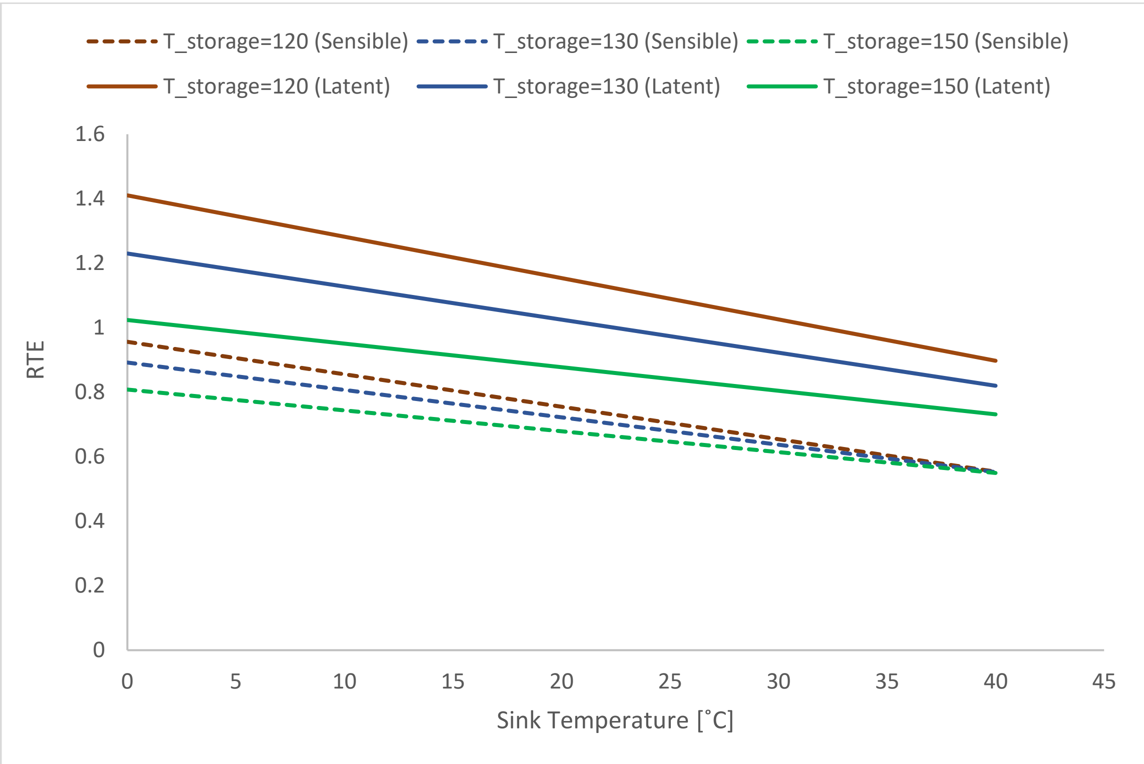


Figure 6-8 RTE as a function of sink temperature for varying storage temperatures with constant heat source temperature 90 ˚C with pinch point differences of 5 K and storage glide of 30 K (sensible) and 0 K (latent).

In a parametric study[39] it was observed that increasing the sink temperature from 15 to 40 reduced the roundtrip efficiency from 0.8 to 0.3. The ORC efficiency is a linear function of sink temperature but for power ratio between a hot and the cold reservoir remains constant for any heat source temperature.

In a thermally integrated reversible model proposed by Dumont[46], Regarding the sink temperature glide a lower glide results in over expansion of the expander which decrease the efficiency so constant sink glide of 10K can be considered.

In the multi-criteria thermodynamic analysis [58], with a constant heat source temperature of 90 ˚C, the roundtrip efficiency decreases with an increase in sink temperature and maximum roundtrip efficiency of 103.3% was observed at 15 ˚C.

6.1.3 Pinch Point Difference

The influence of the pinch point differences associated with the evaporator and condenser section of the heat pump and the ORC systems should by analyzed. The evaporator section of heat pump is where the waste heat source is added to the heat pump and the pinch point difference is $\Delta T_{pp,source}$.

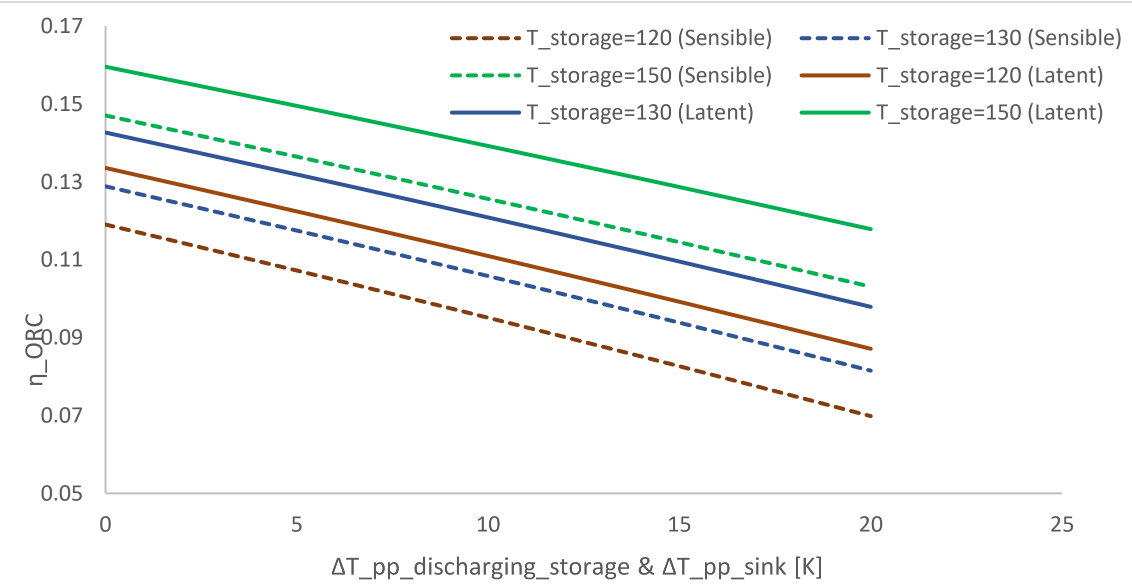


Figure 6-9 ORC efficiency as a function of $\Delta T_{pp,discharging,storage}$ & $\Delta T_{pp,sink}$ for varying storage temperatures with constant heat source temperature 90 ˚C and sink temperature of 15˚C. Storage glide of 30 K (sensible) and 0 K (latent) were assumed.

The condenser of the heat pump exchanges the heat with the hot storage with minimum pinch point difference $\Delta T_{pp,charging}$. Similarly in the ORC system the condenser exchanges heat with the environment and evaporator utilized the heat stored in the hot storage and the corresponding pinch point differences are $\Delta T_{pp,sink}$, $\Delta T_{pp,discharging}$.

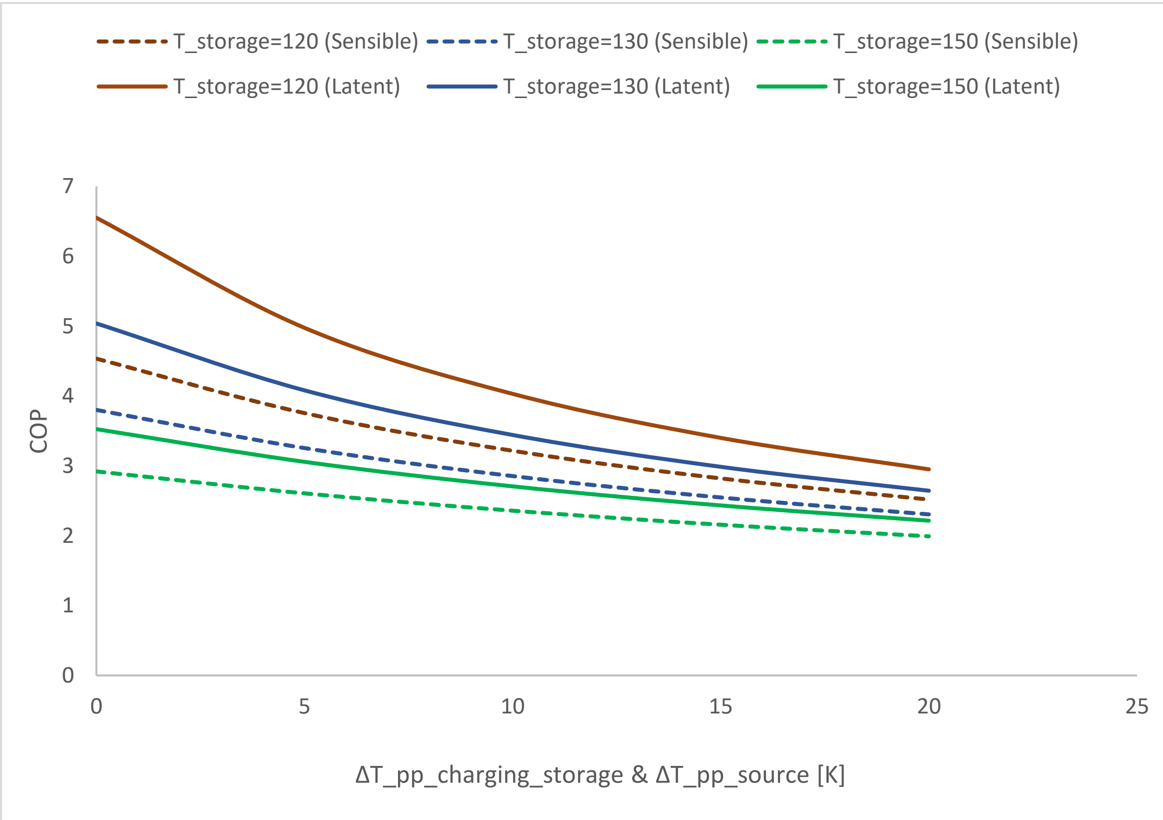


Figure 6-10 COP as a function of $\Delta T_{pp,charging,storage}$ & $\Delta T_{pp,source}$ for varying storage temperatures with constant heat source temperature 90 ˚C and sink temperature of 15˚C. Storage glide of 30 K (sensible) and 0 K (latent) were assumed.

Increasing the pinch point differences decreases both the COP of the heat pump and ORC efficiency and consequently the roundtrip efficiency also decreases. The pinch point difference should be minimized to improve the performance of the PTES system. The pinch point difference can be minimized by oversizing the heat exchangers. Increase in pinch point difference also increases the losses associated with the heat transfer. The losses associated with the heat transfer can be minimized by adopting a regenerative layout. As a result, the evaporator and condenser should be designed with minimum pinch point difference to obtain high roundtrip efficiencies.

Regarding the pinch point difference of the heat sink $\Delta T_{pp,sink}$, a lower temperature difference results in over expansion of the expander which decreases the ORC efficiency so a constant temperature difference of 10K can be assumed. Also, in general a minimum pinch point temperature difference of 2-5K is better.

6.1.4 Storage Temperature Glide

The storage glide $\Delta T_{storage\ glide}$ corresponds to the temperature difference between the hot and cold parts of the thermal storages. The impact of the storage glide is analyzed with respect to the roundtrip efficiency with a constant heat source temperature of 90˚C and sink temperature of 15 ˚C. The pinch point differences are assumed to be 5K except the $\Delta T_{pp,sink}$ which is maintained with a constant temperature difference of 10K.The sensitivity analysis is performed for different storage temperatures, and it can be observed that increase in the storage glide decreases the roundtrip efficiency and the maximum roundtrip efficiency is observed at the lowest storage temperature and low glides.

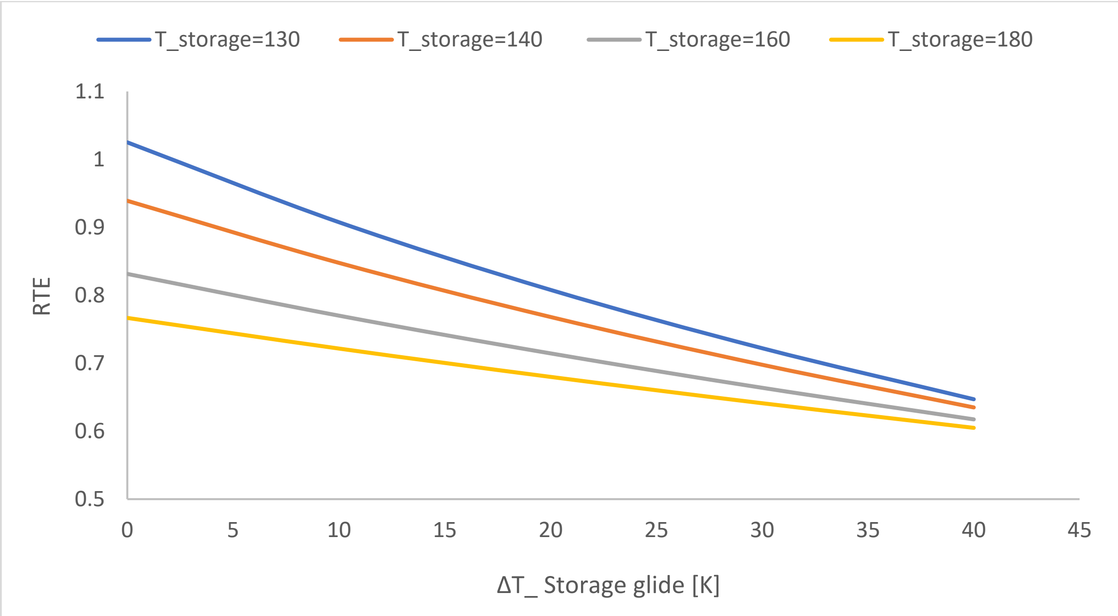


Figure 6-11 Roundtrip efficiency as a function of $\Delta T_{storage\ glide}$ for varying storage temperatures with constant heat source temperature 90 ˚C and sink temperature of 15˚C. Storage glide of 30 K (sensible) and 0 K (latent) were assumed.

As a consequence of high roundtrip efficiency at low storage temperature and low glides it is clear that the COP of the heat pump increases with as the compressor work is reduced if the storage temperature is close to the heat source temperature with low glides. In contrast the ORC efficiency increases with high storage temperatures and high glides, but the COP of the heat pump has higher impact in the overall roundtrip efficiency and therefore the roundtrip efficiency decreases. The increase in storage glides is also associated with the thermal losses in the storage.

In the case of sensible storage, the heat is stored in a liquid or solid material which involves a heat transfer between the working fluid and the storage material. In the process if heat transfer some heat is lost but however in the case of latent heat storage the heat is stored in a phase change material, so heat is stored as a result of isothermal process with very small temperature difference between the working fluid and the phase change material. But selecting a suitable combination of phase change material and working fluid very high round trip efficiencies can be achieved with ORC based PTES systems.

In a parametric study [38] with the increase in the temperature difference between the storage and working cycles, the roundtrip efficiency decreases and a maximum efficiency of 89% was achieved with source temperature 100 ˚C and sink temperature 15 ˚C.

In a thermally integrated reversible model proposed by Dumont [31], low glides achieved high COP and whereas in the case of ORC efficiency it is the vice vera because higher the difference in the ORC higher the efficiency, so ORC efficiency increases with the increase in storage temperature. In the experimental investigation of a small scale ORC based PTES conducted by Dumont, a reversible configuration is considered and scroll expander is used indeed the isentropic efficiency varies from 45-75% for the VCHP compressor and from 35-65% for the ORC expander and the reported roundtrip efficiency is 72.5% with a heat source temperature of 75 ˚C and sink temperature 15 ˚C considering a temperature lift of $\Delta T_{storage\ glide,HP}$ = 8 ˚C in the VCHP mode and $\Delta T_{storage\ glide,ORC}$ = 49 ˚C in the ORC mode.

In the multi-criteria thermodynamic analysis [58], it is observed that the storage glide has higher impact on the COP of HP. Increase in the storage glide reduced the COP of the heat pump and a maximum COP was achieved with the lowest storage glide of with a heat source temperature of 100 ˚C and butene as the working fluid. Also, with the increase in storage glide and at high source temperature, the exergy loss is more, and the roundtrip efficiency decreases. In a study with reversible PTES systems[18], highest roundtrip efficiencies were observed at low storage temperature and low storage glides.

6.2 Exergy Distribution and Exergy Efficiency

The exergy efficiency of the PTES system depends on the roundtrip efficiency and exergy of the heat pump and storage. The roundtrip efficiency decreases with the increase in thermal storage temperature similarly the exergy efficiency decreases with the increase in round trip efficiencies. The exergy increase in terms of thermal storage temperature is higher for heap pump than thermal storage. At lower storage temperature heat pump exergy rate overcomes the rate of decrease in the roundtrip efficiency so the exergy efficiency increases. After reaching the peak the exergy efficiency tends to fall because the rate of decrease of roundtrip efficiency overcomes the rate of increase of the heat pump exergy. It is reported that [32], with R1233zd(E) as working fluid a maximum exergy efficiency of 19.1% is mentioned. Using a regenerator improves the performance of the exergy efficiency by reducing the exergy destruction involved with throttling valve of heat pump and economizer section of ORC. But using a regenerated configuration in the heat pump shows a higher impact than the regenerated configuration in the ORC.

In thermodynamic analysis of PTES with both sensible and latent heat storage were used [58], and it is observed that maximum exergy loss were observed for sensible storage 41.52 kWh and latent heat 36.45 kWh which totally accounts for 65.74% of the total exergy loss. The heat stored in the latent storage is much less than the sensible storage but still sensible storage accounts for high exergy loss because the working medium in the latent storage absorbs heat and phase change takes place which is close of isothermal heat absorption.

In the numerical investigation of PTES with thermal integration[39], it is observed that highest exergy loss is for sensible storage which accounts for 10.8% while latent heat storage only accounts for 2.9%. Other major exergy losses are observed at the heat transfer of evaporator and condenser.

The highest exergy losses are associated with the heat exchanger and throttling valve, and this can be reduced by a adopting a regenerative layout thereby promoting both thermodynamic and economic performance. The regenerators decrease the exergy destruction associated with the heat exchanger by lowering the temperature difference between the inlet and outlet of the heat exchanger[24].

With the increase of storage temperature exergy efficiency first increases and then decreases after reaching the peak. The exergy efficiency is related to the roundtrip efficiency, heat pump exergy and heat storage exergy but the heat pump exergy grows more rapidly than the storage exergy. Adopted regenerators showed better results but regeneration in heat pump configuration is more obvious than the ORC. And the heat pump with regenerated configuration achieved a maximum exergy efficiency of 19.12%[23].

7 CONCLUSION:

Global electricity generation is highly dependent on conventional fossil fuels and has a negative impact on the environment. The change from fossil fuels to renewable energy sources has been the top priority. Energy storage systems is the key factor for the feasibility of clean and sustainable energy by increasing the share of renewable power production.

Energy storage systems can be broadly classified into mech, electrical, electro-chemical, thermal and chemical based on the storage principles. Different energy storage systems can be integrated with different grid services. For short term storage systems which can manage the supply and demand in real-time which can respond quickly. for ancillary services black start capability, frequency, and voltage. But long-term storage is very important for renewable integration which includes Pumped thermal energy storage systems (PTES).

PTES is an energy storage which transforms electricity into heat and stores it as thermal energy. This stored energy is transformed back into electricity when needed. The PTES can be mainly classified into brayton based PTES and rankine based PTES and rankine based systems can be further divided into conventional steam cycles, organic rankine cycles and Transcritical cycles.

PTES based on Organic Rankine cycles is capable to achieve high energy densities and store heat at a much lower temperature compared to brayton cycle. Organic rankine cycles are characterized by low efficiency without thermal integration whereas thermally integrated ORC based PTES systems (TI-PTES) shows much better performance when used with low temperature heat sinks and satisfactory temperatures differences between the source and the sink.

TI-PTES systems use low temperature heat sources like waste heat from industries, geothermal heat, etc. The TI-PTES systems can be classified into hot storage configuration and cold storage configuration. In hot storage configuration waste heat is used to store heat at hot reservoir and whereas in the cold storage configuration the heat is stored in the cold storage by releasing heat to the environment. The hot storage configuration showed better performance when compared to cold storage configuration.

For the storage phase of the PTES systems, three types of thermal energy storages (TES) can be used which includes sensible TES, latent TES, and thermochemical energy storage. Also, in this study PTES system configuration based on these thermal storages are discussed.

In general, the ORC based PTES systems have vapor compression heat pump (VCHP) in the charging phase and ORC in the discharging cycle but for small or medium scale application a reversible PTES system can be considered. A reversible cycle is used by compromising the performance in terms of cost. The reversible PTES can be classified into partly reversible and fully reversible PTES configurations. In partly reversible configuration similar HEX are used for VCHP and ORC configuration but in this case same working fluid. Similar mass flow rates can be achieved during charge and discharge by selecting the nominal charge and discharge durations. In fully reversible configuration volumetric machines can be used which acts the same machine for both the compressor and the expander. A scroll compressor is preferred because of its high efficiency and simple structure. Regarding the configuration, since it is fully reversible, they use the same heat exchangers and the only components that must integrated separately are the expansion valve and pump of ORC.

The next configuration is PTES with sensible thermal energy storage which is very simple, and heat can be stored in liquid or solid material. Energy is stored based on the heat capacity of the material and for the solid material rocks or concrete is used but this requires an additional heat exchanger system for the storage. So, water stank storage is the most widely stored sensible heat. This type of system can be designed with one or two reservoirs which is a tradeoff between the cost and performance. The entropy generation or irreversibilities in this type of system is due to heat losses and pressure drops in heat exchangers. To improve the system's overall performance, especially because of the temperature gradient in the expansion valve after storing heat, this can be overcome by adapting a regenerative configuration in the sub-systems.

In Latent heat storage energy is stored based on the latent heat of evaporation with a phase change. The most used phase change material is a eutectic mixture of lithium nitrate and potassium nitrate. PCM materials are cheap and water ice can be used as PCM material for cold storage could be an effective solution. As the heat is stored with a phase change in isothermal conditions the heat exchange takes place with a small temperature difference which also reduces the losses involved in storing the heat. As stated, before the maximum entropy generation is in the expansion valve and evaporator section of VCHP and ORC respectively. So, a regenerated configuration can be used to preheat the working fluid before the compressor inlet and similarly a regenerated configuration can also be adopted in the power cycle to preheat the working fluid before evaporation using thermal storage which reduces the temperature differences, increases mass flow rates and the overall efficiency.

A hybrid TES can be used with both the STES and LTES to further improve the efficiency of the system. Using two storage increases the energy storage capacity and performance but with an increase in cost. In the charging phase at first the heat is first stored in the LTES and the passes through sensible heat storage where the working fluid is further sub-cooled and reduces the temperature difference in the expansion valve which decreases the losses. Also, a regenerated configuration can be adopted in this system.

TCES are currently in the research phase, but they have high potential for long term energy storage as energy is stored in a chemical reaction. The compressed heat is transferred by the condenser, and which initiates an endothermic reaction at 150 ˚C and heat is stored. During discharging the heat is released by an exothermic reaction at 130 ˚C which is used to run the expander and the condensation temperature is around 60-70 ˚C which can be used for district heating purposes. Also, in this configuration a regenerative configuration can be adopted.

Considering the technological limitation of the heat pump and the heat engine, the maximum temperature reached by these systems can be set in the range of 100˚C to 150 ˚C. But however, the maximum temperature is also determined by the type of working fluid and the type of storage medium used. The environment or ambient acts as the heat sink 15 ˚C for the ORC cycle. The low temperature heat source or waste heat is in the range of 80˚C to 100 ˚C.

The selection of working fluid is the key parameter to minimize the entropy generation and it depends on different parameters. Dry or isentropic fluids are preferred to avoid wet compression or expansion. Using a single working fluid is beneficial to optimize the system with the same heat exchangers but each sub-system has higher performance with different fluids. Sub-critical ORC based PTES is more suitable for utilizing low-temperature heat sources so a minimum critical temperature of 120 ˚C can be considered to have good range of upper temperature limit. lower pressure limits higher than the ambient and the present heat pump systems can work until a maximum temperature of 30 bar. High density of the working fluid results in lower flow rates which leads to smaller components. R1233zd(E), R-1234ze(Z), R245FA, butene are the fluids that have been reported with good performance and are the best candidates for the combination of sensible and latent heat storage system. While Butene is considered a suitable fluid for LTES configurations because PCM is compatible with the isentropic saturation line and critical temperature. R1233zd(E) stated to have better efficiency and lower operational pressure values which makes them suitable for PTES systems and is an environmentally friendly replacement to R245fa.

Isentropic efficiencies of compressor and expander are often characterized by their ability to produce or extract work compared to an isentropic transformation. RTE is dependent on machine efficiencies and isentropic efficiencies are directly dependent on the operating conditions of temperature and pressure so considering a constant efficiency may overrate the performance for different operating conditions. Isentropic efficiency of 70% to 80% can be considered. For large scale systems centrifugal or axial machines can be used while for small or medium scale systems up to 1MW volumetric machines can be chosen. For very small-scale machines in kW range screw and scroll could be a better good option because of small flow rate and simple structure also scroll compressor has good efficiencies for small flow rates. Screw expanders can be used for systems up to 200 kW.

Back work ratio is defined as the ratio between the work done by the pump and work provided by the expander. In general, for steam cycles this ratio is small and negligible but for ORC they have a significant impact with low critical pressures. Another problem with pump performance is cavitation as the organic fluid evaporation temperature is much lower than water. The isentropic pump efficiency of 50%-80% and sub cooling temperature difference of 3-5 K is reported.

Heat exchangers are the key components in the PTES system and the pinch point difference should be minimized to improve the performance and at the same time the heat exchangers also undergo a pressure which is irreversible and causes entropy generation. The minimum pinch point temperature difference of 2-5 K is considered as ideal. The pinch point difference can be minimized by oversizing the heat exchanger. Increasing heat transfer surface area or reducing the flow rate can minimize the pressure drop in the heat exchanger.

From the sensitivity analysis performed on different parameters based on the Carnot efficiency and second law efficiency. As a result, it is observed that the roundtrip efficiency is maximum for lower storage temperature and tends to decrease with increase of the storage temperature. This is because the COP of the heat pump decreases with the increase in storage temperature while in contrast the ORC efficiency increases with the increase in storage temperature. But however, it is noted that the impact of COP is higher than the ORC efficiency and thus the roundtrip efficiency decreases. The impact of the heat source temperature is highly reflected in the heat pump while the ORC efficiency does not include heat source temperature. The heat pump lift (temperature difference between the heat source and storage temperature) should be minimum to achieve high COP. Also, the difference between the heat source and the sink temperature is sufficiently large, the work done by the ORC exceeds the Work done by the compressor of the heat pump and thus high roundtrip efficiencies can be achieved. As the heat source temperature increases the COP and the roundtrip efficiency increases.

The sink temperature is considered as the ambient temperature for the PTES systems. The ORC efficiency is a linear function of sink temperature, and the roundtrip efficiency decreases with an increase in sink temperature thus a constant sink temperature of 15˚C to 25 ˚C can be considered.

The increase in the pinch point temperature difference of the evaporator and condenser is analyzed and with the increase in pinch point difference the roundtrip efficiencies decrease because of the losses or irreversibilities associated with the heat exchange so it should be minimized while designing the systems.

The increase in the storage glide decreases the roundtrip efficiency and the maximum roundtrip efficiency is observed at the lowest storage temperature and low glides. In the case of sensible storage, the heat is stored in a liquid or solid material which involves a heat transfer between the working fluid and the storage material. In the process if heat transfer some heat is lost but however in the case of latent heat storage the heat is stored in a phase change material, so heat is stored because of isothermal process with very small temperature difference between the working fluid and the phase change material. But selecting a suitable combination of phase change material and working fluid very high round trip efficiencies can be achieved with ORC based PTES systems.

7.1 Future Developments

In this thesis, the pumped thermal energy storage based on organic rankine cycles has been studied in detail but still there is a huge scope to analyze these systems in many aspects.

The sensitivity analysis made in this study is based on the Carnot efficiency and the assumed values of second law efficiency from the literature without considering the nature of the working fluid and the components. So, a more detailed analysis can be done by conducting a regression-based sensitivity considering all the technical parameters for the PTES systems.

For a more detailed study, the modelling of the components of turbomachinery, heat exchangers and thermal energy storage can be done to analyze the real performance of the PTES systems. Identifying a suitable model for all the components and design of thermal storage considering the operational conditions for a given power range helps to significantly develop the performance mapping in real operational conditions.

A detailed exergoeconomic analysis should be performed to identify the system with the highest exergoeconomic performance and could be designed in detail.

Developing a realistic and up to date economic models for the PTES system allows to determine the economic potential of the PTES systems. It can be done by calculating all the costs associated with the components of the PTES systems and determine the levelized cost of storage (LCOS). Also, the exergy destruction for all the components can be determined for different configuration helps to design the PTES system with less exergy losses and thereby improving the overall thermoeconomic performance.

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Acronyms and Nomenclature

ESS: Energy Storage Systems

PHS: Pumped Hydro Storage

CAES: Compressed Air Energy Storage

LAES: Liquid Air Energy Storage

SMES: Superconducting Magnetic Energy Storage

PHES: Pumped Heat Electricity Storage

CHEST: Compresses Heat Energy Storage

PTES: Pumped Thermal Energy Storage

TI-PTES: Thermally Integrated Pumped Thermal Energy Storage

VCHP: Vapor Compression Heat Pump

ORC: Organic Rankine Cycle

STES: Sensible Thermal Energy Storage

LTES: Latent Thermal Energy Storage

TCES: Thermochemical Energy Storage

WTTES: Water Tank Thermal Energy Storage

UTES: Underground Thermal Energy Storage

PCM: Phase Change Materials

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