Study of a liquid air energy storage system

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With the emergence of renewable energies, the grid becomes more and more complicated to manage. The intermittency of these new means of production increases the need for energy storage solutions.

Against this background the idea of an innovative energy storage system arises which uses electrical energy to liquefied air, then store the liquid phase in a tank in order to compensate the electricity surplus of the grid, and expand it through a turbine when electricity is needed on the network.

The principle of function is given in the following diagram, Fig. 1.

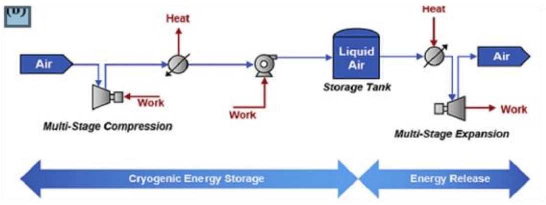

Figure 1: Operating principle of an energy storage system based on air liquefaction.

In the liquefaction part, we need electrical energy to run the compressors and the pumps. During this process, thermal energy could be extracted from the compression (losses), and from the cooling of air to its boiling temperature. For the expansion part, heat is needed to evaporate the liquid air and to improve the expansion in the turbine.

For this study, only the liquefaction part will be investigated. The feasibility and efficiency consideration of the evaporation system will be examined in a future work.

The liquefaction of air goes back to Carl von Linde and W. Hampson who both registered a patent for gas separation in 1895. It is the easiest cycle and contains one or two heat exchangers for cooling the compressed air by the recirculating cold gaseous phase below the inversion temperature.

Later, many other circuits were developed to improve energy efficiency through additional cooling and compression stages upstream of the expansion valve [1-9]. The Claude Cycle for instance uses three heat exchangers in a row. After the first heat exchanger, the compressed air flow is devided in two flows. One air stream passes through the liquefaction line, the second stream is expanded in a turbine and after mixing the turbine exhaust air with the recirculating cold air of the liquefaction line it is used as cooling media in heat exchanger two. Some of the compression energy can thus be recovered but additional plant components are necessary. Another variant of the Claude Cycle is the Kapitza Cycle, whereby the third heat exchanger is removed. Although both cycles are more efficient than the Linde-Hampson Cycle this study deals with the latter for simplicity. In order to find out if the air liquefaction process can be coupled to power generation processes, for instance ORC-Cycles, the influence of parameter variations on a Linde-Hampson process is studied.

In Fig. 2 the Linde-Hampson Cycle is illustrated and the process steps are numbered.

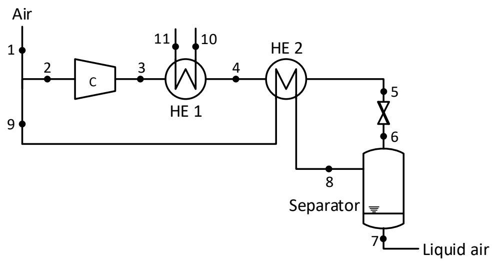

Figure 2: Linde-Hampson cycle

At ambient pressure, fresh air (1) is mixed with the recirculating cold air (9) to the inlet air (2) of the compressor (C). The mass flux at position (2) is m. After the compressor, the air pressure is enhanced to 200 bar (3). The compressed air is then cooled in an externally chilled heat exchanger to state (4). A second heat exchanger, operated by internal heat exchange, reduces air temperature close to air condensation temperature (5). After isenthalpic expansion in the throttle valve air pressure is reduced to ambient pressure (1.015 bar) thus causes the temperature to be lowered to the condensation temperature of -194.3°C (6). In the separator, gaseous (8) and liquid phase (7) are separated and the gaseous air (8) is further used in heat exchanger 2.

The air states can be calculated as follows based on mass flux m at point 2, whereby x is the mass fraction of gaseous air:

Ambient condition:	T_1 = 293.15K, p_1 = 1.015 bar, fixed values
Ideal mixing of fresh and recirculated air	$T_2 = xT_9 + (1-x)T_1$, $p_2 = p_1$, whereas x corresponds to mass fraction of recirculating air
Isothermal compression:	$T_3 = T_1$, p_3 = 200 bar, fixed values
Cooling in HE 1:	$h_4 = (1-\eta_{cool}) \cdot h_3$ with efficient coefficient η_{cool} T_4 can be calculated from h_4 by using REFPROP database for instance, $p_4 = p_3$
Isenthalpic expansion at valve:	$h_5 = h_6$, $p_5 = p_3$
Boiling point at ambient pressure:	T_6 = -194°C, p_6 = 1.015 bar
Gaseous phase at boiling point	$T_7 = T_6$, $p_7 = p_6$, $m_7 = (1-x) \cdot m$
Liquid phase:	$T_8 = T_6$, $p_8 = p_6$, $m_8 = m \cdot x$
Adiabatic cooling in HE 2	$T_9 = T_8 + (T_4 - T_5) / x$, $p_9 = p_6$

What can be seen from the formulas above, x and η_{cool} cannot be varied independently. Some physical limitaions have to be considered. In both heat exchangers, the cold fluid cannot be hotter than the hot one. That leads to following restrictions in HE 2: $T_4 > T_9$ and $T_5 > T_8$. This could not be respected when x or η_{cool} are too low. In Fig. 3 temperatures T_4 and T_9 are shown as function of η_{cool} for given x=0.5.

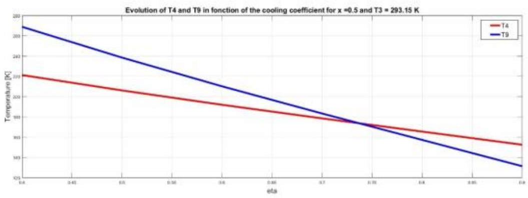

Figure 3: Temperatures T_4 and T_9 depending on η_{cool} for given x = 0.5

As shown in Fig. 3 for x = 0.5, η_{cool} has to be higher than 0.74 which means that the efficiency of HE 1 is restricted in order to provide a minimum cooling rate. Depending on the amount of liquid air (1-x), the first stage cooling in HE 1 is predefined. In addition following criteria for HE 1 has to be fulfilled as well: $T_4 > T_{10}$ and $T_3 > T_{11}$. Therefore, a specific external cooling process must be adapted to the specific temperature level.

In order to understand the behaviour of the system, the influence of the liquid rate (1-x) and the cooling coefficient η_{cool} on the temperatures and the enthalpies of each point are analysed. The interesting values are:

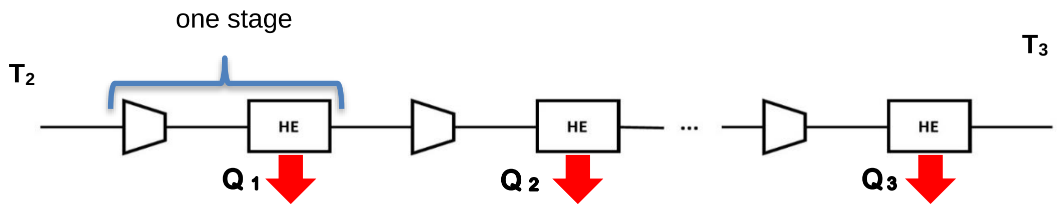

Figure 4: Multi-stage compression with heat extraction

- Temperature at point 4 (T_4)
- Temperature at point 9 (T_9)
- Enthalpy extracted between 3 and 4 (Δh_{34}) in HE 1
- Maximal temperature of the compressor, like if there were not any cooling (T_{comp})
- Enthalpy that could be extracted by the compressor cooling system

The respective parameters are varied in order to examine how much heat can be extracted by combining air liquefaction with external processes. Possible combinations are identified. First, compression heat is used for warming up the working fluid of an external ORC-process of a geothermal energy plant. For this purpose the air compression must be operated in several stages in order not to overheat the compressor. Second, the influence of liquid air ratio (1-x) on the efficiency of HE 1 (η_{cool}) and third, the extraction of gaseous air prior to mixing are investigated.

Influence of parameters

In order to study the influence of parameter variation like x or η_{cool} the isothermal compression is supplemented by a multi-stage adiabatic compression for both enhancing the total energy extraction from the circuit and avoiding overheating of the compressor during adiabatic operation, as shown in Fig. 4. This method allows to increase the released energy yield and to use it technically in another process.

For each stage, we consider an adiabatic compression and then an isobaric heat exchange. The heat exchange can be effected by another system (ORC p.e.). The compression ratio is the same for each compressor:

$$\frac{T_{i+1}}{T_i} = constant\ at\ each\ stage. \qquad (1)$$

It is important to notice that the temperature after each heat exchanger (including T_3) depends on the system used to cool down the air, because its minimal value is the temperature of the inlet of the cooling fluid. After the last stage, it is assumed, that T_3 remains at 20°C like in the isothermal case. The total heat which can be used in external processes is defined as heat extracted in the compression stages and heat released in HE 1. As described below the total heat release depends on x and η_{cool} and is shown in Fig. 5.

Influence of η_{cool} on heat extraction in HE 1

In this examination the liquid air ratio is kept constant at (1-x) = 0.1 whereas η_{cool} is varied between 0 and 0.7. The efficiency of HE 1 is determined by the value of η_{cool} whereby η_{cool} = 0 indicates that no heat is extracted in HE 1. Both a one-stage and two-stage compression is regarded and the total heat, provided by adiabatic compression and by HE 1 is shown in Fig. 5

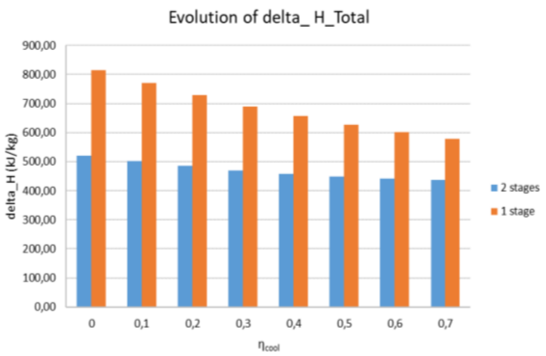

Figure 5: Evolution of the total recovered enthalpy for different values of η_{cool}

Here we can see that when η_{cool} is increasing, the total enthalpy recovered by cooling systems Δh_{total} is decreasing. While the enthalpy between 3 and 4 that is extracted in the HE 1 is higher, the compression enthalpy is even lower because T_2 is decreasing a lot, until 111 K for η_{cool} = 0.7.

If η_{cool} were greater than 0.7, the temperature T_9 would be lower than the boiling temperature, which is impossible because we need gaseous air in this part.

When the number of compression stages increases, the total enthalpy recovered by cooling systems is lower (-36% for 1 to 2 stages), because the maximum temperature of the compression is lower. But the influence of η_{cool} is also reduced: for a one-stage compression, the difference of total enthalpy recovered is 230 kJ/kg for $0 < \eta_{cool} < 0.7$, but only 80 kJ/kg for a two-stage compression.

The choice of η_{cool} depends on the type of cooling system. If the user prefers to give a lot of energy through HE1, he will have better to increase η_{cool}, but if he wants to recover as much energy as possible, he will have to decrease η_{cool} to take advantage of the compression. It has to be mentioned here, that a one-stage adiabatic compression leads to unreasonable high temperatures of 773°C which cannot be tolerated because which would lead to the destruction of the compressor. Taking a temperature limit of 500°C for the compressor into account, a one-stage compression cannot be realized.

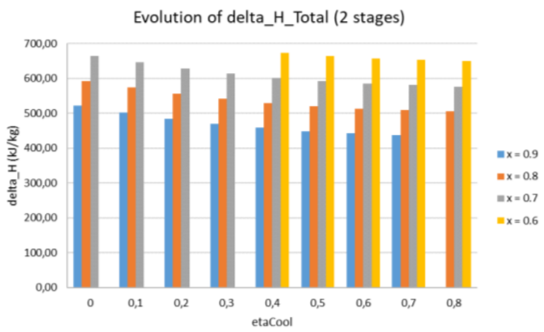

Figure 6: Total enthalpy recovered as function of x and η_{cool}.

As a consequence of the increasing yield of liquid air, the recirculated cold air (indicated as x) decreases, whereby the mixture temperature T_2 is increased. Therefore, more energy is released, in the compression heat exchangers and in HE 1, so that the total enthalpy (delta_H_total) increases, as shown in Figure 6. One more time, it is important to notice that some values are physically impossible: $\eta_{cool} <$ 0.4 for x = 0.6 and $\eta_{cool} >$ 0.7 for x = 0.9.

The goal of this storage system is to liquefy air in order to store energy. When x decreases, more liquid air can be produced. The previous results are therefore good news since the system allows to create more liquid air, while recovering more energy in the cooling systems. But the main problem of decreasing x is the significant limitation of the operating ranges for many parameters. For example, η_{cool} must be between 0.4 and 0.8 for x=0.6.

Influence of temperature T_3 (entrance of HE 1)

This temperature corresponds to the minimum temperature of the compressor cooling system and the maximal temperature for the HE 1. This means that the value of T_3 will be decisive concerning the choice of the system used to cool down the air in the compressor and the first heat exchanger.

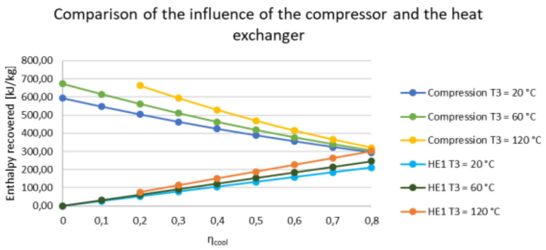

Figure 7: Influence of T_3 on recovered enthalpy from compressor cooling system and HE 1

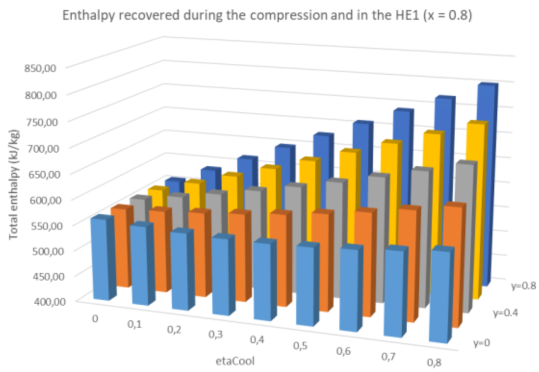

Figure 8: Influence of air extraction ratio y prior to mixing on total recovered enthalpy

The temperature of 60°C corresponds to the minimum temperature of a district heating system, and the temperature of 120°C has been chosen arbitrarily to see the behaviour of the enthalpy at high temperature. It can be seen from Fig. 7 that the higher the temperature T_3, the higher the total enthalpy recovered in the compressor and in the HE1. The second effect of the increase of temperature is the tightening of the range of η_{cool} values, for instance: with $T_3 = 120°C$, η_{cool} must be higher than 0.2. From Fig. 7 it can further be seen the impact of the enthalpy recovered during the compression. For low values of η_{cool}, the difference of the total enthalpy recovered is mainly due to the compression, with increasing impact of HE 1 for higher values of η_{cool}.

Influence of cold air extraction between point 8 and 9

At the point 9, we have got cold air with a mass flow rate of x that could be used for another application instead of being recycled in the cycle at the point 2. The idea is to extract a certain amount of air at the point 9, with a mass flow rate of y. This modification has not a significant impact on the system because it modifies only the temperature 2, which tend to $T_{ambient}$ when y tend to x.

The main result as can be seen from Fig. 8, is that the enthalpy extracted during the compression and in the HE1 is increasing with y and η_{cool}. This happens because the temperature T_2 is increasing when the remaining mass flow rate of the point 9 decreases, so the energy released by the compression will be higher.

Conclusion

This study provides a more detailed insight into the Linde-Hampson cycle for the liquefaction of air as an energy storage medium. Many parameters were varied to investigate their influence on the overall process. As a result of this study, coupling possibilities to other power generation processes, e.g. ORC cycles, are easier to identify.

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