Manuscript Details

Manuscript number	ATE_2016_1648
Title	A novel Pumped Thermal Electricity Storage (PTES) system with thermal integration
Article type	Research Paper

Abstract

Power to heat technologies are becoming more and more important due to the extreme need of energy storage solutions to help manage the mismatch between supply and demand of electric power in grids with a large penetration of intermittent renewable energy systems. Several Electric Energy Storage (EES) technologies have been proposed in the literature, with different characteristics in terms of storage capacity, response time and roundtrip efficiency. In this paper the attention was focused on Pumped Thermal Electricity Storage (PTES), which is a technology that stores electric energy as heat by means of Heat Pumps (HP) and converts it again to power with a Heat Engine (HE). In this study, a hybrid PTES application was studied, which took advantage of a low-grade heat source to boost the electric round-trip efficiency of the system beyond 100%. The main idea was to exploit the heat source to reduce the HP operational temperature difference; this \textit{thermal integration} boosted the HP COP and thus the electric efficiency of the whole system. A Matlab numerical model was developed, using the thermodynamic properties of the Coolprop data base, and the steady state operation of a PTES system composed by a vapour-compression HP and an Organic Rankine Cycle (ORC) we simulated. Heat source temperature values ranging from 80\degree C to 110 °C and different working fluids were studied. Among the refrigerants, which comply with the latest European environmental legislation, the most promising fluid was R1233zd(E): with such fluid a maximum round trip-efficiency of 1.3 was achieved, when the heat source temperature reaches 110 °C and the machinery isentropic efficiencies is 0.8, the heat exchangers pinch points is 5 K and the ORC condensation temperature is 35 °C.

Keywords	Energy storage; Pumped Thermal Energy Storage; Power to Heat; Enhanced Heat Recovery

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To
Professor T.S. Zhao
Editor in chief of Applied Thermal Engineering

Pisa, December 14, 2016

Dear Professor Zhao,

On behalf of the coauthors, I am pleased to submit the manuscript entitled:

"A novel Pumped Thermal Electricity Storage (PTES) system with thermal integration"

by
G.F. Frate, M. Antonelli, U. Desideri

The paper is orignal and is neither under consideration for publication on any other Journal nor was it submitted to any conference.

There is no conflict of interest with any public or private institution.

We are looking forward to hearing from you.

Best regards

Prof. Umberto Desideri, PhD

A novel Pumped Thermal Electricity Storage (PTES) system with thermal integration

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Abstract

Power to heat technologies are becoming more and more important due to the extreme need of energy storage solutions to help manage the mismatch between supply and demand of electric power in grids with a large penetration of intermittent renewable energy systems. Several Electric Energy Storage (EES) technologies have been proposed in the literature, with different characteristics in terms of storage capacity, response time and roundtrip efficiency. In this paper the attention was focused on Pumped Thermal Electricity Storage (PTES), which is a technology that stores electric energy as heat by means of Heat Pumps (HP) and converts it again to power with a Heat Engine (HE). In this study, a hybrid PTES application was studied, which took advantage of a low-grade heat source to boost the electric round-trip efficiency of the system beyond 100%. The main idea was to exploit the heat source to reduce the HP operational temperature difference; this *thermal integration* boosted the HP COP and thus the electric efficiency of the whole system. A Matlab numerical model was developed, using the thermodynamic properties of the Coolprop data base, and the steady state operation of a PTES system composed by a vapour-compression HP and an Organic Rankine Cycle (ORC) we simulated. Heat source temperature values ranging from 80°C to 110°C and different working fluids were studied. Among the refrigerants, which comply with the latest

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European environmental legislation, the most promising fluid was R1233zd(E): with such fluid a maximum round trip-efficiency of 1.3 was achieved, when the heat source temperature reaches 110°C and the machinery isentropic efficiencies is 0.8, the heat exchangers pinch points is 5K and the ORC condensation temperature is 35°C.

Keywords: Energy storage, Pumped Thermal Energy Storage, Power to Heat, Enhanced Heat Recovery

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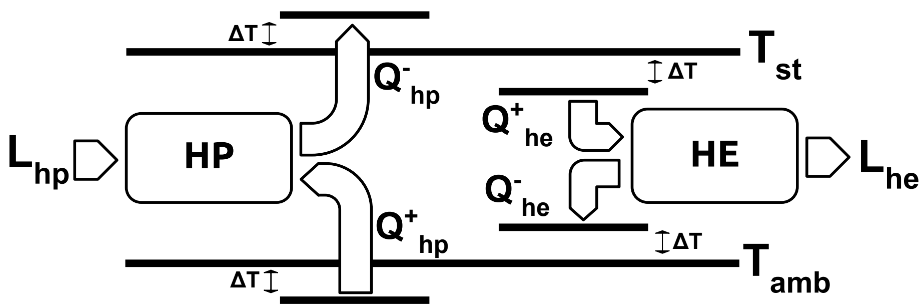

Figure 1: Working principles of classical PTES systems. The superscript + stands for heat gained by the component, while the superscript - stands for heat returned by the component.

1. Introduction

In many developed country an ever increasing share of electric energy is produced by Renewable Energy Sources (RES). The intrinsic aleatory nature of some of the RES poses many management and control issues. In fact the traditional mode of operation of large electric grids was aimed at matching a highly agglomerated demand with a small number of large power generation systems. In this arrangement, peaks in the demand were smoothed by large numbers of users, whose behavior was predictable by using long term statistics, and supply could be forecast in advance enough to guarantee a stable and secure service. The grids with large penetration of renewables are now facing new problems: the deployment of power generation systems has not been programmed by large utilities, but by residential and small industrial users, with significant unbalances between supply and demand capacity, thus creating areas with over capacity and areas with under capacity and with real difficulties in managing generation of small power plants with dispatch priority. The new system has therefore introduced a double mismatch between supply and demand: smaller and local aggregation of the demand makes it more difficult to predict and the installation of large shares of intermittent RES make it much more difficult to predict the supply, which is strongly influenced by weather changes and patterns.

The problems described above are likely to intensify, since the share of installed productive capacity based on RES is intended to grow, in accordance with the general trends that characterize the legislation on greenhouse gases emissions of many developed countries, among which the European Union [1].

It is now well known that a further exploitation of RES is possible only in conjunction with efficient and reliable Electric Energy Storage (EES) technologies [2, 3]. The main EES technology is Pumped Hydro Energy Storage (PHES), which features high efficiency and large storage capacity and has been used for several decades for the management of large electric grids. A PHES plant requires peculiar geographical conditions for its operation and, at least in Europe, the easily exploitable locations have been already utilized [4]. Since it is practically impossible to add PHES capacity to control grids with large shares of RES, in the last years the interest towards alternative EES technologies has grown: the main examples of such technologies are Compressed Air Energy Storage (CAES) and Battery Energy Storage (BES), but more technologies are available and being studied, such as Flywheels, Super-capacitors, Hydrogen, Superconducting Magnetic and many others Energy Storage systems. Further details can be found in [5, 6].

A rather poorly studied EES technology is the Pumped Thermal Electricity Storage (PTES), which has the peculiarity of storing the electric energy as heat. The PTES essentially converts electric energy into heat by means of Heath Pumps (HP), charging a Thermal Energy Storage (TES), and converts the heat back with a Heat Engines (HE).

Two main PTES systems have been studied so far: one which uses closed Brayton cycles and one which uses trans-critical CO_2 Rankine cycles. Some examples of Brayton PTES, using dynamic turbomachinery, are available in [7–10], while two variants with volumetric machines can be found in [11, 12].

Some examples of trans-critical Rankine PTES can be found in [13–16]. A different concept with isothermal compression and expansion can be found in [17].

In the literature a few PTES systems are also available, that are powered by both electrical and thermal energy: this technique, that we identify with the term *thermal integration*, allows to achieve a higher electric efficiency than the standard PTES [18, 19].

Focusing on thermally integrated PTES, in this study we outlined and analysed a novel system, by simulating its steady state operation for several operational parameters and conditions. The idea behind the proposed PTES system is to use a HP to exploit a suitable heat source allowing the PTES to store the heat at a higher temperature, without affecting the HP performance. At the same time, the HE efficiency increases, since the discharge phase has a higher maximum temperature, and then the whole process takes place with a higher round-trip efficiency.

The outlined idea is not completely new, since it is mentioned in general terms in [15, 20] as a potential way to enhance the performance of standard PTES systems. Even though the idea was previously proposed, it was never thoroughly analysed, to the best of the authors knowledge, and the present paper contributes to fill up this gap.

2. Methodology

2.1. Theoretical analysis

PTES systems store electric energy as heat by means of a HP, while the thermal energy is converted back to electricity by using a HE. Hence, most of PTES systems are composed by three main components: a HP, a TES an HE. Each subsystem is characterized by a coefficient of performance: for the HP we have:

$$COP = \frac{Q_{st}}{L_{hp}} \tag{1}$$

where L_{hp} is the electric energy absorbed by the HP, and Q_{st} is the heat provided by the HP to the TES tank.

For the HE we have:

$$\eta_{he} = \frac{L_{he}}{Q_{he}} \tag{2}$$

where L_{he} is the electric energy supplied back by the HE. In conclusion, for the TES we have:

$$\eta_{st} = \frac{Q_{he}}{Q_{st}} \tag{3}$$

where Q_{he} is the heat provided to the HE by the TES.

In standard PTES the heat is exchanged with two thermal reservoirs; the cold reservoir provides the heat to the HP and receives the heat from the HE, while the hot reservoir is the TES. For the sake of simplicity we assumed the thermal reservoirs as isothermal, so the hot reservoir is characterized by the temperature T_{st} and the cold reservoir is characterized by the temperature T_{amb}, as illustrated in Figure 1. The HP, the TES and the HE are arranged in series, hence the round trip efficiency, namely the ratio between the absorbed and returned amounts of electric energy, can be defined as:

$$\eta_{rt} = \frac{L_{he}}{L_{hp}} = \eta_{st}\eta_{he}COP \tag{4}$$

If we assume to use ideal machines, we may write:

$$\begin{cases} COP_{id} = \frac{T_{st}}{T_{st}-T_{amb}} \\ \\ \eta_{id} = \frac{T_{st}-T_{amb}}{T_{st}} \end{cases}$$

and the round-trip efficiency becomes:

$$\eta_{rt}^{id} = \eta_{st}$$

η_{st} ranges from 0 to 1, so even in the ideal case the round-trip efficiency cannot be higher than 1.

It is, however, possible to conceive a PTES system which takes advantage of a suitable heat source to provide heat at temperature $T_s > T_{amb}$, as illustrated in Figure 2. Introducing a third thermal reservoir does not influence the definition

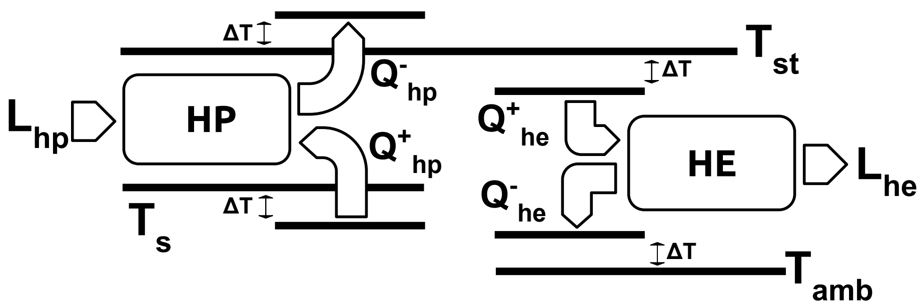

Figure 2: Working principles of the proposed thermally integrated PTES system. The superscript + stands for heat gained by the component, while the subscript - stands for heat returned by the component.

of η_{rt}, since it takes in account only the electric amount of energy, but now we have:

$$\begin{cases} COP_{id} = \frac{T_{st}}{T_{st}-T_s} \\ \\ \eta_{id} = \frac{T_{st}-T_{amb}}{T_{st}} \end{cases}$$

and for η_{rt} we find:

$$\eta_{rt}^{id} = \eta_{st}\left(\frac{T_{st}-T_{amb}}{T_{st}-T_s}\right) \tag{5}$$

In such case the round-trip efficiency can be higher than 1 as long as:

$$T_s > T_{st}(1-\eta_{st}) + \eta_{st}T_{amb}$$

If we have a perfectly insulated TES ($\eta_{st} = 1$), η_{rt} is always higher than 1. This effect takes place because the thermal source allows to the HP to store the same amount of heat absorbing less electric energy; in practical terms the system is powered by both electric and thermal energy inputs; to take in account both those terms we defined a total efficiency as:

$$\eta_{tot} = \frac{L_{he}-L_{hp}}{Q_s} = \frac{\eta_{st}\eta_{he}COP-1}{COP-1}$$

where Q_s is the heat provided by the thermal source. In the ideal case we have:

$$\eta_{tot}^{id} = 1 - \frac{T_{st}(1-\eta_{st}) + T_{amb}\eta_{st}}{T_s} \tag{6}$$

From Equations 5 and 6 we see that both the round-trip and the overall efficiency increase while T_s increase; conversely, if T_s is fixed, both the efficiencies increase while T_{st} tends to T_s, which is the same as saying that the HP has to work with minimum operational temperature difference to maximize the performance of the system.

In practical cases it is a non-sense to use a heat source from the combustion of fossil fuels, while it could be very interesting to use waste or low grade heat sources. In this perspective the main available heat sources are Industrial Waste Heat (IWH), low grade geothermal heat and solar thermal energy. For the sake of simplicity, we assumed the thermal source as a thermal reservoir at constant temperature, even though all the listed potential thermal sources provide sensible heat.

Based on the above assumptions and constraints the PTES system that we propose in this paper consists of a vapour-compression HP for the charging phase of the TES and of an Organic Rankine Cycle (ORC) recovering the stored heat. Figure 3 shows the T-s plane with the thermodynamic cycles of the HP and the ORC for one of the studied cases. Bearing in mind Equations 1, 2 and 3 and

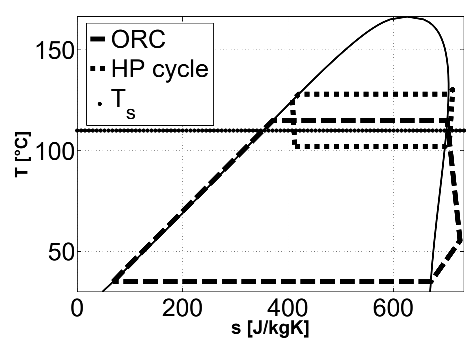

Figure 3: in T-s diagram of the Heat pump cycle and organic Rankine cycle with R1233zd(E); the represented case works with $\Delta T_{op}^{hp} = 10$K and $T_s = 110°$C

performing energy balances on the HP, the ORC and the TES, the main energy flows among the components were calculated and the results are illustrated in Figure 4.

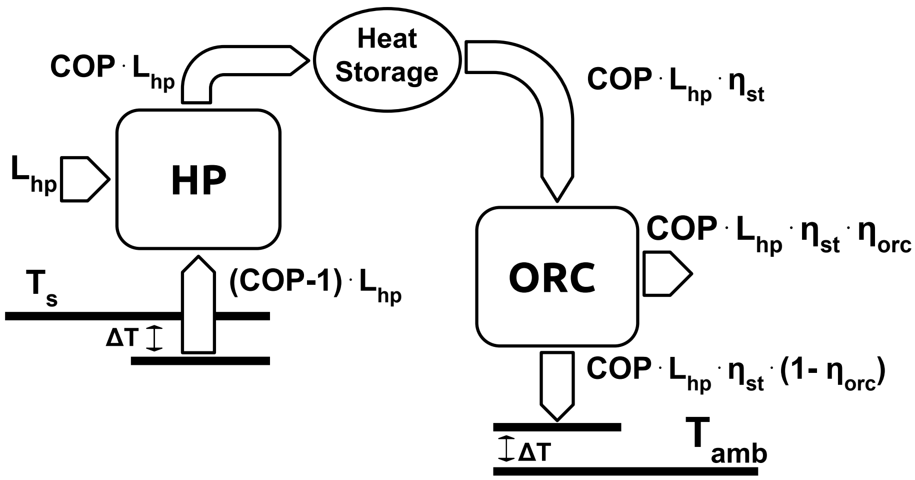

Figure 4: Conceptual diagram of the proposed thermally integrated PTES system. The specified energy flows are calculated with Equations 1, 2 and 3 and the energy balance of each component.

2.2. Numerical model

The paper deals with the steady state simulation of the outlined storage system, focusing on the influence of T_s and T_{st} on the round-trip efficiency. A total of 17 fluids, chosen among the artificial and natural refrigerants contained in the Coolprop data base [21], were investigated; from this initial pool of fluids we discarded all that underwent, or will be subjected in the near future, to bans or restrictions due to European environmental legislation [22]. The remaining *environmentally friendly*, or *clean*, fluids were further analyzed. The initial 17 refrigerants were selected considering their critical temperature T_{crit}: only the subcritical operation for the HP and the ORC was assumed, thus many refrigerants were discarded because their T_{crit} was too low. We performed all the simulations with MATLAB (ver. 2012b) and all the thermodynamic data were retrieved by the Coolprop data base. The developed numerical model is based on some assumptions:

- the charge and the discharge phases are performed with the HP and the ORC using the same working fluid. Water is not among the tested fluids because it would have required a vacuum pump to extract the non-condensable gasses from the condenser, while none of the other fluids presented such requirement;
- $\eta_{is} = 0.8$ for both ORC expander and HP compressor;
- $\eta_{st} = 0.9$;
- the evaporator and the condenser of the HP and the evaporator of the ORC have the same pinch point $\Delta T^{hp}_{evap} = \Delta T^{hp}_{cond} = \Delta T^{orc}_{evap} = 5\text{K}$;
- at the exit of HP condenser the fluid is subcooled by three degree ($\Delta T^{hp}_{sc} = 3\text{K}$);
- at the exit of HP evaporator the fluid is superheated by three degree ($\Delta T^{hp}_{sh} = 3\text{K}$);
- the HP compression ends in the superheated or saturated steam state;
- the ORC expansion has a final vapor quality higher than 0.85;
- if the ORC expansion ends in the superheated steam state at a temperature 15K higher than the condensation temperature, the cycle can be regenerated. In other words, if $T^{orc}_{sh,cond}$ was the temperature at the exit of the expander and T^{orc}_{cond} was the ORC condensation temperature, the cycle can be regenerated only when $\Delta T^{orc}_{reg} = T^{orc}_{sh,cond} - T^{orc}_{cond} \geq 15\text{K}$. Unlike other ORC applications, the regeneration is beneficial, since the heat provided to the ORC is stored before being used, thus a more efficient ORC leads to a more compact TES;
- the ORC condensation temperature is $T^{orc}_{cond} = 35°\text{C}$;
- the pressure losses in the heat exchangers are negligible;
- the minimum HP operational temperature difference $\Delta T^{hp}_{op} = T_{st} - T_s = 10\text{K}$. This was assumed as the minimum achievable value in practice.

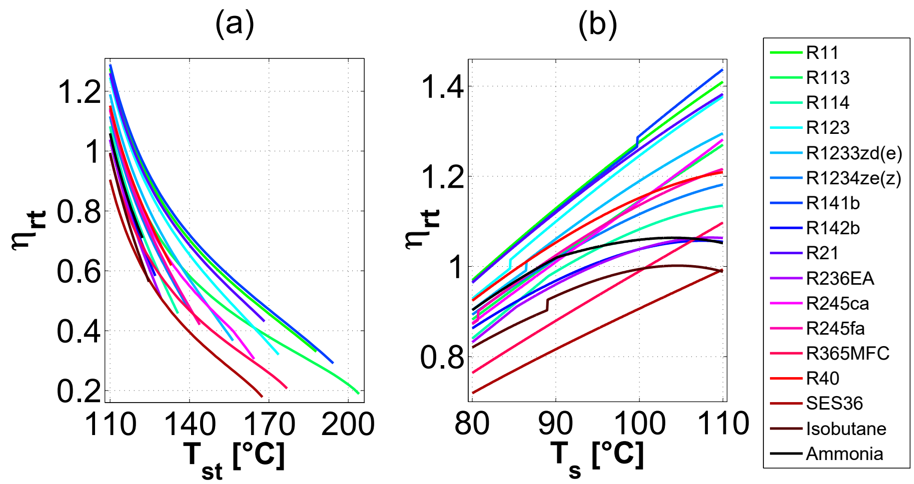

Figure 5: Round-trip efficiency η_{rt} as function of T_{st} and T_s for all the investigated fluids. (**a**): fixed $T_s = 100°C$ and varying T_{st}. (**b**): varying T_s with $\Delta T_{op}^{hp} = 10K$ and $T_{st} = T_s + \Delta T_{op}^{hp}°C$. Discontinuities and sudden changes of slope were due to the ORC regenerator, which started working only when the condensation steam was sufficiently superheated.

3. Main results

The results obtained from the simulations are in agreement with the theoretical conclusions deduced in Section 2.1, confirming that also in the practical case both η_{rt} and η_{tot} are maximized by small ΔT_{op}^{hp}.

Figures 5 shows η_{rt} as a function of T_{st} and T_s for all the investigated fluids; Figure 5(**a**) shows the case with $T_s = 100°C$ and variable T_{st}; while Figure 5(**b**) shows the case with variable T_s and $T_{st} = T_s + 10$.

The results related to only the environmentally friendly refrigerants were isolated for the sake of clarity and illustrated in Figures 6(**a**) and 6(**b**). Figure 7(**a**) shows η_{tot} as a function of T_s. The negative values correspond to the case in which $\eta_{rt} < 1$ and thus the net work becomes negative.

For a better characterization of the performance of the proposed system, we compared the overall efficiency with that of a system which directly exploits the heat source; for a fair comparison we assumed to use an ORC with the same characteristics of that used in the PTES; this ORC exploited the heat source

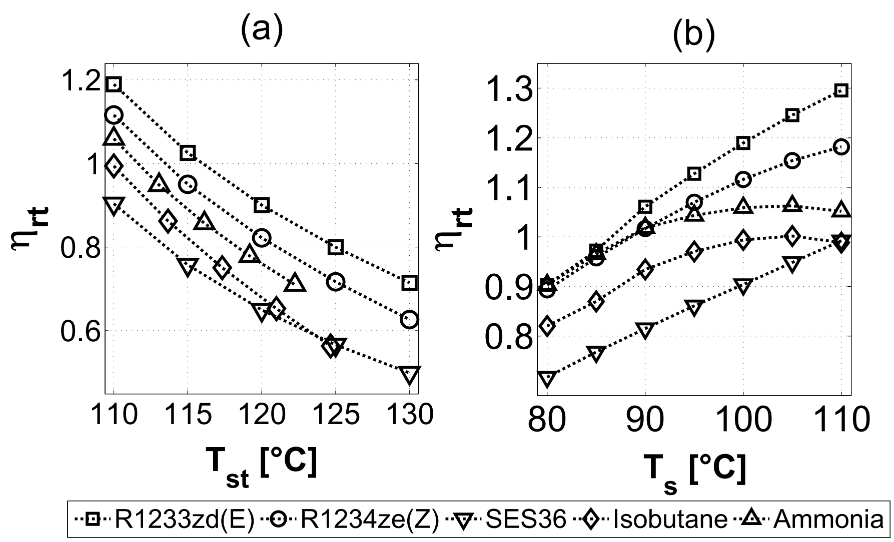

Figure 6: Round-trip efficiency η_{rt} as function of T_{st} and T_s for only the environmentally friendly fluids. **(a)**: fixed $T_s = 100°$C and varying T_{st}. **(b)**: varying T_s with $\Delta T_{op}^{hp} = 10$K and $T_{st} = T_s + \Delta T_{op}^{hp}$°C. Discontinuities and sudden changes of slope are due to the ORC regenerator, which starts working only when the condensation steam is sufficiently superheated.

directly so its evaporation temperature was $T_{evap}^{orc} = T_s - \Delta T_{evap}^{orc}$. Conversely, the PTES ORC had $T_{evap}^{orc} = T_{st} - \Delta T_{evap}^{orc}$.

We compared η_{tot} with the direct exploitation efficiency η_{dir} by defining:

$$\gamma = \frac{\eta_{tot}}{\eta_{dir}}$$

Figure 7**(b)** shows γ as a function of T_s.

4. Discussion and ancillary results

4.1. Round trip efficiency and total efficiency

By varying T_s in the range from 80°C to 110°C, the highest values of η_{tot} and η_{rt} was found when $T_s = 110$°C.

Although working fluids such as R11 and R141b allowed to achieve $\eta_{rt} > 1.4$, the maximum efficiency with an allowed fluid did not exceed $\eta_{rt} = 1.3$. The most promising environmentally friendly fluid is R1233zd(E), followed by R1234ze(Z) and Ammonia. The main results for those three fluids are summarized in table 1.

The results about efficiency confirmed that in principle η_{rt} and η_{tot} increase

Fluids	**R1233zd(E)**			**R1234ze(Z)**			**Ammonia**		
T [°C]	**80**	**95**	**110**	**80**	**95**	**110**	**80**	**95**	**110**
η_{rt}	0.903	1.128	1.295	0.893	1.07	1.182	0.903	1.043	1.052
η_{tot}	-0.0104	0.014	0.032	-0.012	0.008	0.021	-0.011	0.0051	0.007
γ	-0.126	0.130	0.249	-0.141	0.073	0.167	-0.128	0.046	0.053

Table 1: Summary of main results for the three most performing operative fluids. Negative values of η_{tot} and γ correspond to the case in which $\eta_{rt} < 1$ and thus the net work $L_{net} = L_{orc} - L_{hp} < 0$.

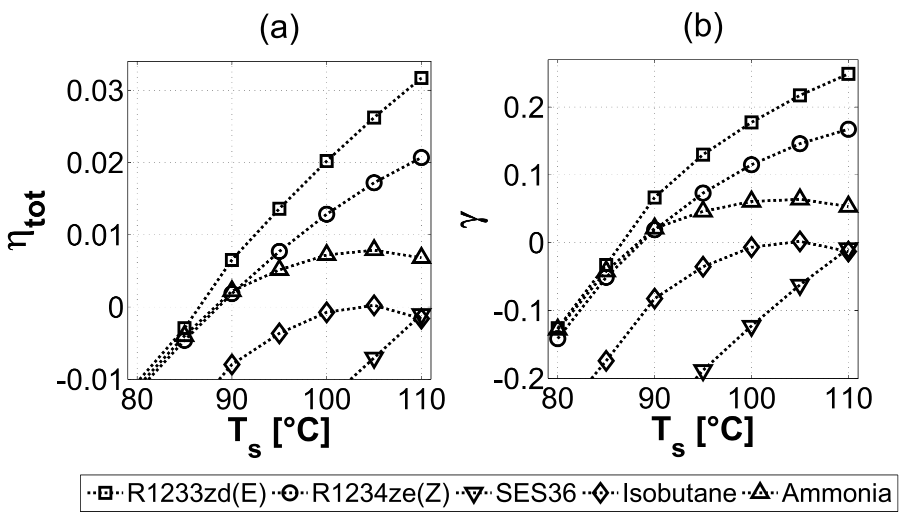

Figure 7: Total efficiency η_{tot} (figure **(a)**) and γ, the ratio between η_{tot} and the direct exploitation efficiency, (figure **(b)**) as function of T_s for the environmentally friendly fluids. Discontinuities and sudden changes of slope were due to the ORC regenerator, which starts working only when the condensation steam is sufficiently superheated

with higher values of T_s in agreement with what has been discussed in Section 2.1. However, we found that η_{rt} and η_{tot} does not increase monotonously when

T_s approaches T_{crit}: in fact, they both showed a maximum before starting to drop while T_s increased. This effect is clear in Figure 5 and 7 only for Isobutane and Ammonia, which had the lowest T_{crit} among the investigated fluids. The critical temperatures of the other fluids were well beyond the investigated temperature range, so the peaks of efficiency were not visible.

This effect could not be predicted by the theoretical analysis in Section 2.1, since the Carnot efficiency does not depend on the nature of the working fluid. η_{rt} and η_{tot} showed similar trends and the R1233zd(E) was again the fluid with which the highest overall efficiency was achieved: the maximum η_{tot} was achieved in correspondence of $T_s = 110°\text{C}$ and its value was slightly higher than 3%.

γ has similar trends with η_{tot} and η_{rt}, its maximum value being slightly lower than 0.25, which means that the PTES converts the provided energy (both the electric and the thermal amounts) with an efficiency four times lower than an hypothetical ORC which exploits the heat source directly.

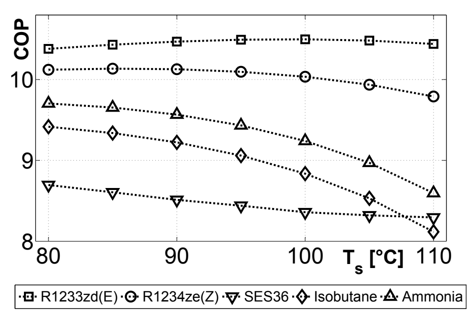

Figure 8: COP as a function of T_s for the environmentally friendly fluids.

4.2. Technical considerations

4.2.1. Operational pressures

The HP condensation and evaporation pressures are shown as a function of T_s in Figures 9**(a)** and 9**(b)**, respectively. Due to the operational temperatures, it is important to work with pressures as low as possible, in order to reduce the mechanical stress on the piping and thermal equipment.

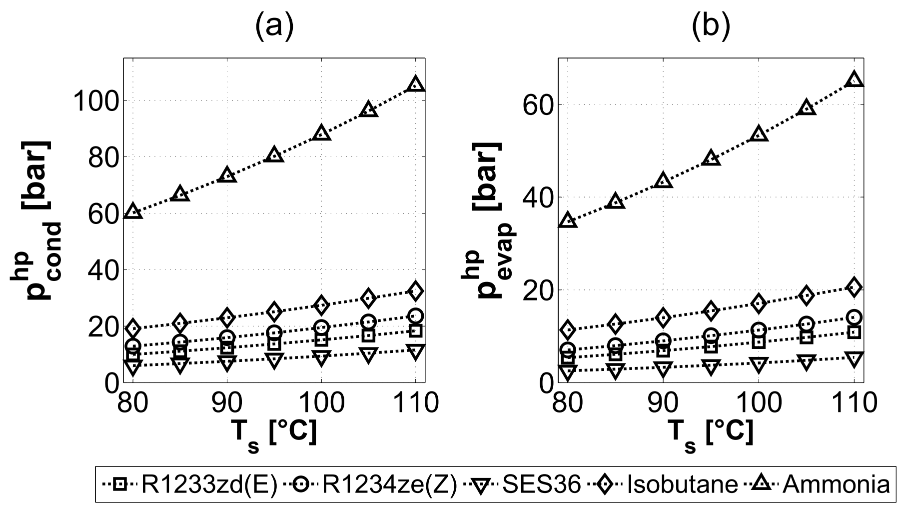

Figure 9: HP condensation and evaporation pressures, in **(a)** and in **(b)** respectively, as a function of T_s for the environmentally friendly fluids.

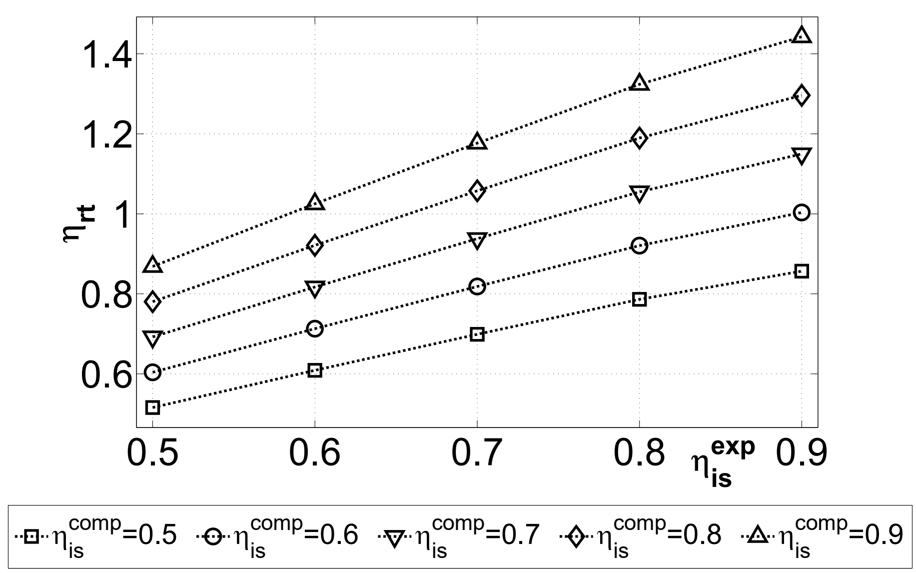

Figure 10: η_{rt} as a function of η_{is}^{exp} for different values of η_{is}^{comp} with R1233zd(E) as operative fluid, $\Delta T_{op} = 10$K and $T_s = 100°$C

The Ammonia showed the highest pressures values, while the R1233zd(E) and the SES36 showed the lowest. As a rule of thumb the HP operational fluid pressures should be as low as possible in order to reduce the thermal equipment costs. In our study, the fluid which showed the highest efficiency, the R1233zd(E), had also the second lowest operational pressure values, which confirms such fluid as the most suitable for the proposed PTES system.
It is worth noting that the R1233zd(E) pressures were in line with that of the traditional heat pumps; in Table 2 we reported the desirable values of operational pressures for three commercial HP refrigerants that are comparable with that obtained with R1233zd(E) in our analysis. Therefore, the development of a suitable thermal equipment for the high temperature HP, which is necessary for the outlined PTES system, should not arise serious technical issues.

Fluids	**R134a**	**R410a**	**R407c**	**R1233zd(E)**
P^{hp}_{cond} [bar]	11.6	27.3	19.7*	18.4**
P^{hp}_{evap} [bar]	3.5	9.3	6.7*	10.9**

Table 2: Saturation pressures for three common HP refrigerants at the temperatures 5°C and 45°C. (*)R407c is a blend and only the liquid saturation pressures are reported. (**) The pressures of R1233zd(E) are calculated with $T_s = 110°C$.

4.2.2. Thermal and electric storage capacity

The storage capacity is the amount of electric energy that can be stored in the system. In our analysis such parameter was represented by L_{hp}, which is exactly the electric energy absorbed during the storage charging phase.
The energy balance on the HP yields:

$$\begin{cases} Q_s = L_{hp}(COP - 1) \\ Q_{st} = L_{hp}COP \end{cases} \tag{7}$$

In the investigated temperature range the COP of R1233zd(E) was quite constant and it showed values comparable to 10.5, as can be seen in Figure 8, hence, from the second of the Equations 7 it follows that, for every kWh of stored electric energy, we have to accumulate more than ten kWh of heat. This thermodynamic requirement establishes a practical and economical limitation to the size of the storage system, encouraging the use of heat storage media that can guarantee high energy density, such as Phase Change Materials (PCM). An accurate analysis about the optimal PCM for the proposed PTES system is beyond the purpose of this paper.

From the second of Equation 7 it also follows that higher COPs lead to higher efficiency, but also to a larger heat storage tank volume, which generally entails higher costs and larger thermal losses. These contrasting effects must be taken into account during the storage sizing, thus the choice of the magnitude of L_{hp} needs to be the outcome of a thermo-economic optimization process. Such an in-depth analysis requires to specify at least the nature of the thermal source and the TES materials, hence it is beyond the scopes of the present paper.

From the first of Equations 7 it follows that also Q_s is almost ten times greater than L_{hp}. Therefore, a system with high η_{rt}, i.e. with high COP, is limited in size not only by the volume of the TES, but also by the maximum amount of heat provided by the source. Moreover, Equations 7 are written in terms of energy but are also valid in terms of power, thus the heat source has to supply not only the adequate amount of energy, but also the adequate amount of power.

If the heat is provided by IWH or geothermal resources, the size of the thermal source is fixed both in terms of energy and power, setting a practical limitation to the size of the PTES electric capacity. If there are no pre-existing thermal sources, Q_s has to be produced and thus it will have a production cost, for example the costs of purchase and installation of the solar collectors, in case of solar energy exploitation. Hence, from the economical point of view the optimum size of L_{hp} can be obtained only with a thermo-economic optimization that takes into account the efficiency, the thermal energy production cost and the limitations imposed by the Equations 7.

4.2.3. Comparison with other PTES systems

Round trip efficiency

As previously mentioned, the main kinds of PTES are those which use transcritical CO_2 cycles or closed Brayton cycles. Both those systems have a round trip efficiency that is hardly higher than 0.6, which is less than half of the efficiency achieved by our system. However this comparison is rather unfair, because those systems are designed to work with only electric inputs, while our system owes its higher round trip efficiency to the thermal integration.

The proper comparison has to be done between systems that take advantage of both thermal and electric energy inputs. To the best of the author knowledge the examples of such systems available in the literature are:

- [18] a PTES system integrated with IWH that achieves an η_{rt} slightly higher than 0.8 with a heat input provided at a maximum temperature of 80°C;
- [19] a PTES system integrated with solar thermal energy, where the heat is provided at a maximum temperature slightly lower than 100°C and the system achieves a maximum $\eta_{rt} = 0.84$;
- [15] a transcritical CO_2 PTES system. Such system utilizes only electric input and the thermal integration is mentioned only as way of enhancement of η_{rt}. No further details are given in the paper;
- [20] a system designed to work with only electric input whose performances can be boosted with an additional thermal input. the round-trip efficiency exceeds 1 when the heat is provided at $T_s > 88$°C.

It is interesting to compare the efficiency of thermally integrated PTES, being equal the temperature of the heat source: in fact, this guarantees that the compared systems are taking advantage of comparable heat sources. It is true that a complete comparison requires to compare the exergetic efficiencies of the examined systems, but being such data unavailable in the literature, we can settle for the specified approximate comparison.

Since our system achieved $\eta_{rt} > 0.9$ for $T_s \geq 80$ and $\eta_{rt} > 1$ for $T_s \geq 86.5$, we can conclude that, under the simplifying assumption of an isothermal heat source, the efficiency of our system is higher than any other described in the literature. It is worth noting that in the literature examples, the process that provides the heat is sometimes not isothermal, thus the comparison would require further analysis.

Plant design complexity

Apart from featuring high efficiency, a PTES system should have a simple plant design. The efficiency can be a good indicator of the quality of the system, only as long as its complexity does not impair its own technical and economical feasibility. On the other side, although the design simplicity is one of the most important features, it is often necessary to complicate the system in order to achieve satisfactory efficiency. From this point of view, the thermal integration can bring great advantages, boosting the η_{rt} to such an extent that many add-ons intended to enhance it might become unnecessary, promoting a much simpler plant design.

Plant complexity often has the effect of pushing the system towards high ranges of capacity (from ten to hundreds of MWh), in order to justify the technical and economical efforts. Some examples of high capacity PTES with complex design can be found in [7, 20, 23].

The fundamental problem of a high capacity PTES is that it has to withstand the comparison against similar capacity PHES systems, which usually have higher efficiency and are already a well consolidated technology. However, the PTES have the great advantage of offering a comparable storage service, without requiring any particular geographical constraint. In this perspective the thermal integration can be a double-edged sword since it actually boosts the η_{rt}, but it links the PTES to the thermal source geographical position, forcing it to lose its main advantage over the PHES.

Among the thermal sources that can be exploited, only the solar one establishes no geographical constraints. Despite this, solar energy has to be produced rather than recuperated from low grade resources often considered unprofitable, furthermore, the required solar field has to be of adequate size, as stated in Section 4.2.2, and the economical feasibility of such solution requires careful evaluation. For all the highlighted reasons, the thermal integration seems to be more suitable for systems with little or medium capacity ($L_{hp} < 5 \div 10$ MWh), in which range the systems have to be as cheap and simple as possible, thus they can benefit a lot from the efficiency boost due to thermal integration. Moreover, small size PTES do not have to compete against PHES, so they can endure some geographical constraints dictated by the availability of the heat sources. Finally, for such medium or small systems, the solar integration could become more affordable, especially in conjunction with microgeneration scenarios, in which the same array of solar collectors could provide heat for multiple applications.

4.3. Sensitivity analysis

A sensitive analysis was performed on the round-trip efficiency results, in agreement with the other studies on PTES systems available in the literature. The investigation was focused on the influence of isentropic efficiencies, heat exchangers pinch points and condensation temperature of ORC, by studying a total of six variables, as indicated in Table 3.

Since it is not practical to analyse the separate effects of each variable, we followed a *Monte Carlo* approach: the system was iteratively simulated, randomly drawing the analysed variables from a range of acceptable values; within this range the probability of being picked up is uniform and each range is centered around a mean value, corresponding to that used in the main analysis for the related variable. The selected variation ranges and the related mean values are listed in Table 3.

Every simulation produces a value of η_{rt} that is function of T_s, as pointed out in the main analysis, and of η_{is}^{comp}, η_{is}^{exp}, ΔT_{evap}^{hp}, ΔT_{cond}^{hp}, ΔT_{evap}^{orc} and T_{cond}^{orc}. We repeated the whole process a suitable number of time and we obtained a set of $\eta_{rt} values$ on which a multi-variable linear regression was performed, in order to find the most appropriate linear relation between the efficiency, T_s and the aforementioned six variables.

Variables	η_{is}^{comp}	η_{is}^{exp}	ΔT_{evap}^{hp}	ΔT_{cond}^{hp}	ΔT_{evap}^{orc}	T_{cond}^{orc}
mean value	0.8	0.8	5	5	5	15
Variation range	±0.2	±0.2	±3	±3	±3	±10

Table 3: Mean values and variation ranges of the variables selected for the sensitivity analysis.

Coefficient	B	A_1	A_2	A_3	A_4	A_5	A_6	A_7
Reference Variable	-	T_s	ΔT_{evap}^{hp}	ΔT_{cond}^{hp}	η_{is}^{comp}	T_{cond}^{orc}	ΔT_{evap}^{orc}	η_{is}^{exp}
Numerical Value	-1.0634	0.0114	-0.0464	-0.0466	1.1796	-0.0146	-0.0111	1.4332

Table 4: Numerical values of linear regression coefficients

In practical terms, we assumed that η_{rt} can be written as:

$$\eta_{rt} = \sum_{i=1}^{n} A_i x_i + B$$

where B is a constant and each of the x_i is a variable among T_s and the six listed in Table 3.

The residual ϵ_j was the error that affects the linear model with respect to the j-th random evaluation of η_{rt} and it can be defined as:

$$\epsilon_j = \eta_{rt}^j - \sum_{i=1}^{n} A_i x_i^j + B$$

where x_i^j and η_{rt}^j are respectively the input variables and the result of the j-th random evaluation of the round trip efficiency.

The best linear relation minimizes the sum of $(\epsilon_j)^2$ and as a measure of the quality of the fit the parameter R^2 is used, which in our particular case is defined as:

$$R^2 = 1 - \frac{\sum_j \epsilon_j}{\sum_j (\eta_{rt}^j - \langle \eta_{rt}^j \rangle)}$$

where $\langle \eta_{rt}^j \rangle$ is the mean values of η_{rt}^j. R^2 ranges from 0 to 1 and the closer to 1 it is, the better the model traces the fitted data.

The linear model parameters may change with number of randomly generated points, thus in order to achieve a satisfactory independence from the number of random evaluations of η_{rt}, we generated an adequate number of points. Then, we monitored each parameter of the linear model while increasing the number of generated points and we stopped when the relative variation from an iteration to the following was smaller than 5%. We found that a number of 10^4 random evaluations was appropriate to satisfy the established precision criterion. In Table 4 we indicated the coefficients of the linear model; such model was characterized by $R^2 = 0.9684$.

With the linear model we have $A_i = \frac{\partial \eta_{rt}}{\partial x_i}$, that can be seen as a measure of the influence of x_i on η_{rt}. From Table 4 we can see that the isentropic efficiencies had the greatest impact, followed by ΔT^{hp}_{evap} and ΔT^{hp}_{cond}, while ΔT^{orc}_{evap} was the least influential variable.

The sensitivity analysis suggests that the ORC expander, followed closely by the HP compressor, is the piece of equipment that has to be selected with the greatest care. On the contrary, the ORC evaporator could have had a relatively high pinch-point, without severely affecting the round trip efficiency.

In order to isolate the effects of the isentropic efficiencies, which were the two most influential variables, we repeated the sensitivity analysis, fixing all the other variables to their mean values and assuming $T_s = 100°\text{C}$; Figure 10 summarizes the results of this last analysis, illustrating how crucial can be to work with a high quality equipment for expansion and compression.

5. Conclusions

PTES systems generally store electrical energy in the form of heat by means of a HP and convert it back with a HE. In the previously studied configurations, the HP takes the heat at the same temperature at which the HE gives it back. Conversely, in the present paper we propose a system that takes advantage of a suitable heat source, in order to enable the HP to absorb the heat at temperatures higher than that at which it is discharged. We referred to such technique as *thermal integration* and we found that, by means of reducing the operational ΔT of the HP, it enhances the round-trip efficiency of the storage system.

PTES is an emerging electric storage technology, but it is difficult to achieve round-trip efficiencies higher than 0.6. In this perspective, we found that the thermal integration can be a great way to boost the PTES performance. The number of papers that analyse such application is scarce compared with that of papers about batteries, CAES or other innovative electric storage technologies, but our results confirmed the considerable potential of thermally integrated PTES systems as electric storage technologies.

A number of heat source temperatures in the range from 80°C to 110°C was studied and the performance of the system for several working fluids was simulated. By focusing on the fluids that comply with the latest European environmental legislation, we found that the R1233zd(E) is the most promising, since such fluid showed a maximum round-trip efficiency equal to 1.3 when the heat source temperature was 110°C. Such value is not surprising, since the round-trip is usually defined taking in account only electric energy terms and it can be higher than 1 if the heat source is at sufficiently high temperature.

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Highlights:

- A novel thermally integrated Pumped Thermal Electricity Storage was proposed
- A numerical model was developed and the steady state operation of PTES was simulated
- The thermal integration boosted the electric round-trip efficiency beyond 100%
- A comparison between standard and thermally integrated PTES was proposed
- Practical limitations to capacity size due to required amount of heat were discussed