MODELLING OF THE DYNAMIC BEHAVIOUR OF AN ADVANCED ADIABATIC COMPRESSED AIR ENERGY STORAGE (AA-CAES)

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Pero no hay colisión, ni ley ni gravedad, que te pueda hacer caer, aunque tiren a dar.

Pucho, Vetusta Morla.
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1. RESUMEN EN ESPAÑOL

1.1. Introducción

Debido al rápido agotamiento de las reservas de combustibles fósiles, los cuales proporcionan un porcentaje mayoritario de la energía en el mundo, es inevitable que ésta y las futuras generaciones de ingenieros investiguen y desarrollen nuevos métodos para satisfacer la creciente demanda energética de la población de manera ecológica y sostenible.

La gran apuesta para abastecer limpiamente la energía necesitada es la utilización de fuentes de energía renovables como la solar o la eólica, las cuales están cobrando cada vez más importancia y para 2030 deben representar el 27% de la oferta de energía de Europa, según el plan “2030 Energy Strategy” de la Unión Europea. Un gran inconveniente de este tipo de fuentes es la necesidad de utilizar su energía prácticamente simultáneamente a su generación (especialmente en el caso de la eólica) para que esta energía no sea perdida. Es decir, por ejemplo, un parque eólico funcionando de noche cuando la demanda de electricidad es baja está desaprovechando gran parte de la energía que genera.

Como solución a esto están empezando a cobrar especial importancia los métodos de almacenamiento temporal de energía excedente para su posterior uso en tiempos de picos de demanda y así dar más flexibilidad y potencial a las energías renovables.

Es en este contexto donde surge la idea del Almacenamiento de Energía por Aire Comprimido o CAES por sus siglas en inglés. Este método de almacenamiento de energía proporciona grandes ventajas sobre otros tipos, como pueden ser el almacenamiento mecánico o magnético, entre las cuales están una mayor capacidad de potencia y mayores tiempos de almacenamiento. Consta de tres etapas principales, las cuales son la compresión del aire y su almacenamiento, el calentamiento del aire previa su entrada a las turbinas, y la posterior expansión del aire almacenado para generar energía.

A pesar de ser una posible solución para el problema energético, el modelo tradicional de CAES, presente en ambas plantas existentes de este tipo actuales, se sigue dependiendo en ciertas etapas del proceso de la combustión de combustibles que contribuyen a la contaminación.

Con el objetivo de eliminar toda combustión del proceso y crear un sistema totalmente limpio y sostenible se desarrolla el modelo de Almacenamiento de Energía por Aire Comprimido Avanzado Adiabático (AA-CAES), cuyo esquema se muestra a continuación. Gracias al acople al sistema de un Almacenamiento de Energía Térmica (TES) para almacenar el calor que posteriormente calentará el

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aire antes de su expansión para maximizar la potencia obtenida, como se explicará a continuación, se elimina la etapa de combustión que tenía este fin en el CAES tradicional.

A continuación, se muestra un esquema del funcionamiento de un sistema de AA-CAES.

![Esquema de un sistema de AA-CAES. Fuente: European Association for the Storage of Energy.](image)

En este proyecto se pretende explicar el sistema, además de presentar un posible modelo termodinámico para dos posibles alternativas de diseño de cámara de aire, una de volumen y temperatura constante y otra de presión y temperatura constante, además de modelo para el almacenamiento de energía térmica, lo cual permitirá calcular, entre otros resultados, la capacidad de potencia de un sistema de este tipo y su eficiencia.

1.2. Objetivos

Siendo este un concepto muy moderno que aún no tiene ninguna aplicación real, pero sobre el cual existen varias investigaciones con muchas organizaciones implicadas (ver [28]), es importante analizar las posibilidades que presenta y analizar termodinámicamente sus componentes para conocer las potencias que ofrece, su eficiencia y sus posibles pérdidas térmicas.

Los objetivos de esta investigación son, por lo tanto:

- introducir el concepto de CAES además de sus diferentes vertientes que están siendo actualmente propuestas.
• desarrollar un modelo termodinámico para la cámara de almacenamiento de aire comprimido, estudiando los resultados para hasta tres etapas de compresor y de turbina y en función del número de ciclos llevados a cabo por el sistema.
• desarrollar además un modelo termodinámico para el Almacenamiento de Energía Térmica, mostrando de nuevo la dependencia de los resultados con el número de etapas de compresión y expansión y con el número de ciclos de carga y descarga completados.
• mostrar los resultados combinados ambos modelos.

1.3. Advanced Adiabatic Compressed Air (AA-CAES) energy storage

1.3.1. Concepto

Los sistemas de Almacenamiento de Energía por Aire Comprimido tradicionales funcionan de la siguiente manera. En primer lugar, la potencia sobrante proveniente de, por ejemplo, un parque eólico, se utiliza para accionar una serie de compresores que trabajan con aire. El aire comprimido, que está a altas temperaturas, se va almacenando en una cámara de almacenamiento subterránea. Cuando se necesita energía, este aire comprimido sale de la cámara de almacenamiento y se calienta mediante una fuente externa de calor (por la combustión de un combustible fósil) para finalmente entrar en las turbinas a mayor temperatura, pudiéndose así extraer más trabajo de ésta y consiguiendo mayor capacidad de potencia.

Para poder tener un sistema autosuficiente, sin necesidad de una fuente externa de calor, eliminando también la necesidad de combustión y así reducir totalmente las relacionadas con este tipo de planta, el AA-CAES intenta aprovechar las altas temperaturas del aire tras ser comprimido para almacenar una alta cantidad de calor (que en el CAES tradicional era perdido al almacenar directamente) en el TES. De esta manera, cuando se necesite recalentar el aire antes de su expansión, se puede extraer este calor del Almacenamiento de Energía Térmica, consiguiente un proceso independiente y sostenible. La palabra Adiabático, de las siglas AA-CAES del proceso en inglés, hace referencia a este calor que se almacena y se extrae del Almacenamiento de Energía Térmica sin salir del sistema.

1.3.2. Tipos de CAES

Junto a esta mejora propuesta en el AA-CAES, han surgido nuevas innovaciones para los sistemas de almacenamiento de energía, como lo son, por ejemplo:
• el Isobaric A-CAES, que trata de mejorar la eficiencia del proceso mediante la utilización de un fluido volátil en equilibrio con el aire comprimido, en una cámara de presión constante en la que ambos fluidos están separados por un pistón.
• el Isothermal CAES, en el cual se utilizan compresores y turbinas con procesos casi isotérmicos, lo cual implica eliminar la necesidad de almacenar y extraer el calor de un TES.
• el High Temperature Hybrid CAES (HTH-CAES), el cual es estudiado como posible mejora, y que incluye un segundo almacenamiento térmico de alta temperatura antes de las turbinas que consigue aumentar aún más la temperatura del aire. Con esto se aumenta la capacidad de potencia del proceso a la vez que reduce el volumen de cámara de aire necesario.

1.3.2. Componentes

Para poder llevar a cabo el proceso se necesitan, el sistema necesita los siguientes componentes:

1. Compresores.
   Utilizados para aumentar la presión del aire, considerado como gas ideal, utilizado antes de almacenarlo en la cámara. Se consideran adiabáticos en el proceso. Debido a al alto caudal másico necesario en el proceso, los compresores más adecuados son los axiales.

2. Cámara de almacenamiento de aire.
   Es clave en el proceso ya que es donde se almacena el aire a altas presiones antes de ser reutilizado. Aunque se puede hacer en depósitos ideales, es muy ventajoso construirlos a partir de reservas naturales, especialmente de cavidades salinas ya existentes y en desuso.

   Este tipo de cavidades naturales se acoplan perfectamente al sistema: sus paredes tienen grandes propiedades para aguantar las altas presiones y sus continuas variaciones además de ser impermeables y no porosas.

   Asimismo, no suponen altos gastos de inversión ni de construcción, y su proceso de acondicionamiento y preparación es relativamente simple. Consiste en la inyección de agua en la cavidad salina obsoleta para disolver la sal y su posterior extracción, dejando vacío un gran volumen para el almacenamiento de aire. Por esta razón, ubicar una planta de AA-CAES en regiones con depósitos subterráneos de sal sería muy aconsejable. De hecho, una de las dos plantas actualmente activas de CAES, en Huntorf, Alemania, utiliza este tipo de cámara de aire.

   Cabe mencionar en este apartado que existe otro factor importante para la implantación del proceso en una ubicación conveniente. Debido a la gran adaptabilidad que tiene el sistema con la energía eólica, se deben también considerar importantes las zonas con parques eólicos ya operativos o con potencial
para tenerlos, ya que localizar la planta en estas regiones permite reducir las pérdidas asociadas al transporte de electricidad desde el parque hasta la posible planta de AA-CAES. En el mapa a continuación de muestran en azul las zonas con cavidades salinas subterráneas y en círculos rojos las zonas de buena calidad de viento. Por lo tanto, las zonas azules dentro de los círculos rojos son las más adecuadas para la instalación de un sistema de este tipo.

![Mapa de zonas adecuadas para AA-CAES](image)

**Figure 2.** Zonas de buena calidad de viento y con presencia de cavidades salinas en para la cámara de almacenamiento de aire. Fuente: [28].

Para cumplir uno de los objetivos fundamentales de AA-CAES y eliminar por completo la utilización de un combustible en el proceso, es fundamental el papel del TES, que debe poder almacenar y aportar cantidades de calor similares a aquellas que se podrían extraer de una combustión.

El aporte de calor antes de la expansión es necesario para aumentar la eficiencia del sistema de almacenamiento de energía, que se define como el trabajo generado en las turbinas entre el consumido por los compresores. Con el Almacenamiento Térmico de Energía, se almacena el calor previamente extraído en intercambiadores de calor mediante otro fluido caloportador durante el proceso de carga. Este calor se transmite durante el proceso de descarga del sistema al aire que entra en las turbinas, de nuevo mediante un fluido caloportador y unos intercambiadores de calor, lo cual supone un mayor trabajo producido, como se muestra a continuación (considerando compresores y turbinas adiabáticas).

\[
\begin{align*}
\{ \frac{h}{q} &= q + w, \quad h = w_t = c_p \Delta T \rightarrow \Delta T \rightarrow w_t \rightarrow P
\end{align*}
\]

El Almacenamiento Térmico se puede clasificar con distintos criterios. En primer lugar, se puede clasificar según el método de almacenamiento de calor, dependiendo de si es sensible, latente o termoquímico. El utilizado en esta investigación es el sensible, consistente en el almacenamiento
mediante un incremento en la temperatura de un material. Este depende, para un valor de masa constante, de la capacidad calorífica específica del material y de la variación de temperatura que sufre.

El material de almacenamiento puede ser tanto líquido como sólido, lo cual lleva a una segunda clasificación de los TES: los almacenamientos activos, cuyo medio de almacenamiento cumple tanto esa función como la de fluido caloportador, y los pasivos, que necesitan a parte del medio un fluido que transporte el calor que poseen.

En un primer lugar, este proyecto se planteó la utilización de agua como medio de almacenamiento debido a sus buenas propiedades térmicas y la posibilidad de utilizarlo como medio activo. Sin embargo, las altas temperaturas que se alcanzan durante el proceso, mostradas en los resultados, hacen imposible su utilización puesto que sufriría constantes cambios de fase. Se optó por esta razón por un almacenamiento pasivo de lecho de rocas empaquetadas.

El sistema se considera en proceso de carga cuando se está comprimiendo aire y almacenando calor, y en proceso de descarga cuando el aire se está expandiendo y por lo tanto se está generando energía. Los esquemas de los procesos de carga y descarga del Almacenamiento de Energía Térmica se muestran a continuación. Los esquemas muestran tres etapas de compresión en la carga y tres de expansión en la descarga. Para sistemas con una o dos etapas, el sistema sería equivalente, pero con uno o dos compresores y turbinas.

![Esquema del proceso de carga del Almacenamiento de Energía Térmica](image)

Figure 3. Esquema del proceso de carga del Almacenamiento de Energía Térmica. Fuente: Elaboración propia.
3. **Turbinas.**

El fluido de trabajo utilizado es aire, lo cual implica la utilización de turbinas de gas. Igual que los compresores, las turbinas se pueden clasificar según las direcciones de los flujos de entrada y de salida. En este caso las turbinas utilizadas también serán axiales, las cuales dan mejores rendimientos cuando se utilizan fluidos compresibles, como es el aire.

5. **Intercambiadores de calor.**

Los intercambiadores de calor son claves para transmitir el calor desde el fluido de trabajo al Almacenamiento Térmico en el proceso de carga, y viceversa en el de descarga. Se pueden clasificar según la dirección relativa de flujo entre los dos fluidos (paralela o cruzada, y a corriente o a contracorriente), según la configuración geométrica del intercambiador o según la naturaleza de los fluidos utilizados, que en el caso de este proyecto es de gas-gas.
1.3.3. Esquema del proceso

A continuación, se muestra el esquema general del proceso en el cual también aparecen algunas variables de temperatura y presión en distintos puntos del sistema.

![Esquema general de un sistema de Almacenamiento de Energía por Aire Comprimido](image)

Las figuras 3 y 4 de la página anterior son también esquemas de la carga y descarga del sistema, y muestran de manera específica las temperaturas de entrada y salida a cada etapa del compresor o turbina y a los intercambiadores de calor. Una vez conocidas las temperaturas, puede calcularse el trabajo que utilizan los compresores y que generan las turbinas durante el proceso.

1.3.4. Modelo termodinámico de la cámara de aire

En primer lugar, se procede a modelizar la cámara de aire con dos alternativas como se ha mencionado anteriormente. Una será una cámara de aire isoterma e isócora y la otra isoterma es isobara. Para ambas opciones, se estudia también la variación de los resultados en función del número de etapas de compresión y expansión, que se define como $N$ y cuyo máximo valor es 3.

**Compresores**

Las condiciones que se fijan para los compresores son las siguientes. El aire entrante al primer compresor se obtiene del ambiente y por lo tanto se define la temperatura de entrada al primer compresor como, $T_{c1} = T_0$. 
Para calcular las temperaturas de salida de los compresores a partir de las de entrada, se utiliza la Ley de Poisson, considerando el aire como un gas ideal y definiendo la relación de compresión con la letra $\beta$:

$$T_{ci}^f = T_{ci} \beta^N$$  \hspace{1cm} (1)

Por otra parte, se aclara que la temperatura de salida del último compresor se considera igual a la de entrada a la cámara de aire.

**Cámara de almacenamiento**

Al ser en ambos casos la cámara de aire isoterma, su temperatura es constante e igual a $T_0$. El gas utilizado y almacenado es aire, considerado como un gas ideal, y por lo tanto se puede definir su diferencial de masa con la ecuación de estado de los gases ideales de la siguiente manera:

- para la cámara isócóra, $dm = \frac{V}{RT_0} dp$,
- para la cámara isobara, $dm = \frac{p}{RT_0} dV$.

Además, la relación de presión de la cámara de aire en cualquier momento se define como $\beta_i = \frac{p_i}{p_0}$, y se considera que la presión de salida de la última etapa del compresor es igual a la existente dentro de la cámara.

**Intercambiadores de calor**

De igual forma se deben de definir las condiciones de los intercambiadores de calor. En primer lugar, está la eficiencia, $\varepsilon$, que se fija en un valor conservador de 0.7. Asimismo, se define una relación entre las capacidades caloríficas de ambos fluidos (el de trabajo y el portador de calor) que entran en el intercambiador igual a la unidad, es decir:

$$(mc_p)_{HTF} = (mc_p)_{WF} = c_{p\text{air}} \int dm = \frac{c_{p\text{air}} V p_0}{R T_0} \int_{p_1}^{p_2} \frac{dp}{p_0} = \frac{p_0 V}{\alpha T_0} (\beta_2 - \beta_1)$$

Considerando que los intercambiadores se colocan en paralelo con respecto al almacenamiento térmico, como se muestra en las figuras 3 y 4, este valor se debe multiplicar por el número de intercambiadores utilizados, que es igual al número de etapas N.

**Turbinas**

En las turbinas, es necesario definir las temperaturas de entrada, lo cual se consigue mediante las ecuaciones de la eficiencia de los intercambiadores, como se verá a continuación. Las temperaturas de salida se calculan de nuevo con la Ley de Poisson.

$$T_{ti}^f = T_{ti} \beta^N$$  \hspace{1cm} (2)
1.3.4.1. Trabajo y calor del sistema

Una vez conocidas las condiciones, se procede a determinar las ecuaciones que permiten el cálculo de los trabajos de compresión y de expansión y del calor almacenado y liberado por el Almacenamiento de Energía Térmica.

Proceso de carga

La temperatura entrante al compresor $i$ de un sistema con un total de $N$ etapas se calcula con la ecuación de la eficiencia de los intercambiadores de calor y la relación establecida entre las capacidades caloríficas, en la que la variable $T^i_{TES,n}$ es la temperatura del almacenamiento térmico al principio del proceso de carga del ciclo $n$, que para $n = 1$ equivale a $T_0$.

$$T_{ci} = (1 - \varepsilon)T_{ci-1}\beta^{-1}T^i_{TES,n}$$  \hspace{1cm} (3)

Siendo los compresores adiabáticos, el trabajo realizado por ellos se calcula con la primera ley de la termodinámica.

$$dW_c = dH_c = \frac{c_p}{\eta_{sc}}[\beta^{-1} - 1]\left(\sum_{i=1}^{N} T_{ci}\right)dm$$

Para el calor almacenado se sigue un razonamiento similar, y, teniendo en cuenta que el trabajo de los intercambiadores de presión es nulo, el calor que se almacena en el fluido caloportador y posteriormente en el TES se calcula también con la primera ley de la termodinámica.

$$dQ_s = dH = c_p \sum_{i=1}^{N} (T_{ci+1} - T_{ci}\beta^{-1}) dm$$

La temperatura final del almacenamiento térmico se puede calcular conocido $Q_s$ de la siguiente manera:

$$T^f_{TES,n} = \frac{Q_s}{mc_p} + T^i_{TES,n}$$

Proceso de descarga

Análogamente al proceso de carga, la temperatura de entrada a la turbina $i$ de un total de $N$ se calculan con la ecuación de la eficiencia de los intercambiadores de calor.

$$T_{ti} = (1 - \varepsilon)T_{ti-1}\beta^{-1}T^i_{TES,n}$$

Las turbinas también se consideran adiabáticas al igual que los compresores, y de nuevo para calcular del calor liberado se parte de que el trabajo realizado por los intercambiadores de calor es nulo. Por lo tanto, el trabajo de expansión en las turbinas y el calor liberado por el TES se pueden calcular con la primera ley de la termodinámica.
\[ dW_t = dH_t = -c_p \eta_{st} \left( \sum_{i=1}^{N} T_{ti} \left( 1 - \beta^{-\frac{a}{N}} \right) \right) dm \tag{4} \]
\[ dQ_s = dH = -c_p \left( T_{t1} - T_0 + \varepsilon \sum_{i=2}^{N} \left( T_{TES} - T_{i-1} \beta^{-\frac{a}{N}} \right) \right) dm \tag{5} \]

### 1.3.4.2. Cámara de presión y temperatura constante (modelo V, T)

Para calcular el valor del trabajo y calor del sistema en este modelo, se introduce el diferencial de masa correspondiente a la cámara isócora, \( dm = \frac{V}{RT_0} dp \), en las ecuaciones y se integra entre las presiones máxima y mínima, \( p_2 \) y \( p_1 \), del almacenamiento de aire. Estas presiones valen 66 bar y 50 bar respectivamente.

### 1.3.4.3. Cámara de presión y temperatura constante (modelo P, T)

Para llegar a los resultados en este caso, se utiliza el diferencial de masa definido anteriormente para la cámara isobara, \( dm = \frac{P}{RT_0} dV \). Antes de integrar, se deben relacionar los volúmenes máximo y mínimo con las presiones máximas y mínimas definidas para el modelo V, T.

El volumen máximo, \( V_2 \), que es igual al volumen total de la cámara de aire \( V \), correspondería a la presión mínima \( p_1 \) de la cámara V, T. Por lo tanto, la relación entre los volúmenes y las presiones según la ecuación de los gases ideales es la siguiente:

\[ V_2 p_{\text{min}} = V_1 p_{\text{max}} \Rightarrow V_1 \frac{p_1}{p_2} = \frac{p_1}{p_2} \]

Una vez conocida esta relación, se procede a integrar las ecuaciones del trabajo y el calor igual que en el caso anterior.

### 1.3.5. Modelo termodinámico del almacenamiento de energía térmica

Tras el estudio de varios posibles medios de almacenamiento de calor, tanto activos como pasivos, se determinó que la solución más viable, tanto por coste como por condiciones de temperatura muy específicas que hacían imposible el uso de otras opciones, es utilizar aire como fluido caloportador y un lecho de rocas de granito como medio de almacenamiento térmico. Además de ser más económico y simple, proporciona un intercambio de calor efectivo entre el fluido y las rocas debido a su contacto directo. La porosidad, \( \phi \), del almacenamiento térmico se ha fijado en un valor común en la literatura de este tipo de lechos, igual a 0.35. El siguiente esquema muestra como serían los procesos de carga y descarga del almacenamiento.
La dirección del flujo es opuesta en el proceso de carga y descarga para una mejor eficiencia en la transmisión de calor aprovechando los fenómenos de estratificación de temperaturas que se dan en este tipo de almacenamientos.

1.3.5.1. Dimensionamiento

Al igual que en el caso de los intercambiadores, se precisa que la capacidad calorífica del almacenamiento térmico es igual a la del aire con el que se trabaja y por lo tanto al del aire caloportador. Tras el proceso de carga, se considera que el TES acaba a la temperatura del aire entrante y que el aire sale a la temperatura inicial del Almacenamiento de Energía Térmica. A pesar de que esto es una idealización, las pérdidas de calor consideradas en los intercambiadores de calor hacen que la transmisión de calor en el TES sea la misma que si el aire fluyera directamente después del compresor por el Almacenamiento de Energía Térmica y transmitiera su calor con una eficiencia del 0.7, lo cual hace válido el modelo, a fin de cuentas.

Las dimensiones necesarias para el almacenamiento de calor se calculan igualando las capacidades caloríficas como explicado anteriormente. Al aumentar la capacidad calorífica total del fluido caloportador necesaria cuando aumenta el número de etapas N, también lo hará el volumen del TES. Por otra parte, la relación entre la altura y el radio del depósito se debe fijar para poder determinar las medidas de éste. Teniendo esto en cuenta, resolviendo el siguiente sistema de ecuaciones, conocida la capacidad calorífica para un número de etapas $N \left( mc_p \right)_N$, se calcula el volumen total y las dimensiones del almacenamiento térmico.
\[
\left\{ \begin{array}{c}
\frac{h_{TES}}{r_{TES}} = 5 \\
r_{TES} = \sqrt[3]{\frac{V_{TES}}{5\pi}} \\
V_{TES} = \frac{(mc_p)_N}{\phi \rho_{air} c_p r_{air} + (1 - \phi) \rho_{rocks} c_p_{rocks}}
\end{array} \right.
\] 

(6)

Los resultados obtenidos son los siguientes.

<table>
<thead>
<tr>
<th>Number of stages</th>
<th>( V_{TES}(m^3) )</th>
<th>( r_{TES}(m) )</th>
<th>( h(m) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3595,3</td>
<td>6,1</td>
<td>30,6</td>
</tr>
<tr>
<td>2</td>
<td>7190,7</td>
<td>7,7</td>
<td>38,5</td>
</tr>
<tr>
<td>3</td>
<td>10786,0</td>
<td>8,8</td>
<td>44,1</td>
</tr>
</tbody>
</table>

Table 1. Dimensiones necesarias del TES según el número de etapas de compresión/expansión. Fuente: Elaboración propia.

1.3.5.2. Pérdidas de calor

Con el fin de determinar cómo varían los resultados del sistema a medida que se van cumpliendo ciclos de carga y descarga antes de llegar al régimen permanente, se calculan las pérdidas de calor del Almacenamiento Térmico. Una vez conocidas, se puede calcular la temperatura final alcanzada en el TES tras un ciclo, que equivale a la inicial del siguiente ciclo, según la siguiente fórmula.

\[
(\rho V c_p)_{TES} \frac{dT_{TES}}{dt} = \frac{Q_s}{t_{charge}} - \frac{Q_r}{t_{discharge}} - \dot{Q}_{loss}
\]

(7)

El último término, correspondiendo a las pérdidas de calor se calcula de la siguiente manera.

\[
\dot{Q}_{loss} = U A_{outer} (T_{TES} - T_0)
\]

(8)

Una vez conocida la temperatura final del TES tras un ciclo, pueden calcularse los nuevos valores del trabajo de expansión en las turbinas y por tanto de la capacidad de potencia del sistema. El valor de UA, siendo U el coeficiente global de trasferencia de calor y A la superficie de contacto entre el depósito y el ambiente, es función de los coeficientes convectivos de transferencia de calor interior y exterior y de la conductividad térmica del aislante térmico utilizado, que en este caso es fibra de vidrio, y de su espesor (ecuación (65)).
La conductividad térmica de la fibra de vidrio es un dato conocido. Para el coeficiente convectivo exterior, \( h_e \), teniendo en cuenta que la planta probablemente estaría localizada en una zona de viento abundante por estar cerca de un parque eólico, se estima un valor considerablemente alto de 20 \( \frac{W}{m^2K} \).

El coeficiente convectivo interior, \( h_i \), se calcula utilizando una correlación aproximada desarrollada específicamente para este tipo de depósitos, en el que \( D_r \) es el diámetro equivalente de las rocas de granito utilizadas, y Re y Pr los números adimensionales de Reynolds y Prandtl, calculados fácilmente conocidas las propiedades del fluido utilizado a la temperatura media del depósito:

\[
h_i = \left( \frac{\lambda_{\text{air}}}{D_r} \right) \left( 2.576Re^{\frac{1}{3}}Pr^{\frac{1}{3}} + 0.0936Re^{0.8}Pr^{0.4} \right)^2 \tag{9}
\]

Los resultados obtenidos para los coeficientes convectivos y el valor del coeficiente global de transferencia de calor se muestran a continuación.

<table>
<thead>
<tr>
<th>Número de etapas</th>
<th>Número de Reynolds</th>
<th>Tipo de flujo</th>
<th>( h_i ) [( \frac{W}{m^2K} )]</th>
<th>( h_e ) [( \frac{W}{m^2K} )]</th>
<th>( UA ) [( \frac{W}{K} )]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>97248,87</td>
<td>Turbulente</td>
<td>202.59</td>
<td>20</td>
<td>159.40</td>
</tr>
<tr>
<td>2</td>
<td>122525,90</td>
<td>Turbulente</td>
<td>240.80</td>
<td>20</td>
<td>251.83</td>
</tr>
<tr>
<td>3</td>
<td>140257,15</td>
<td>Turbulente</td>
<td>266.41</td>
<td>20</td>
<td>549.26</td>
</tr>
</tbody>
</table>

Table 2. Coeficientes convectivos y globales de transferencia de calor del TES según el número de etapas. Fuente: Elaboración propia.

Con los coeficientes globales de transferencia de calor, se calculan las pérdidas del Almacenamiento de Energía Térmica para ambos casos de cámara de aire en función del número de etapas del proceso.

---

Como se muestra en la gráfica, aumentar el número de etapas de compresión y expansión reduce considerablemente las pérdidas de calor del Almacenamiento Térmico. La incorporación de una segunda etapa reduce las pérdidas en un 25.7%, y una tercera etapa haría que las pérdidas bajaran un 28.2% con respecto de aquellas de una sola etapa. Esto se debe a que las temperaturas alcanzadas en el proceso son menores a medida que se añaden etapas.

Figure 8. Pérdidas de calor del TES para un sistema con cámara de aire V, T.

Figure 7. Pérdidas de calor del TES para un sistema con modelo de cámara de aire P, T.
En el caso de la cámara de aire de presión y temperatura constante, los valores de las pérdidas son más altos para cualquier número de etapas respecto al otro modelo. La diferencia entre las pérdidas de N=1 y N=2 es más grande aún, sufriendo una caída de 42.7% al añadir la segunda etapa y una caída del 44.7% si se añaden dos etapas más al proceso. Esto se debe a que se alcanzan temperaturas muy altas con una etapa (hasta 967 K) debido al hecho de que la temperatura del TES es función del calor almacenado, que siempre se reduce con el número de ciclos excepto para el proceso con N=1, en cuyo caso se mantiene constante, como se ve en la ecuación (45).

Ambos modelos de cámara de aire muestran la misma tendencia a medida que aumenta el número de ciclos. La evolución con el número de ciclos de carga y descarga (es decir con el tiempo) es clara: el valor de las pérdidas aumenta con éste provocado por el aumento de temperaturas que se verá en el siguiente apartado. Se comprueba también que, para el modelo V, T con dos y tres etapas de compresión y expansión, se tardan más ciclos en llegar a un régimen permanente que con una sola etapa. En el modelo P, T, se llega a un régimen permanente en 4 ciclos, independientemente del número de etapas del proceso.

1.3.6. Resultados

Tras haber cuantificado las pérdidas del Almacenamiento de Energía Térmica, los resultados finales pueden ser calculados, incluyéndose aquellos de una tercera etapa a pesar de los altos valores que implica de caudal másico con fines comparativos. Las pérdidas han sido modelizadas para un ciclo entero de carga y descarga en el que se tiene una etapa de 6 horas de carga (compresión y almacenamiento), 6 horas de inactividad, 6 horas de descarga (expansión) y otras 6 horas finales de inactividad. Por ello la temperatura del TES al final de un ciclo es igual a la inicial del siguiente.

Se presentan los resultados de las temperaturas máxima y mínima del TES, las cuales son importantes a la hora de valorar la posibilidad de utilizar fluidos caloportadores diferentes, el trabajo de expansión adimensional, el cual es una medida de la densidad de energía del sistema, la eficiencia del sistema (definida como el trabajo de expansión dividido por el de compresión), y su capacidad de potencia. Se considera que el volumen de la cámara de aire es de 300000 m³, el cual es un valor común similar al utilizado, por ejemplo, en la planta de CAES de Huntorf, y una presión ambiente de 100000 MPa. Los resultados se han calculado mediante el documento Excel elaborado (ver 12.2. APPENDIX 2: Guide to Excel document for the calculations and graphs).

A continuación, también se discutirán brevemente los resultados, con un análisis completo más adelante en la sección.
6.5. Results of the model and evolution with the number of stages

1.3.6.1. Modelo V, T
Temperaturas TES

Para cada color, las curvas superiores marcan la temperatura máxima correspondiente a un sistema con ese número de etapas, y la inferior la mínima. Como se puede observar, ambas temperaturas decresen con el número de etapas. Al añadir una segunda etapa, la temperatura mínima desciende un 18.9% y la máxima un 30.9%, reduciéndose así la diferencia entre amabas. Añadir una tercera etapa supone un descenso de otro 5.88% en la temperatura mínima y de un 15.1% en la máxima.

En cuanto a la evolución con el número de ciclos completados, a medida que crecen también lo hacen ambas temperaturas. La entrada en régimen estacionario de las temperaturas se produce más tarde para los procesos con dos y tres etapas de compresión que para el de una sola.

Trabajo de expansión y capacidad de potencia

Las gráficas de ambas variables siguen la misma variación y por tanto tienen la misma forma, debido a su proporcionalidad (mostrada en la ecuación (68)).
Figure 10. Evolución del trabajo de expansión adimensional con el número de ciclos completados, para el modelo V, T. Fuente: Elaboración propia.

Figure 11. Capacidad de potencia del sistema en función del número de ciclos, para el modelo V, T. Fuente: Elaboración propia.
La gráfica muestra que la diferencia de potencia producida de los modelos no es muy distinta al variar el número de etapas. Al pasar de una etapa a dos se baja un 6.6%, con otro descenso del 3.7% con la inclusión de una tercera etapa.

Por otra parte, a medida que avanza el tiempo y se completan más ciclos de carga y descarga, la capacidad de potencia incremente considerablemente sin importar el número de etapas, con mayor intensidad en los primeros dos ciclos. De nuevo la potencia del proceso de una sola etapa tarda menos en llegar a un régimen permanente que las otras dos alternativas.

**Eficiencia**

![Gráfico de eficiencia según el número de ciclos](image)

**Figure 12.** Eficiencia del sistema en función del número de ciclos para el modelo V, T. Fuente: Elaboración propia.

La evolución en la eficiencia es la contrario con respecto al número de etapas. En este caso, al aumentar N también aumenta la eficiencia a pesar de reducir su capacidad de potencia. El paso de una etapa a dos supone un rendimiento un 8.19% mayor, y la inclusión de otra etapa más supone un aumento adicional de un 5.93%.

La variación de la eficiencia con el número de etapas es menos apreciable que la de las otras variables (sobre todo para N=2 y N=3), pero muestra una tendencia ascendente. La eficiencia se hace constante con menos etapas debido a que las variaciones a partir de la tercera etapa en el trabajo de compresión y expansión son muy pequeñas respecto a su valor real, lo cual provoca cambios casi inapreciables en la eficiencia.
1.3.6.2. Modelo P, T

Para el modelo P, T, la evolución de las temperaturas máxima y mínima del Almacenamiento de Energía Térmica es muy similar a aquella del anterior modelo. Sin embargo, los valores son mayores y también lo son las variaciones de estos. Pasar de una etapa de compresión a dos supone una caída del 42.3% en la temperatura máxima y de 26.8% en los de la mínima.

A medida que se completan ciclos de carga y descarga, las temperaturas, dándose los cambios más importantes en los primeros ciclos tras la puesta en marcha del proceso. Estos cambios también son más pronunciados cuando más altas sean las temperaturas, y por lo tanto al bajar el número de etapas de compresor y turbina. El número de ciclos completados antes de que se alcancen valores constantes, para este modelo, independiente del número de etapas.

**Figure 13.** Temperaturas máxima y mínima del TES para el modelo P, T. Fuente: Elaboración propia.
Trabajo de expansión y capacidad de potencia

Igual que con el modelo V, T, un aumento en el número de etapas de compresión supone un descenso en la capacidad de potencia del sistema. Esta caída es mayor al pasar de un proceso de una etapa a una de dos, tomando un valor del 20.0%. Al incorporarse una tercera etapa, la capacidad de...
potencia sólo decrece un 3.78% más, lo cual significa un cambio casi inapreciable.
A medida que se completan ciclos, la capacidad de potencia también aumenta para las tres alternativas. Este incremento es mucho mayor en los primeros dos ciclos. Independientemente del número de etapas, en este modelo el régimen permanente se alcanza una vez completados 10 ciclos.

**Eficiencia**

![Diagrama de eficiencia según el número de ciclos](image)

**Figure 16.** Eficiencia en función del número de ciclos para el modelo V, T. Fuente: Elaboración propia.

Las curvas de eficiencia muestran una tendencia diferente en el modelo P, T, especialmente la que corresponde a un proceso de una sola etapa. A pesar de ser éste el que más ineficiente al final del primer ciclo, y el intermedio en los ciclos 2 y 3, su rápido crecimiento hace que a partir del 4º ciclo sea la alternativa con mayor eficiencia. Este fuerte crecimiento se debe a que el trabajo de compresión para este proceso es constante e independiente de las temperaturas del TES (ecuación (43)), al contrario que con dos y tres etapas, mientras que el trabajo de expansión crece con la temperatura del TES (como muestra la Figura 14) y por lo tanto con el número de ciclos. Después del segundo ciclo, el proceso con N=2 es el menos eficiente, mientras que el de N=3 es el de mayor eficiencia hasta el ciclo cuatro en el que es superado.

La eficiencia de las tres alternativas crece con el número de ciclos, como en el caso del modelo V, T. Los procesos con N=2 y N=3, cuyo crecimiento es muy reducido, alcanzan el régimen permanente de eficiencia tras tres ciclos completos, con el de N=1 necesitando hasta 6 ciclos de carga y descarga para llegar a un valor constante con el tiempo.
1.3.6.3. Comparación con otras fuentes de energía

Con la finalidad de poner en perspectiva los valores de potencia obtenidos en la investigación, estos se comparan a continuación con aquellos que tienen distintas plantas de producción de energía, entre las que se incluyen el parque fotovoltaico de Olmedilla, el más grande de España, la central del Embalse de Alcántara, y las dos plantas de CAES tradicional existentes en el mundo en la actualidad, localizadas en Huntorf, Alemania, y en McIntosh, USA.

![Diagrama de potencia de diferentes fuentes renovables](image)

**Figure 17.** Comparación de la capacidad máxima de los modelos de AA-CAES con otras fuentes de energía. **Fuente:** Elaboración propia.

Las distintas alternativas del AA-CAES se muestran en amarillo. Se comprueba que se obtienen valores similares a aquellos de la planta de McIntosh para el modelo P, T con una etapa de compresión y expansión, mientras que las otras alternativas consiguen potencias un poco menores a estas. De esto se concluye que es posible obtener potencias similares con un sistema de AA-CAES totalmente sostenible que con una planta de CAES tradicional que sigue empleando combustibles fósiles y por tanto emitiendo gases contaminantes. La planta de Huntorf tiene potencias mucho mayores, debido sobre todo a sus dos cámaras de almacenamiento de aire y a su rápida fase de descarga, que tarda 3 horas y tiene caudales máximos de 417 kg/s.

En comparación con el parque solar de Olmedilla, la potencia generada es mayor en los procesos de AA-CAES estudiados. Sin embargo, al estar limitados por la potencia del parque eólico cuya potencia es utilizada para accionar los compresores, los valores de capacidad están limitados y son mucho menores que los obtenidos en platas hidroeléctricas como la de Alcántara.
Finalmente, teniendo en cuenta que una persona en España consume de media 550W de electricidad, se muestra a continuación el número de personas cuyas necesidades energéticas podrían ser abastecidas por una planta de AA-CAES.

<table>
<thead>
<tr>
<th>Model</th>
<th>Number of people supplied</th>
</tr>
</thead>
<tbody>
<tr>
<td>V, T ; N=1</td>
<td>169024</td>
</tr>
<tr>
<td>V, T ; N=2</td>
<td>157865</td>
</tr>
<tr>
<td>V, T ; N=3</td>
<td>152176</td>
</tr>
<tr>
<td>P, T ; N=1</td>
<td>206184</td>
</tr>
<tr>
<td>P, T ; N=2</td>
<td>164862</td>
</tr>
<tr>
<td>P, T ; N=3</td>
<td>158622</td>
</tr>
</tbody>
</table>

Tabla 3. Número de personas que pueden ser abastecidas por los diferentes modelos de AA-CAES. Fuente: Elaboración propia.

1.3.7. Conclusiones

En líneas generales, se debe mencionar que la modelización del sistema de AA-CAES de este trabajo, que evidentemente tiene sus limitaciones, sirve solo como un primer paso en la investigación de esta tecnología para la que se deben considerar muchos más factores que han quedado fuera del alcance del texto.

De los resultados se pueden extraer varias conclusiones claras. En primer lugar, que independientemente del tipo de cámara de aire utilizada, la evolución de las variables con el número de etapas de compresión y expansión, $N$, y con el número de ciclos del sistema completados, $n$, es similar. Las temperaturas máxima y mínima, el trabajo de los compresores y las turbinas, las pérdidas de calor crecen y la eficiencia baja a medida que aumenta $n$ y disminuye $N$. La elección más lógica desde el punto de vista tanto de resultados y de la viabilidad es la de un proceso de dos etapas de compresión y expansión, el cual es más eficiente y estable que el de una sola etapa y tiene un caudal másico más realista que un proceso de tres etapas, además de tener una capacidad de potencia aceptable.

Además, se comprueba que la cámara de aire isobara e isoterma (P, T) proporciona mayores potencias y eficiencias más altas que la isócora e isoterma (V, T), siendo por lo tanto un sistema mejor.
termodinámicamente. Sin embargo, sería necesario estudiar la viabilidad económica y tecnológica de este modelo, siendo probablemente más costosa de construir y mantener.

Los valores obtenidos de capacidad de potencia de la instalación, a pesar de ser mucho menores que los de otros tipos de plantas como una hidroeléctrica o con plantas de energía tradicionales, es aceptable. Es más alto que el de todas las plantas solares y se encuentra en el rango de la planta de CAES tradicional de McIntosh, Alabama, pero evidentemente sin la necesidad de utilizar combustibles fósiles y por tanto siendo más ecológica y sostenible. Con el modelo planteado se podrían abastecer entre 152000 y 206000 personas en España, en función del tipo de cámara de aire y de etapas de compresión/expansión.

Con los objetivos energéticos y medioambientales fijados por la Unión Europea, la implantación de un sistema de este tipo está cada vez más cerca, como muestran proyectos como el ADELE-ING y el RICAS2020 existentes actualmente en este ámbito (ver el Apéndice 4). Las condiciones climáticas y geográficas en muchas zonas de Europa son favorables para la construcción de una planta cercana a un parque eólico existente o potencial y sobre un depósito de sal donde se colocaría la cámara de aire.

Sin embargo, aún quedan avances necesarios antes de que una planta de esta índole sea una realidad. En primer lugar, el estudio de nuevos materiales y tecnologías que puedan soportar las altas exigencias mecánicas que el proceso requiere. Por otro lado, y de mayor importancia, está el impedimento que supone la baja viabilidad económica actual del proyecto, más aún teniendo en cuenta que con inversiones menores se pueden conseguir mejores resultados, por ejemplo, con plantas de producción de energía tradicionales.
1.4. Futuras mejoras

Como línea de investigación y futura mejora, se considera importante el concepto expuesto en el capítulo 5.1.3. High Temperature Hybrid CAES (HTH-CAES) y explicado en profundidad en el capítulo 8.1. Addition of a High Temperature Thermal Energy Storage (HHTES AA-CAES model).

La mejora del Sistema de AA-CAES que se estudia es la inclusión en el sistema de un segundo Almacenamiento de Energía Térmica, de altas temperaturas, denominado HTTES por sus siglas en inglés. Implicaría la utilización de una fuente de calor externa para esta parte del proceso, lo cual reduciría la independencia y sostenibilidad del sistema, pero conseguiría grandes mejoras en la capacidad de potencia, en la eficiencia de la planta, y por tanto también en la viabilidad económica del proyecto. El HTTES puede estar formado por un medio térmico económico ya que no se necesitan propiedades térmicas excepcionales, y el calor que se le aporta puede ser ajustado según las necesidades energéticas que se intentan satisfacer.

El calor aportado a este segundo TES para incrementar su temperatura puede tener distintos orígenes, como pueden ser, además de cualquier fuente convencional de calor, el de una resistencia eléctrica por el efecto Joule, la cual podría ser accionada por el parque eólico en el caso de tener éste energía excedente tras la carga del sistema, o el de una reacción exotémica.

Este segundo TES, por el cual fluye directamente el aire de trabajo para mayor superficie de intercambio de calor, se colocaría inmediatamente antes de la entrada a las turbinas del aire, y mediante un aumento extra de la temperatura de éste conseguiría un fuerte incremento tanto en el trabajo extraído por las turbinas como por tanto en la capacidad de potencia de la planta (como se puede comprobar en la Figure 55. T-s diagram of an AA-CAES and an AA-CAES with an extra high temperature TES. Source: ). Además de poder aumentar la potencia para el mismo volumen de cámara de aire, también se puede conseguir una potencia igual con un volumen menor o con un menor trabajo de los compresores, pudiendo tener así plantas más pequeñas y adaptables a más localizaciones.

La eficiencia de la planta pasaría a calcularse de otra manera, necesitando añadir al denominador el trabajo correspondiente al calor aportado externamente, quedando de la siguiente manera si por ejemplo el calor viene de una fuente eléctrica con pérdidas despreciables:

\[ \eta_{WHTTES} = \frac{w_t}{w_c + w(q_{ext})} = \frac{w_t}{w_c + q_{ext}} \]

El desarrollo de esta mejora puede ser clave en la construcción de un proyecto de este tipo, ya que con ella se podrían reducir considerablemente la inversión inicial y obtener mayores potencias, superando de esta manera el fuerte impedimento que supone la viabilidad económica de los sistemas de AA-CAES.
1.5. Análisis económico temporal del proyecto

A continuación, se muestran de forma resumida la estructura del proyecto y el análisis temporal (diagrama de Gantt) y económico (Presupuesto). Se pueden encontrar más detallados en la sección 9.

**ECONOMIC AND TEMPORAL ANALYSIS**

A la hora de realizar un proyecto de esta índole, es fundamental dividir la labor en etapas bien definidas y estructurar estas etapas por orden de importancia y duración. Es también un factor clave aproximar el presupuesto del proyecto a la hora de saber si finalmente merece la pena realizarlo o no.

Dentro de la planificación temporal, se muestran:

- la Estructura de Descomposición del Proyecto, que muestra la jerarquía de los procesos seguidos y la relación entre ellos.
- El diagrama de Gantt, que muestra la duración de cada una de las fases, además de presentar gráficamente la duración de cada etapa, las relaciones entre ellas, y su jerarquía de realización.

### 1.5.1. Estructura de descomposición del proyecto (EDP)

![Diagrama de Estructura de Descomposición del Proyecto](image.png)

*Figure 18. EDP del proyecto. Fuente: Elaboración propia.*
1.5.2. Diagrama de Gantt

En primer lugar, se muestra a continuación una tabla con el desglose de las actividades llevadas a cabo, además de su relación y duración, en las que aparecen en negrita las tareas principales de manera que se ve claramente su jerarquía.

<table>
<thead>
<tr>
<th>Task number</th>
<th>Task name</th>
<th>Start date</th>
<th>End date</th>
<th>Duration (days)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Documentation and reading reports</td>
<td>01/03/2018</td>
<td>20/04/2018</td>
<td>37</td>
</tr>
<tr>
<td>1.1</td>
<td>Thermodynamics</td>
<td>01/03/2018</td>
<td>08/03/2018</td>
<td>6</td>
</tr>
<tr>
<td>1.2</td>
<td>CAES systems</td>
<td>08/03/2018</td>
<td>29/03/2018</td>
<td>16</td>
</tr>
<tr>
<td>1.3</td>
<td>Thermal Energy Storage</td>
<td>02/04/2018</td>
<td>20/04/2018</td>
<td>15</td>
</tr>
<tr>
<td>2</td>
<td>Programming of excel document</td>
<td>08/03/2018</td>
<td>19/10/2018</td>
<td>162</td>
</tr>
<tr>
<td>2.1</td>
<td>Air chamber equations</td>
<td>08/03/2018</td>
<td>15/06/2018</td>
<td>72</td>
</tr>
<tr>
<td>2.2</td>
<td>Thermal Energy Storage equations</td>
<td>24/09/2018</td>
<td>19/10/2018</td>
<td>20</td>
</tr>
<tr>
<td>3</td>
<td>Air storage chamber modelling</td>
<td>05/03/2018</td>
<td>24/04/2018</td>
<td>37</td>
</tr>
<tr>
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<td>12/03/2018</td>
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</tr>
<tr>
<td>3.2</td>
<td>Equations</td>
<td>12/03/2018</td>
<td>24/04/2018</td>
<td>32</td>
</tr>
<tr>
<td>4</td>
<td>Thermal Energy Storage modelling</td>
<td>17/09/2018</td>
<td>05/10/2018</td>
<td>15</td>
</tr>
<tr>
<td>4.1</td>
<td>Thermodynamic basis</td>
<td>17/09/2018</td>
<td>24/09/2018</td>
<td>6</td>
</tr>
<tr>
<td>4.2</td>
<td>Heat losses</td>
<td>28/09/2018</td>
<td>05/10/2018</td>
<td>6</td>
</tr>
<tr>
<td>5</td>
<td>Results and conclusions</td>
<td>23/10/2018</td>
<td>30/10/2018</td>
<td>6</td>
</tr>
<tr>
<td>6</td>
<td>Report development</td>
<td>09/05/2018</td>
<td>05/11/2018</td>
<td>129</td>
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<td>6.1</td>
<td>Completion of report</td>
<td>09/05/2018</td>
<td>23/10/2018</td>
<td>120</td>
</tr>
<tr>
<td>6.2</td>
<td>Format, bibliography and annexes</td>
<td>23/10/2018</td>
<td>30/10/2018</td>
<td>5</td>
</tr>
<tr>
<td>6.3</td>
<td>Spanish summary</td>
<td>30/10/2018</td>
<td>05/11/2018</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 4. Análisis temporal del proyecto. Fuente: Elaboración propia.

En la siguiente página, se muestra esta información en forma de gráfica gracias al software gratuito Gantt Project.
1.5.3. Presupuesto

1.5.3.1. Coste del personal de trabajo

En primer lugar, se tiene en cuenta el coste del personal. Se ha aproximado el salario de un ingeniero industrial recién graduado a 16 €/h, y el de un docente de la Escuela a 30 €/h, dedicando entre todos los tutores alrededor de 20 horas en total al proyecto.

<table>
<thead>
<tr>
<th>Worker</th>
<th>Salary (€/h)</th>
<th>Time worked (h)</th>
<th>Total cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Recently graduated industrial engineer</td>
<td>16,00</td>
<td>320</td>
<td>5120,00</td>
</tr>
<tr>
<td>Experienced engineers (tutors)</td>
<td>30,00</td>
<td>20</td>
<td>600,00</td>
</tr>
</tbody>
</table>


1.5.3.2. Coste de las instalaciones y licencias

El equipamiento utilizado tiene los siguientes costes. En primer lugar, el portátil utilizado, un Lenovo Ideapad, tiene un coste de 550,00 €, lo cual con una esperanza de vida de 5 años da un coste de amortización de 110,00 €.

Por otro lado, durante la estancia y realización del proyecto en Francia, se puso a disposición del alumno una oficina compartida con otros dos estudiantes. Estimando el alquiler del espacio con un valor de 100 €/mes en total, el precio mensual por alumno sería de 33,30 €. Teniendo en cuenta que la oficina se utilizó por un periodo de 4 meses completos.

<table>
<thead>
<tr>
<th>Item</th>
<th>Total cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laptop Amortization</td>
<td>133,30</td>
</tr>
<tr>
<td>Office rent (4 months)</td>
<td>110,00</td>
</tr>
</tbody>
</table>


1.5.3.3. Presupuesto total del proyecto

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Workers</td>
<td>5.720,00</td>
</tr>
<tr>
<td>Laptop amortisation</td>
<td>110,00</td>
</tr>
<tr>
<td>Rent</td>
<td>133,30</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>5.963,30</strong></td>
</tr>
</tbody>
</table>

2. EXECUTIVE SUMMARY

When analysing the future of the global society and of planet earth, there are numerous difficulties that desperately need an efficient solution for the maintenance of the quality of life that is present nowadays, of which global warming and pollution and the search for new sources of energy to combat the inevitable depletion of fossil fuels are key examples, especially with a population which is rapidly growing and with it are its demands.

Many important organisations, such as the European Union and the United Nations, are already imposing objectives which need to be reached in the future in the energetic and environmental fields with agreements such as the “2030 Energy Strategy” (in the case of the European Union), according to which greenhouse gases emissions have to be greatly cut and renewable energy sources’ importance must grow substantially before 2030. The obvious alternative for the generation of power in a sustainable manner is the use of renewable energy sources, such as wind, solar or hydropower, which are already steadily raising their share of energy supply and will account for at least 27% of the energy supply by 2030.

With this increase needed in contribution comes another challenge implicit in the nature of these renewable sources: the power they supply is wasted if it is not immediately used. In other words, if a wind farm’s turbines are operating at night (when there is very low energy demand), their power is being lost. It is here that energy storage systems begin to gain relevance, and that the Compressed Air Energy Storage method was conceived.

In this project, the study of a process that will tackle both global warming and the energetic challenge is proposed: An Advanced Adiabatic Compressed Air Energy Storage system model.

Compressed Air Energy Storage systems were first developed to be able to store energy from renewable sources, with wind power being especially adaptable to them, which would otherwise be lost if not taken advantage of simultaneously with its generation. However, the traditional adiabatic CAES models still required the burning of fossil fuels, problem which led to the introduction of newer versions of the same concept, such as AA-CAES.

The idea behind Compressed Air Energy Storage is to use the surplus wind power to compress air and store it for it to be turbined when large amounts of energy are demanded by the population, such as peak times. It takes place in three main stages. Firstly, the extra energy from wind farms is used to drive a motor that powers the compressors to increase the air pressured before storing it in an underground air storage chamber. Afterwards, when energy is needed, this air is released from the air storage chamber at high pressures and heated in order to achieve higher work outputs from the last stage, which is the expansion of the air in turbines that will power a generator and provide electricity.
While giving a solution to the storage of energy, the traditional CAES method implies the burning of fossil fuels to heat the air before its expansion, therefore making it disadvantageous for the tackling of pollution and global warming. Advanced Adiabatic CAES introduces a new step in the system: the compressed air, which is at high temperatures as a result of its high pressures, is used to heat a Thermal Energy Storage before its storage, not losing this heat like in CAES. This is then used to reheat the air before its expansion without the need for an external heat source, therefore becoming a totally environmentally friendly solution to the storage of energy from renewable sources.

The main components of the system are: the compressors and turbines (turbomachinery) with which energy is generated with the use of air, the air storage chamber where the high-pressure air is stored until energy is needed and the Thermal Energy Storage, where heat is temporarily reserved between the compression and expansion stages. In this investigation, a thermodynamic AA-CAES model is proposed in which the two main components of the system are taken into consideration: the air storage chamber and the Thermal Energy Storage.

The air storage chamber has a very simple installation process through the utilisation of obsolete underground salt caverns. It is modelled following two possible alternatives, as has been seen in several studies regarding the matter. The first alternative is a storage chamber with a constant volume and temperature. The second one is a more modern concept, in which the air storage chamber is isothermal and isobaric. Its construction and maintenance are costlier and more complicated than the first proposal, and has not yet been realised practically, but gives better results and a higher power capacity.

The Thermal Energy Storage is perhaps the main area of investigation in this concept nowadays, with the search for better materials, heat transfer fluids and the optimisation of its location. In this project, it is modelled as an above ground cylindrical tank composed of a packed bed of rocks through which air can flow and exchange heat, which is popular for its low cost, simplicity, and good functioning. Other heat transfer media such as water or molten metals have better properties but more limited working ranges and therefore are not adequate, whereas recently developed salts and compounds still have unknown results and are very expensive.

Throughout the whole investigation, the analysis and results are given for different alternatives of the compressors and turbines which basically consists of the number of stages chosen for the compression and expansion of the air. The alternatives proposed are one, two or three stages for these processes. The analysis is also based on the number of cycles for which the system has been working and the entry into a permanent regime.

The results obtained show a power capacity which is higher than that of any solar plant and which is comparable to that of the two existing and working traditional CAES plants, located in
Huntorf, Germany and in McIntosh, Alabama, USA, giving a good perspective on the possibility of a future implantation of the system.

Despite all the advantages given by this type of energy storage, there is still to be an existing power plant of this type due to the system’s limitations, which include the extremely high demands on its turbomachinery, the lack of an existing installation to see its results and costs, and especially the economic viability of the project, which is very low primarily due to the very high investment costs of the materials, components and technology needed. This investigation is therefore considered as a first step into the study of the topic and as a founding for the base of future investigations.

Despite this, many organisations grouped into two main investigation programmes (the ADELE-ING and the RICAS2020 projects) especially in Germany, which is Europe’s leading wind power producer, are convinced of the system’s liability and are carrying out different tests and studies to make the model a reality.
3. INTRODUCTION

With the planet’s non-renewable energy sources and fossil fuels reserves rapidly decreasing, it is inevitable for this generations’ engineers to investigate into new ways of meeting the world’s high and continuously growing energy and electricity demands.

The use of renewable energy sources, one of the main alternatives to traditional energy sources, is constantly on the rise and is set to provide 30% of the worlds’ electricity demands by 2030\(^3\). However, renewable energy sources can only provide energy when the weather conditions meet certain specifications, and, especially in the case of solar and wind power, the energy generated by them must be immediately used, otherwise being lost. Due to the constant variations in the energy demand of all sectors throughout a day, a week, or even a season, a very usual situation is that, during favourable climatic conditions, the energy generated by these is higher than that demanded, and during unfavourable conditions, the power demanded is much higher than that supplied. This is the case in times of peak electricity demand, in which the power that is today supplied by conventional sources will in the future have to be provided by an alternative method.

It is in here that Advanced Adiabatic Compressed Air Energy Storage (AA-CAES) becomes useful. This system allows for the storage of energy during low demand-hours to use it in periods of peak necessity, in which the production capacity of the renewable energy source is exceeded.

In this project, the system will be explained, and a thermodynamic model will be proposed for the process with two different types of air storage chambers, one with a constant volume and one with constant pressure. The non-dimensional work and heat produced by the cycle will be calculated for up to three compression and expansion stages, allowing us to calculate the possible power output and roundtrip efficiency of the system as well as a model for the Thermal Energy Storage. Lastly, a possible improvement of the process is analysed, in which the objective will be to maximise the system’s power output in order to be able to provide for a larger part of the population.

\(^3\) [https://www.iea.org/topics/renewables/](https://www.iea.org/topics/renewables/)
Figure 20. Schematic of a Compressed Air Energy Storage Facility. Source: [1]

Before the developing and study of AA-CAES models, that has been a recent point of interest for engineers in the energy sector, the process used was traditional diabatic Compressed Air Energy Storage, developed in the late 1970s. As can be seen in the above figure, the excess power is used to compress air, which is stored underground and used to drive turbines when the power provided by the energy source is not enough to meet the electricity demands of the population.

The advantages of CAES compared to other energy storage methods, such as mechanical, chemical, biological or magnetic storage, are its large capacity and its capability of storing energy during long periods of time, for example inter-seasonally.

In traditional CAES, the stored air is preheated before entering the turbines with the heat coming from the combustion of fossil fuels, therefore releasing CO₂ and other greenhouse gases, going against the purpose of eliminating the necessity of traditional energy sources and against the objective of the use of renewable energy sources.
A key parameter when evaluating CAES models is its round-trip efficiency. The round-trip efficiency gives the energy produced in the turbines during the expansion of the stored air relative to the energy consumed by the compressor before the air storage chamber. It is defined as:

\[ \eta_{RT} = \frac{E_T}{E_C + \eta_F E_F} \]

In which:
- \( E_T \) is the electrical energy generated by the turbine
- \( E_C \) is the electrical energy used by the compressor
- \( E_F \) is the chemical energy supplied by the fuel
- \( \eta_F \) a conversion factor to compare \( E_F \) as an actual electrical energy

The round-trip efficiency of the traditional CAES process is around 40-50\%. The reason for such a low number lies in the fact that all the heat gained by the air due to its increase in pressure during its compression stages is lost to the environment before and during its storage, meaning that a large extra energy input is needed to reheat this air before its expansion via the combustion of fuel. It is aiming to increase this efficiency while simultaneously eliminating the necessity of combustion that new Compressed Air Energy Storage models, such as the Advanced Adiabatic model studied in this project, were developed.

There are currently two operating CAES plants in the world, the Huntorf power plant in Germany and the McIntosh power plant in the United States and, although there is not an operating AA-CAES plant yet, there are several projects for these to be a reality as soon as 2020, such as the ADELE or the RICAS projects, which are explained in the annexes.

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4 Performance assessment of Adiabatic Compressed Air Energy Storage (A CAES) power plants integrated with packed-bed thermocline storage systems
Vittorio Tola, Valentina Meloni, Fabrizio Spadaccini, Giorgio Cau

5 A novel isobaric adiabatic compressed air energy storage (IA-CAES) system on the base of volatile fluid. Long Xiang Chena, Mei Na Xiea, Pan Pan Zhaoc, Feng Xiang Wanga, Peng Hub, Dong Xiang Wangd.
4. OBJECTIVES

As has been mentioned in the introduction, the engineers of the future must face the challenge of adapting the increasing energy demands of the world to a new scenario in which the source of this energy will be very different to that of today. This implies finding and developing new methods and technologies to generate power and store it, if necessary, one of which, seeing that there are already two existing plants and that several projects are being developed, will probably be with Compressed Air Energy Storage.

It is therefore important to introduce the concept behind this generation and storage mechanism, in this case for the Advanced Adiabatic CAES, and its main components, analyse them thermodynamically to be able to determine key results such as their power consumption and generation, efficiency and heat losses implied.

As a result of the concept behind AA-CAES being very modern, especially in the introduction of the Thermal Energy Storage needed, the characteristics and mechanical demands on its components are very high and entail challenges in both the technological and economical aspects, this being the main reason behind the current inexistence of such power plants.

This being said, the main objectives of this investigation are:

- to introduce the concept behind the Compressed Air Energy Storage method as well as the different alternatives being proposed and investigated at the moment with their corresponding advantages and disadvantages.
- to develop a thermodynamic model for the air storage chamber that will allow for the determination of the power generated by the system, for up to 3 compression and expansion stages and according to the number of charging and discharging cycles the of the plant.
- to develop a thermodynamic model for the heat transfer and heat losses in the Thermal Energy Storage, depending again on the number of stages in the compression and expansion process and on the number of functioning cycles of the plant.
- to provide the results obtained by combining the models for both components.
5. COMPRESSED AIR ENERGY STORAGE

5.1. Types

5.1.1. Adiabatic CAES (ACAES)

To improve round-trip efficiency and in an attempt to make CAES totally independent of fossil fuels, and therefore reducing its environmental impact and CO$_2$ emissions, an improvement was made to traditional CAES: the addition of a Thermal Energy Storage (TES) system, which allows heat to be stored during the compression stages to later be used to preheat the air before entering the turbine, removing the need of a combustion chamber and therefore of the burning of fossil fuels. This makes the system totally independent of an external heat source, thus the term “Adiabatic” Compressed Air Energy Storage.

Schematic of the process

![Diagram of A-CAES process](image)

**Figure 21. Schematic of A-CAES process. Source: [5]**

**Advantages:**

- Higher round trip efficiency due to the elimination of the necessity of fuel.

\[
\eta_{RT} = \frac{E_T}{E_C + \eta_F E_F} \quad \Rightarrow \quad \eta_{RT,ACAES} = \frac{E_T}{E_C} \eta_{RT,CAES} = \frac{E_T}{E_C + \eta_F E_F}
\]

- No fuel needs mean no combustion of fossil fuels and therefore no emission of greenhouse gases.
- This model can be applied after the extinction of fossil fuels.

**Disadvantages:**
• Not currently as feasible economically as traditional CAES.
• Thermal Energy Storage technologies still need to be investigated and improved, as well as the thermal storage medium with which the heat is stored (solid bricks, liquids, molten salts, etc.)

To make the Adiabatic CAES concept economically sustainable, it was developed into the Advanced Adiabatic CAES.

5.1.2. Advanced Adiabatic CAES (AA-CAES)

The idea and schematic behind the Advanced Adiabatic method is the same as with Adiabatic CAES, but with improvements in the efficiency of the compressor and turbine systems.

![Schematic of AA-CAES process. Source: European Association for the Storage of Energy.](image)

**Advantages:**

• It makes Adiabatic CAES economically feasible.
• Round trip efficiencies of about 70%\(^6\) are achieved.

**Disadvantages:**

• The main reason behind not reaching higher efficiencies is the exergy losses the process experiences during the depressurising of the stored compressed air\(^7\). The compressed air needs

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\(^7\) Long Xiang Chen, Mei Na Xie, Pan Pan Zhao, Feng Xiang Wang, Peng Hu, Dong Xiang Wang *A novel isobaric adiabatic compressed air energy storage (IA-CAES) system on the base of volatile fluid.*

49
to be depressurised to meet the requirements of the turbine, which are usually an inlet pressure which is lower than that of the stored air.

- To achieve desired energy storage densities, large storage volumes and high operating pressures are needed, resulting in very high costs and difficulties in applying this model currently.

5.1.3. High Temperature Hybrid CAES (HTH-CAES)

A new model that, as well as removing the need of burning fossil fuels for combustion of traditional CAES, is in development with the objective of improving the performance of the Advanced Adiabatic CAES as well as increasing its output power.

It consists of using two stages of heating and two different Thermal Energy Storages, the LTES (Low temperature Thermal Energy Storage), and the HTES (High temperature Energy Storage), in order to achieve higher temperature in the turbine inlets and therefore a higher power output. This process can convert electricity into heat through Joule resistive heating, with the electricity coming from the wind turbines.

![Schematic of the High Temperature Hybrid AA-CAES process. Source: [3]](image)

**Advantages:**

- Smaller volumes and lower pressures than in AA-CAES (higher energy densities) can be achieved.
- Lower investment costs than for AA-CAES due to lower storage volumes needed.

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8 Sammy Houssainy, Mohammad Janbozorgi, Peggy Ip, Pirouz Kavehpour. *Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system.* (2017)
• Higher efficiency than AA-CAES\textsuperscript{9}.
• The HTES can achieve high temperatures without a heat transfer fluid, eliminating all its associated losses, complexity and limitations\textsuperscript{10}.
• The higher temperatures allow for a higher power capacity without increasing the air pressure of the air storage chamber.

Disadvantages:
• The high efficiency is limited by the need of a specific value for the inlet pressure, temperature and mass flow, which therefore have to be regulated.
• In order to maintain the High Temperature Thermal Energy Storage at the required temperature levels, large amounts of energy are needed. This energy comes from an external source, such as by the Joule effect with an electrical resistance and may imply the utilisation of non-renewable energy sources thus reducing the process’ efficiency.

5.1.4. Isobaric A-CAES

This system in development aims to improve the performance of the CAES process.

It is based on the storage of a volatile fluid, usually CO\textsubscript{2} because of its properties and its high saturation pressure, in equilibrium with the air in the storage vessel, separated by a piston. The heat recovered after the compression stage of the process is used to keep the CO\textsubscript{2} at the required pressure for the latter discharge process to be stable. It is called isobaric because the pressures in the storage vessels are the same as the pressure needed to work the turbines that generate the electricity.

The compression and expansion processes are very similar to those of normal AA-CAES, but with a constant pressure air storage chamber (which will also be studied in this project), rather than the constant volume variation that can be found in the existing plants in the Huntorf and McIntosh plants.

The following images show the charging and discharging process of the Thermal Energy Storage system in this type of CAES facilities.

\textsuperscript{9} Sammy Houssainy, Mohammad Janbozorgi, Peggy Ip, Pirouz Kavehpour, *Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system*. (2017)

\textsuperscript{10} Idem.
Advantages:

- Round trip efficiency is up to 6% higher, reaching values of 72.60%.
- Exergy losses are also lower as the stored air does not need to be expanded before reaching the turbine (the air storage pressure and the turbine inlet pressure are equal)\(^\text{11}\), where as in traditional CAES the air pressure needs to be throttled before entering the turbines.

• The effective air storage density increases by 42.3%\textsuperscript{12}, allowing for smaller chamber volumes and therefore less construction and investment costs.

**Disadvantages:**

• Discharge time increases considerably compared to traditional diabatic CAES\textsuperscript{13}.

• Performance is very sensible to changes in ambient temperature, as these affect the saturation pressure of the volatile fluid and therefore its stability and that of the air storage chamber.

• Extra pumps and large amounts of pumping energy are required to keep the equilibrium in the air storage chamber between the volatile fluid and the water on the other side of the piston.

**5.1.5. Isothermal Compressed Air Energy Storage (I-CAES)**

I-CAES is an alternative to AA-CAES being studied and developed with the aim of replacing the traditional method of compressing-cooling-heating-expanding in several stages implicit in traditional as well as in AA-CAES with a single stage of isothermal compression and expansion. In other words, the process is very similar to AA-CAES, with sole difference being the use of isothermal compressors and turbines instead of adiabatic components.

**Advantages:**

• Compression can be achieved in one stage. The traditional process of compressing-cooling-heating-expanding in several stages is not needed, meaning fewer material costs and a smaller area needed for the installation.

• Removes the need of the storage of compression heat by a secondary means\textsuperscript{14}.

• The work input needed to achieve the desired compression rate isothermally is lower than in any other case, as is explained with the help of the following P-v diagram.

---

\textsuperscript{12} Idem.


\textsuperscript{14} Idem.
The work needed to compress the air from $P_1$ to $P_2$ in a P-v diagram is the area under the curve. As we can see, the isothermal compression (with a polytropic factor of 1) is the one with the smaller area under it, indicating that an isothermal compression would need less work input, allowing the system to have a higher round trip efficiency.

**Disadvantages**

- It is technologically challenging to achieve the isothermal conditions due to the high heat transfer rates needed\textsuperscript{15}, which imply the constant and simultaneous removal/addition of heat during the compression/expansion process, for which large heat transfer surface areas are needed.
- The utilisation of liquid water which is highly advantageous in the heat exchangers is technically very difficult due to the high temperatures and pressures reached.

6. ADVANCED ADIABATIC COMPRESSED AIR ENERGY STORAGE (AA-CAES)

6.1. System Components

6.1.1. Compressors

The surplus in the generation of a renewable energy source is used to power the compressors in the system. The compressors are used to increase the pressure of the air in the AA-CAES system before its storage. The utilisation of air as the gas that enters the process (as opposed to water vapour) allows for the supposition that the fluid follows the perfect gas equation of state\(^\text{16}\).

6.1.1.1. Types of compressors

1. Axial compressor

   The fluid enters and exits the compressor parallel to its axis. The fluids’ speed is first increased by the rotor and then diffused in the stator, resulting in an increase in its pressure. They are used for large fluid flow rates\(^\text{17}\) and especially in gas turbine applications\(^\text{18}\), such as that in this model. The efficiency of the gas turbine placed after the compressor increases with the compressor’s pressure ratio.

2. Radial compressor

   The fluid enters the compressor parallel to its axis and exits in a perpendicular direction. With a higher reliability and large tolerance of fluctuations in the mass flow of the working fluid, they are the most common selection in industries such as the petrochemical sector\(^\text{19}\).

According to the properties and uses of both options, the AA-CAES system will use axial compressors due to their adaptability to the high compression ratios and mass flows needed for the process.

6.1.2. Air Storage Chamber

After the compression experienced by the air, it must be stored for a variable period of time in a storage chamber, before its use to power the turbines. This storage can either be done in natural


\(^{17}\) Helsingen, E. (2015). *Adiabatic compressed air energy storage*. Norwegian University of Science and Technology

reservoirs or in artificial air storage tanks. In this project, two possibilities for this air storage chamber are proposed and studied: an air storage chamber with constant air pressure and temperature and one with constant volume and air temperature.

The most common location for these types of chambers in existing CAES studies are underground existing salt caverns by cause of their relatively low investment cost-- the investment in material, transport and installation is significantly lower than in other alternatives-- and, their simple creation and preparation procedure, and their great properties and adaption to the system: they are waterproof, impermeable, non-porous\(^{20}\) and can easily withstand the high pressures and the constant pressure variations that the process developed requires considering its high elasto-plasticity\(^{21}\).

The preparation technique for these potential air deposits, called solution mining, is as follows: water is injected into the salt caverns to dissolve the salt stored in them. The solution of salt and water is then extracted, leaving behind a large empty volume that is adapted for the storage of air at high pressures.

![Figure 27. Schematic of the air storage chamber in a former salt cavern used in the Compressed Air Energy Storage plant in Huntorf, Germany. Source: [18]](https://www.storengy.com/en/expertise/type-of-storage/storage-in-salt-caverns.html)

The following map shows potential locations for the implementation of CAES by taking advantage of existing and exploitable underground salt caverns. The areas in blue and green are the existing the salt deposits in Europe. As can be seen, there are many potential locations for these types of caverns and thus for the construction of a CAES plant. Large areas are available in the north of

---


Europe, for example in Germany, where the Huntorf power plant, CAES project has already been implemented using these natural salt caverns.

In Spain, possible locations could be throughout the entirety of its east coast and along the French border.

![Figure 28. Location of salt deposits in Europe for the possible implantation of an AA-CAES system. Source: [29]](image)

The location of an AA-CAES plant is highly influenced by the location of potential air storage chambers, but other factors must be considered. Because of the great adaptability that this system has with wind farms, by allowing to level the electricity load produced by these, it is also important to study regions with high quality wind to locate the plants nearby and minimise the losses associated with the transport of the electricity. The following image shows areas with high quality winds together with areas that are potential sites for CAES systems.
In this map, together with the potential air storage chamber locations that can be seen in the blue areas, the red circles show regions where the wind quality is high. Therefore, the regions inside the red circles would be the most appropriate for the installation of both wind farms and AA-CAES power plants nearby.

The areas in Spain with advantageous locations are in the South-East coast and in near the Spanish-French border.
6.1.3. Thermal Energy Storage

6.1.3.1. Charging cycle TES schematic

As can be seen, the Thermal Energy Storage is initially at a temperature of $T_{TES,i}$, which for the first charging/discharging cycle is equal to ambient temperature. The heat transfer fluid, if the Thermal Storage material is passive, or the actual Thermal Storage fluid in the case of an active storage approach (see Thermal Energy Storage in 4.1.3.4), flows through a heat exchanger in which it is heated by the compressed air, reaching a final temperature of $T_{TES,f}$, which is the final TES temperature neglecting the heat losses in pipes.

The mass flow rate of the heat transfer fluid leaving and entering the TES is dependent on the number of compression stages. As this number increases (form 1 to a maximum of 3), the heat exchangers get arranged in parallel and the same mass flow rate crosses each one.
4.1.3.2. Discharging cycle TES schematic

Initially neglecting heat losses in the TES over storage time, the TES will remain at its final charging temperature, $T_{TES,f}$, until the start of the discharging. After the discharging process, in which the storage medium transfers its heat to the air by means of a heat exchanger, the TES is at a temperature of $T_{TES,i}$, which is equal to the initial TES temperature of the following charging cycle.

Due to the stratification inside the TES tank, the flow direction of the heat transfer fluid during the charging and the discharging operations must be opposite in order to maximise the efficiency of the heat transfer:\footnote{Jung-Wook Park, Dohyun Park, Dong-Woo Ryu, Byung-Hee Choi, Eui-Seob Park. (2014). Analysis on heat transfer and heat loss characteristics of rock cavern thermal energy storage.}: during the heat charging, the heat transfer fluid flows from the top to the bottom of the tank. As a result of the density of fluids decreasing with temperature, the hotter layers of the TES
will be at the top and thus the heat discharging process is carried out by a bottom to top flow of the heat transfer fluid.

4.1.3.3. Utility

For the Advanced Adiabatic Compressed Air Energy Storage to be efficient in reducing the combustion of fossil fuels thus reducing pollution and being an environmentally friendly solution to the otherwise wasting of renewable energy sources, the thermal energy storage unit is vital.

The Thermal Energy Storage unit’s role is to store heat from the rise in temperature due to the compression of air. This heat would otherwise be lost, requiring to be generated again before the expansion stages with the help of an external heat source, as seen with the fuel combustion in the initial CAES models. With the TES unit, the heat stored during the compression stages is later input into the air before its entry into the expansion stages.

By heating the air before each expansion stage, its temperature is increased achieving a higher temperature difference in each expansion stage. In Advanced Adiabatic CAES, both the compressors and the turbines are modelled as adiabatic. This being considered, an increase in air temperature means a higher work output, and therefore a higher power capacity for the process

\[
\begin{align*}
q &= 0 \Rightarrow h &= w + c_p \Delta T \\
\Rightarrow h &= c_p \Delta T \Rightarrow \Delta T \Rightarrow w_t \Rightarrow P
\end{align*}
\]

4.1.3.4. Heat storage methods

There are three methods of storing heat in these systems:

1. Sensible Thermal Storage

The heat is stored by increasing the temperature of a liquid or solid storage material. According to

\[
Q = mc_p \Delta T
\]

the amount of heat stored depends, for a constant mass, on the heat capacity of the storage medium and the temperature change.

The TES system is composed of

- A heat storage medium or material
- A heat transfer fluid

being the heat transfer fluid also the heat storage material in active storage mediums (as seen section c.i. of the next paragraph).

There are several ways to classify sensible thermal storage systems.

a. A first classification can be done according to the period for which the thermal energy needs to be stored. They can be divided into
i. Short term storage, which is considered as a diurnal storage. This means the heat is stored for a maximum period of some hours. The most common materials used in this case are: water, pebbles or oil.

ii. Long term storage. The heat is stored for seasonal periods. In this case, the most common methods are underground bore-holes, aquifers, caverns, or rock-bed systems.

b. A classification can also be done according to the temperature of the TES.
   i. Cool storage. Used for the cooling and air conditioning of a building.
   ii. Low temperature energy storage (T<100ºC)
   iii. Medium temperature energy storage (100ºC<T<250ºC)
   iv. High temperature energy storage (T>250ºC)

c. Lastly, TES systems can be classified based on the function of the heat storage medium.
   i. Active storage. The heat storage material does the function of heat collection, storage and released. In other words, the heat storage material is also the heat transfer fluid.
   ii. Passive storage. The heat storage medium only stores the heat, needing an additional material to collect and release the heat into the system.

2. Latent Thermal Storage
The heat is stored or released by using a phase change in a certain substance, either from liquid to solid or solid to liquid.

3. Thermochemical Thermal Storage
Heat is stored and released by using an endothermic or exothermic chemical reaction respectively.

Heat storage media
If we express the mass as the product of the volume of the TES and the density of the material used, we arrive at

\[ Q = V \rho c_p \Delta T \]  \hspace{1cm} (11)

From this formula we can see that it is important for the heat storage material to have a high density in order to reduce the volume of the Thermal Energy Storage tank and therefore the dimensions and costs of the system.

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24 ***
Furthermore, it is possible to classify the Thermal Energy Storage regarding the material employed and its state of matter:

i) **Liquid sensible heat storage materials.** (can be used in active storage-both to store and transport the heat)

<table>
<thead>
<tr>
<th>Medium</th>
<th>Density, ρ (kg/m³)</th>
<th>Viscosity (average-at the medium temperature), μ (mPa s)</th>
<th>Specific heat capacity (J/kg K)</th>
<th>Heat capacity (*10⁶ J/m³ K)</th>
<th>Minimum temperature (ºC)</th>
<th>Maximum temperature (ºC)</th>
<th>Cost (USD/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>1000</td>
<td>0.5465</td>
<td>4190</td>
<td>4.190</td>
<td>0</td>
<td>100</td>
<td>0.01</td>
</tr>
<tr>
<td>HITEC molten salt</td>
<td>1680</td>
<td>2.5[25]</td>
<td>1560</td>
<td>2.621</td>
<td>142</td>
<td>454[26]</td>
<td>0.93</td>
</tr>
<tr>
<td>Water-ethylene glycol 50/50</td>
<td>1050</td>
<td>2.8</td>
<td>3412</td>
<td>3.644</td>
<td>-36.8</td>
<td>107.2[27]</td>
<td>1.00</td>
</tr>
<tr>
<td>Lithium</td>
<td>510</td>
<td>0.34</td>
<td>4190</td>
<td>2.137</td>
<td>180</td>
<td>1342</td>
<td>11.82[28]</td>
</tr>
<tr>
<td>Sodium</td>
<td>960</td>
<td>0.21</td>
<td>1300</td>
<td>1.248</td>
<td>98</td>
<td>883</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 8. Properties of different liquid sensible heat storage media. Source: [9]

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25 Coastal Chemical Co, LLC. HITEC Heat Transfer Salt
ii) Solid thermal storage materials (passive storage-can only be used as storage).

In the case of the AA-CAES model proposed, it is possible that no heat transfer fluid is needed, as the air can go directly into the TES and not through an exchanger, or that the heat transfer fluid used is gaseous, and can enter a solid thermal energy storage medium and heat it up. The study of solid storage media is therefore also important. The following table shows the most common solid TES media.

<table>
<thead>
<tr>
<th>Medium</th>
<th>Density, $\rho$ (kg/m(^3))</th>
<th>Specific heat capacity (J/kg K)</th>
<th>Heat capacity ($*10^6$ J/m(^3) K)</th>
<th>Maximum temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Granite</td>
<td>2750</td>
<td>892</td>
<td>2.453</td>
<td>1215</td>
</tr>
<tr>
<td>Sandstone</td>
<td>2200</td>
<td>710</td>
<td>1.562</td>
<td>1500(^{29})</td>
</tr>
<tr>
<td>Limestone</td>
<td>2500</td>
<td>900</td>
<td>2.250</td>
<td>825</td>
</tr>
<tr>
<td>Gravelly earth</td>
<td>2050</td>
<td>1840</td>
<td>3.772</td>
<td>-</td>
</tr>
<tr>
<td>Cast iron</td>
<td>7900</td>
<td>837</td>
<td>6.612</td>
<td>400</td>
</tr>
<tr>
<td>Magnesia brick</td>
<td>3000</td>
<td>1130</td>
<td>3.390</td>
<td>1200</td>
</tr>
</tbody>
</table>

Table 9. Properties of different solid sensible heat storage media. Sources: [9]

Other important properties that the thermal energy storage medium should have are:

- A wide range of temperatures in which the medium is in liquid form.
- Low viscosity in order to reduce the friction in the tubes transporting the fluid
- If possible, low production costs in order to make the method economically feasible.

### 6.1.4. Turbines

Throughout the whole process, the working fluid for the compressors and turbines is air. This implies the utilisation of gas turbines. As with compressors, they can be classified into 2 categories\(^{30}\) according to the direction of the flow of the fluid in the entry and exit of the turbines.

1. Radial inflow turbines

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The flow enters the turbines in their radial direction. A variation of these, called Francis turbines, are the most common water turbines used for electrical power production in the present. They can’t be applied to the model studied as they function with water as the working fluid.

2. Axial flow turbines

The fluid enters and exits the turbine parallel to its axis. For the use of compressible fluids, such as air, they are the most employed option. This variation is more efficient than radial inflow turbines. Frequently, the gas enters these turbines at very high temperatures, as is the case in the model studied, making them require the addition of several cooling schemes.

6.1.5. Heat exchangers

For the AA-CAES system to be feasible thermodynamically, heat from the compressed air must be absorbed and stored temporarily, being released in peak demands of electricity, in which the supply from the energy source, for example a wind farm, is not enough to satisfy the populations’ needs. This is achieved with the implementation of heat exchangers.

The heat exchangers allow the surplus heat from high pressure air, otherwise lost in other CAES systems, to be transferred to the heat transfer fluid, and to transfer this heat back to the air without the use of an additional external heat source.

6.1.5.1. Types

Heat exchanger can be classified taking into consideration different criteria.

1. According to the flow of the fluids.
   a. Parallel currents. The hot fluid (heat releaser) and the cold fluid (heat absorber) flow in the parallel directions. They can be co-current, in which both fluids travel in the same direction, or counter-current, in which the fluids move in opposite directions.
   b. Crossed currents. The hot and cold fluids’ flow are in different directions. They can also be co-current, if both enter the exchanger from the same end, and counter-current if each fluid enters the exchanger through its opposite ends.

2. According to the geometrical configuration of the exchanger.
   a. Concentric tubes heat exchanger.
   b. Plate heat exchanger.
   c. Shell and tube exchanger.

---

32 Idem.
d. Spiral exchanger.

With this classification, the heat exchanger choice is made according to the nature of the fluids circulating through it, their thermodynamic properties, especially the pressure and the temperature, and the size limitations of the installation\textsuperscript{34}.

3. According to the nature of the fluids used.
   a. Without change of phase:
      i. Gas-gas, such as air or smoke.
      ii. Liquid-gas.
      iii. Liquid-liquid, the most common being water, oil or refrigerant fluids.
   b. With change of phase:
      i. Condenser.
      ii. Evaporator, by the production of overheated water vapour.
      iii. Vapour generator\textsuperscript{35}, by the production of saturated vapour.

\textsuperscript{34} Marín, J.M., Guillén, S (2013). Diseño y cálculo de intercambiadores de calor monofásicos. Paraninfo.
6.2. Process schematics

6.2.1. General process schematic

![General process schematic](image)

*Figure 32.* Process schematic of an AA-CAES plant (similar to the process used in this study, with the only difference in the source of the cooling medium). Source: [6]

6.2.2. Charging cycle schematic

![Charging cycle schematic](image)

*Figure 33.* Diagram for the charging process of an AA-CAES with a 3-stage compression process. For 1 and 2 stages, the diagram is the same but with only 1 or 2 compressors respectively. Source: Own elaboration.
During this process, surplus energy from the renewable energy source is used to compress air through a certain number of compressors. This air is then stored, previously having removed and stored surplus heat from it for later use during the discharge phase.

The compressors are number 1 to 3 depending on the number of compression studies, which will be studied later on. The inlet temperatures into the compressors will be calculated using the heat exchangers, and the outlet temperatures will be determined using Poisson’s Law. The work done by the compressors can therefore be calculated.

6.2.3. Discharging cycle schematic

During this process, which takes place when energy is needed, heat is removed from the TES system to preheat the stored air which is then expanded in a turbine and released into the atmosphere. The air enters the turbines at a temperature which is calculated using the heat exchanger efficiencies and the temperatures of the heat transfer fluid. The outlet temperature of the turbines is calculated using Poisson’s Law, and once both temperatures are known, the work produced in the turbines can be determined.
6.3. Thermodynamic modelling of the air storage chamber

To start off with the modelling of the process, the air storage chamber will be determined. This air storage chamber consists of a previously full and obsolete underground salt cavern that has been emptied and prepared to store and release air and is connected to the rest of the system through pipes. In fact, in both existing CAES power plants worldwide, Huntorf and McIntosh, the chamber used was originally a salt cavern.

The air storage chamber must be able to withstand constant and large pressure variations and must resist against high pressures of up to almost 70 bar. For example, in the existing power plant located in Huntorf, Germany, that has been operating since 1978, the minimum and maximum air storage chamber pressures are 46 and 66 bar respectively\(^{36}\). These high pressures levels are necessary in order to reach the power output required to meet the demands of electricity, which increases with an increase in maximum storage pressure or in the difference between maximum and minimum pressures, as will be seen in the equations further on.

The main challenges faced currently in the development of the storage chamber in these sorts of plants are the constant high-pressure levels suffered by its walls, the structural stability and liability, important and difficult to achieve in underground locations surrounded completely by enormous amounts of rock, and the objective of a minimum lifetime of 25 years\(^{37}\).

6.3.1. System conditions and hypotheses

Firstly, the ambient temperature and pressure are defined. The ambient temperature is set to be \(T_0 = 293 \, K\), whereas the ambient pressure is \(p_0 = 0.1 \, MPa\). In order to simplify future equations and operations, a new constant \(a\) is defined as \(a = \frac{y-1}{y}\)

**Compressors**

For the compression stages, the following conditions are set:

The initial temperature of the air entering the first compressor, \(T_1\), is equal to ambient temperature \(T_0\), as it is air taken directly from the environment.

\[ T_{c1} = T_0 = 293K \]

---


The temperature of the compressed air leaving the compressor is calculated according the Poisson Law, which is as follows

\[ T_{ci}^f = T_{ci} \beta^N \]  \hspace{1cm} (12)

where N is the number of stages. This will assure minimum total compression work, as is demonstrated below:

The work input needed in each compressor is

\[ w_{ci} = c_p T_{ci} (\beta_i^a - 1) \]

The total work is the sum of the work in each compressor is

\[ w_c = \sum_{i=0}^{N} w_{ci} = c_p T_1 (\beta_1^a + \cdots + \beta_N^a - n) \]

This value will be minimum when the sum of all the partial pressure ratios is minimum. Because the product of all partial pressure ratios is constant and equal to \( \beta^a \), in which \( \beta \) is defined as the pressure ratio between the final and initial pressures in the air storage chamber, the sum of these terms will be minimum when all of them are equivalent. Therefore \( \beta_1 = \cdots = \beta_N = \beta_1^N \). This proves that the relation between the entering and leaving compression temperatures is that established in equation (12)).

The temperature of the air input into the chamber is, in the equations, numbered as

\[ T_{in,sc} = T_{N+1} \] \hspace{1cm} (13)

**Air storage chamber.**

For the air storage chamber, the conditions set are:

As in both models the air chamber is isothermal, the temperature of the air in it is constant and equal to ambient temperature, \( T_{sc} = T_0 \).

In the air storage chamber, with air considered as an ideal gas, according to the equation of state for ideal gases for a pressure differential, in the case of constant volume, the mass differential is

\[ dm = \frac{V}{RT_0} dp \] \hspace{1cm} (14)

The same rationale can be applied to for the case of a volume differential in the case of a constant pressure air storage chamber.

\[ dm = \frac{p}{RT_0} dV \] \hspace{1cm} (15)
The pressure ratio in the air storage chamber in a certain moment is

$$\beta_i = \frac{p_i}{p_0}$$

The air pressure after the last compression stage is equal to the pressure inside the air chamber at any time\(^{38}\).

**Heat exchangers**

Firstly, in all heat exchangers, the exchanger efficiency, \(\varepsilon\), is considered to be equal to 0.7.

With the objective of simplifying the model and calculations, the heat capacity of the working fluid, air and of the heat transfer fluid, between which the heat exchange takes place, is equal

$$\left(m c_p\right)_{WF} = \left(m c_p\right)_{HTF} = c_p \int dm$$

Substituting the expression for \(dm\) seen in (6) into (7), the heat capacity of both the working fluid and the heat transfer fluid is

$$\left(m c_p\right)_{HTF} = \frac{c_p V}{R T_0} \int_{p_1}^{p_2} \frac{dp}{p_0} = \frac{p_0 V}{\alpha T_0} (\beta_2 - \beta_1)$$

The heat exchangers are set in a parallel distribution, separating the mass flow of the fluid entering the heat exchangers into various lanes. Therefore, depending on the number of compression/expansion stages, which determines the number of heat exchangers in the system, the total heat capacity of the heat transfer fluid necessary for the process is this value multiplied by the number of stages.

**Turbines**

In the expansion process, the heat exchangers are before each turbine (instead of after as in the compressors) in order the heat the air right before its entry. Therefore, the entry temperature into the first turbine, \(T_{t1}\), is determined by the heat exchanger efficiency formula, considering that the air leaves the air storage chamber at a temperature of \(T_0\).

The temperature of the air leaving each turbine stage, according to the total number of stages \(N\) is calculated via the Poisson Law, as in the compressors.

$$T_{ti}^f = T_{ti} \beta_i^{-\frac{\alpha}{N}}$$

---


6.3.1.1. Sign convention

For the following equations to make sense, it is necessary to state the sign convention used for the heat and work entering and leaving the system. Work and heat will be considered positive when they are entering the system, and negative when leaving.

\[ Q, W > 0 \quad \text{System} \quad Q, W < 0 \]

6.3.2. Compression and turbine work and stored and released heat equations

6.3.2.1. For the charging cycle (compression stages)

The initial temperature of the air entering compression stage \( i \) for a system with \( N \) total compression stages is calculated using the heat exchanger efficiency for the exchanger situated immediately before that stage:

\[
\varepsilon_{i-1} = \frac{(mc_p)_{\text{air}}(T_{\text{ex, in}} - T_{\text{ex, out}})}{(mc_p)_{\text{min}}(T_{\text{ex, in}} - T_n^i)} = \frac{T_{c,i-1}^a \beta N_i - T_{CI}^i}{T_{c,i-1}^a \beta N_i - T_{TES,n}^i} \Rightarrow
\]

\[ T_{CI}^i = (1 - \varepsilon)T_{c,i-1}^a \beta N_i + \varepsilon T_{TES,n}^i \quad (19) \]

in which \( T_{TES,n}^i \) is the initial temperature of the TES for charge/discharge cycle \( n \). For the \( n = 1 \), this value is equal to ambient temperature, \( T_0 \).

Because in AA-CAES, the compressors are modelled to be adiabatic, \( Q_c = 0 \), and the working fluid is air, by applying the first law of thermodynamics for an open system to each compressor, the total work done by the compressors according to the number of stages \( N \) can be calculated by:

\[ dH = dQ_c + dW_c = dW_c = c_p(T_{c, out} - T_{c, in})dm \Rightarrow \]

\[ dW_c = c_p \left( \beta N - 1 \right) \left( \sum_{i=1}^{N} T_{CI}^i \right) dm \quad (20) \]

---

On the other hand, the heat exchangers that remove the heat from the compressed air and take it to the Thermal Energy Storage have a constant volume and pressure and therefore there is no work input or output, \( W_s = 0 \). By applying the first law of thermodynamics to the exchangers, the total heat absorbed by the heat transfer fluid and stored in the TES, depending on the number of stages, can be calculated:

\[
dH = dQ_s + dW_s = dQ_s = \int c_p \left( T_{s,\text{out}} - T_{s,\text{in}} \right) \text{d}m
\]

\[
dQ_s = c_p \sum_{i=1}^{N} \left( T_{c,i+1} - T_{cil} \right) \beta^\alpha \text{d}m \quad (21)
\]

Once the total heat stored has been calculated, the final temperature of the Thermal Energy Storage system is easily calculated using the heat capacity of the storage medium and its initial temperature for the corresponding cycle, \( n \)

\[
Q_s = Q_{TES} = mc_p(T_{TES}^f - T_{TES,n}) \Rightarrow
\]

\[
T_{TES,n}^f = T_{TES,n}^i + \frac{Q_{TES}}{mc_p} \quad (22)
\]

### 6.3.2.2. For the discharging cycles (expansion stages)

The initial temperature of the air entering expansion stage \( i \) for a system with \( N \) total turbine stages is calculated using the heat exchanger efficiency for that stage

\[
\varepsilon_{i-1} = \frac{(mc_p)_\text{air}(T_{t,in} - T_{t,\text{out}})}{(mc_p)_\text{min}(T_{t,in} - T_{TES,n})} = \frac{T_{t,i-1} \beta^\alpha - T_{ti}}{T_{t,i-1} \beta^\alpha - T_{TES,n}} \Rightarrow
\]

\[
T_{ti} = (1 - \varepsilon)T_{t,i-1} \beta^\alpha + \varepsilon T_{TES,n} \quad (23)
\]

In the case of the turbines, they are also modelled to be adiabatic and have air as the working fluid, \( Q_t = 0 \). The total work produced in the turbine stages can be expressed by applying the first law of thermodynamics for an open system to each turbine, similarly to with the compressors.

\[
dH = dQ_t + dW_t = dW_t = \int mc_p \left( T_{t,o} - T_{t,in} \right) = -\int mc_p \sum_{i=1}^{N} T_{ti} \left( 1 - \beta^\alpha \right) \text{d}m \Rightarrow
\]

\[\text{Yuan Zhang, Ke Yang, Xuemei Li, Jianzhong Xu. (2013). The thermodynamic effect of air storage chamber model on Advanced Adiabatic Compressed Air Energy Storage System.}\]
The heat exchangers are also used to heat the air before its entry into each turbine stage, so that the maximum quantity of work can be produced with its expansion. As said before, no work is done or produced by the heat exchangers, \( W_r = 0 \), as they have a constant pressure and volume. The heat released from the Thermal Energy Storage can be calculated using a similar development to that in the heat stored.

\[
dH = dQ_r + dW_r = dQ_s = c_p(T_{r,\text{out}} - T_{r,\text{in}})dm = -c_p \left( T_{t1} - T_0 + \sum_{i=2}^{N} \left( T_i - T_{i-1} \beta^{-\frac{a}{N}} \right) \right) = -c_p \left( T_{t1} - T_0 + \sum_{i=2}^{N} \left( (1 - \varepsilon)T_{i-1} \beta^{-\frac{a}{N}} + \varepsilon T_{\text{TES},n} - T_{i-1} \beta^{-\frac{a}{N}} \right) \right) dm
\]

\[
dQ_s = -c_p \left( T_{t1} - T_0 + \varepsilon \sum_{i=2}^{N} \left( T_{\text{TES}} - T_{i-1} \beta^{-\frac{a}{N}} \right) \right) dm
\]

These equations require integration by substituting the mass differential for equations (14) or (15).

In the expansion, the heat exchangers are situated before each turbine. The temperature entering the first turbine, \( T_{t1} \) after exchanging heat in the first exchanger, we use the following equation

\[
T_{t1} = (1 - \varepsilon)T + \varepsilon T_{\text{TES}}^f = (1 - \varepsilon)T_0 + \varepsilon T_{\text{TES}}^f
\]

where \( T \) is the temperature of air leaving the air storage chamber, which is constant and equal to the temperature inside the chamber \( T_0 \) (the chamber being isothermal in both models studied).

To compare the values for both heat and power, and for the results to be easily changed if the for different values of air storage chamber volume, the equations found will be nondimensionalised by dividing both \( W \) and \( Q \) by \( p_0V \). This can be done because the units of \( p_0V \) are equivalent to those of \( W \) and \( Q \):

\[
[p_0V] = \frac{Nm^2}{m^3} = Nm = Joules
\]
6.3.3. AA-CAES with constant air storage chamber volume and temperature (V, T model)

6.3.3.1. Thermodynamic basis

The general equation for the first law of thermodynamics in an open system is:

\[ \dot{Q} = \frac{dE}{dt} + h_{\text{out}}\dot{m}_{\text{out}} - h_{\text{in}}\dot{m}_{\text{in}} + \dot{W} \]

The specific enthalpy for air inside the chamber is

\[ h = c_p(T_s - T_0) \]

Whereas the specific internal energy can be calculated related to the specific enthalpy \( h \) as follows

\[ \begin{align*}
    u &= h - pv \\
    pv &= rT = (c_p - c_v)T^{41} \Rightarrow u = c_pT - c_pT_0
\end{align*} \]

For the V, T model, we take the temperature of the air storage chamber as constant and equal to ambient temperature, \( T_0 \), and the volume equal to \( V \).

I. Charging process-compressed air enters the air storage chamber

With the hypothesis of a constant volume and temperature and the fact that there is no work exchanged between the air storage chamber and the environment (\( \dot{W} = 0 \)) and that during the charging process \( \dot{m}_{\text{out}} = 0 \), the heat exchanged between the air storage chamber and the environment during the charging process is

\[ d\dot{Q} = (c_pT_0 - c_pT_{\text{in}})dm \]

II. Discharging process-air is extracted from the air storage chamber

In this case, \( \dot{m}_{\text{in}} = 0 \) and there is still no work exchanged with the environment, \( \dot{W} = 0 \). Therefore, the heat exchanged with the environment can also be calculated, in this case through the change in pressure of the inside of the chamber.

\[ \begin{align*}
    T &= T_{\text{out}} = T_0 \text{(isothermal)} \\
    h_{\text{out}} &= c_p(T - T_0) 
\end{align*} \]

\( h_{\text{out}} = 0 \Rightarrow \dot{Q} = -RT_0dm = -Vdp \)

6.3.3.2. Equations according to the number of compression/expansion stages

The equations calculated will be according to the number of stages \( N \), and the number of cycles, \( n \).

---

1. For a 1-stage compression/expansion process (N=1)

The inlet temperature of the air chamber, which is \( T_{in} \), in this case is the same value as \( T_2 \), and is calculated using heat exchanger efficiency for heat exchanger 1

\[
T_2 = (1 - \varepsilon)T_0\beta^a + \varepsilon T_{iES,n}^i
\]  

(a. Compressor work

\[
\begin{align*}
\left\{ \begin{array}{l}
\frac{dW_c}{dm} = c_p \left[ \beta^a - 1 \right] \sum_{i=1}^{N} T_{ci} \frac{dm}{dP} = c_p (\beta^a - 1)T_0 \frac{dm}{dP} \\
\Rightarrow dW_c = \frac{V_c p}{R} \int_{P_1}^{P_2} [\beta^a - 1] dP
\end{array} \right.
\end{align*}
\]

\[
W_c = \frac{V_p_0}{a} \left[ \frac{1}{a+1} (\beta_2^{a+1} - \beta_1^{a+1}) - (\beta_2 - \beta_1) \right]
\]  

(b. Turbine work

\[
\begin{align*}
\left\{ \begin{array}{l}
\frac{dW_t}{dm} = -c_p [1 - \beta^{-a}] T_{t1} \frac{dm}{dP} = -c_p \frac{T_{t1}}{R} \int (1 - \beta^{-a}) dP \\
\Rightarrow W_t = -\frac{V_c p}{R} \int_{P_1}^{P_2} (1 - \beta^{-a}) dP
\end{array} \right.
\end{align*}
\]

\[
W_t = -\frac{V_p_0 T_{t1}}{a T_0} \left[ \beta_2 - \beta_1 - \frac{1}{1-a} (\beta_2^{-a+1} - \beta_1^{-a+1}) \right]
\]  

(c. Heat stored

\[
\begin{align*}
\left\{ \begin{array}{l}
\frac{dQ_s}{dm} = dmc_p (T_{c2} - T_0\beta^a) \\
\Rightarrow dQ_s = \frac{V}{RT_0} \int_{P_1}^{P_2} (1 - \varepsilon)\beta^a + \varepsilon T_{iES,n}^i - \beta^a \right) dP
\end{array} \right.
\end{align*}
\]

\[
Q_s = \frac{V_p_0 \varepsilon}{a} \left( T_{iES,n}^i \right) (\beta_2 - \beta_1) - \frac{1}{a+1} (\beta_2^{a+1} - \beta_1^{a+1})
\]  

d. Heat released

\[
\left\{ \begin{array}{l}
\frac{dQ_r}{dm} = -dmc_p (T_{t1} - T_0) \\
\Rightarrow Q_r = -\frac{V}{aT_0} \int_{P_1}^{P_2} [T_{t1} - T_0] dP
\end{array} \right.
\]

\[
Q_r = -\frac{V_p_0}{a} \left( \frac{T_{t1}}{T_0} - 1 \right) (\beta_2 - \beta_1)
\]
2. For a 2-stage compression/expansion process (N=2)

The inlet temperature of the air chamber - Tin-is in this case is the same value as T₃, and is calculated using heat exchanger efficiency for heat exchanger 2

\[
\varepsilon_2 = \frac{(mc_p)_{air}(T_{c,in} - T_{c,out})}{(mc_p)_{min}(T_{c,in} - T_{TES}^{i})} = \frac{T_2\beta_a - T_3}{T_2^i\beta_a - T_{TES}^i} \Rightarrow
\]

\[
T_{c3} = (1 - \varepsilon)T_{c2}\beta_a^a + \varepsilon T_{TES}^i = (1 - \varepsilon)^2T_0\beta_a + \varepsilon(1 - \varepsilon)T_{TES}^i\beta_a + \varepsilon T_{TES,n}^i \tag{32}
\]

a. Compressor work

\[
\left\{ \begin{aligned}
d\mathcal{W}_c &= c_p \left[ \beta_a^a - 1 \right] \sum_{i=1}^{2} T_c dm = c_p \left[ \beta_a^a - 1 \right] \left( 1 + (1 - \varepsilon)\beta_a^a + \varepsilon T_{TES,n}^i \right) dm \\
&= \int d\mathcal{W}_c = \\
&= \frac{V_{cp}}{R} \int_{p_1}^{p_2} \left( (1 - \varepsilon)\beta_a^a + \varepsilon \left( \frac{T_{TES,n}^{i}}{T_0} + 1 \right) \beta_a^a - \left( 1 + \varepsilon \frac{T_{TES,n}^{i}}{T_0} \right) \right) dp
\end{aligned} \right.
\]

\[
\mathcal{W}_c = \frac{V_{p_0}}{a} \left( \frac{1 - \varepsilon}{a+1} \left( \beta_2^{a+1} - \beta_1^{a+1} \right) + \frac{\varepsilon}{a+1} \left( \beta_2^{a+1} - \beta_1^{a+1} \right) - \left( 1 + \varepsilon \frac{T_{TES,n}^{i}}{T_0} \right) (\beta_2 - \beta_1) \right) \tag{33}
\]

b. Turbine work

\[
\left\{ \begin{aligned}
d\mathcal{W}_t &= -c_p \left( T_{t1} - T_{t1}\beta_a^{-a} + T_{t2} - T_{t2}\beta^{-a} \right) dm \\
&= \frac{V_{p_0}}{aT_0} \int_{p_1}^{p_2} \left( T_{t1} - T_{t1}\beta_a^{-a} + (1 - \varepsilon)T_{t1}\beta_a^{-a} + \varepsilon T_{TES}^f - (1 - \varepsilon)T_{t1}\beta_a^{-a} - \varepsilon T_{TES}^f \right) dp
\end{aligned} \right.
\]

\[
\mathcal{W}_t = -\frac{V_{p_0}}{aT_0} \left( T_{t1} + \varepsilon T_{TES}^f \right) (\beta_2 - \beta_1) - \frac{\varepsilon}{1-a} \left( \beta_2^{1-a} - \beta_1^{1-a} \right) - \frac{(1-\varepsilon)}{1-a} T_{t1} (\beta_2^{1-a} - \beta_1^{1-a}) \tag{34}
\]
c. Heat stored
\[
dQ_s = c_p \left( T_{c2} - T_{c1\beta^2} + T_{c3} - T_{c2\beta^2} \right) dm
\]
\[
dm = \frac{v}{RT_0} dp
\]
\[
\Rightarrow Q_s = \frac{Vf}{aT_0} \int_{P_1}^{P_2} \left( \epsilon^2 - \epsilon \right) \beta^a - \epsilon \left( \frac{\tau^i_{T_{ES,n}}}{r_0} + 1 \right) \beta^2 + 2\epsilon \frac{\tau^i_{T_{ES,n}}}{r_0} \right) dp \Rightarrow
\]
\[
Q_s = \frac{Vf}{a} \left( \frac{\epsilon(\epsilon-1)}{a+1} \right) \left( \beta_2^{a+1} - \beta_1^{a+1} \right) - \left( \frac{\epsilon}{\frac{\tau^i_{T_{ES,n}}+1}{a}} \right) \left( \beta_2^{a+1} - \beta_1^{a+1} \right) + 2\epsilon \frac{\tau^i_{T_{ES,n}}}{r_0} \left( \beta_2 - \beta_1 \right)
\]

(35)
d. Heat released
\[
dQ_r = -c_p \left( T_{t1} - T_0 + T_{t2} - T_{t1\beta^2} \right) dm
\]
\[
dm = \frac{v}{RT_0} dp
\]
\[
\Rightarrow Q_r = -\frac{Vf}{aT_0} \int_{P_1}^{P_2} \left( T_{t1} - T_0 + (1 - \epsilon)T_{t1\beta^2} \right) + \epsilon \frac{T^f_{T_{ES}}}{T_{t1\beta^2} \epsilon} \right) dp \Rightarrow
\]
\[
Q_r = -\frac{Vf}{aT_0} \left( T_{t1} - T_0 + \epsilon T^f_{T_{ES}} \right) \left( \beta_2 - \beta_1 \right) - \frac{\epsilon T^f_{T_{ES}}}{1 - \frac{a}{2}} \left( \beta_2^{1-a} - \beta_1^{1-a} \right)
\]

(36)

3. For a 3-stage compression/expansion process (N=3)
The inlet temperature of the air chamber- \( T_{in} \)-is in this case the same value as \( T_4 \), and is calculated using heat exchanger efficiency for heat exchanger 3

\[
\epsilon_3 = \frac{(mc_p)_{air}(T_{c,in} - T_{c,out})}{(mc_p)_{min}(T_{c,in} - T^i_{T_{ES,n}})} = \frac{T_{c3}\beta^a - T_{c4}}{T_{c3}\beta^a - T^i_{T_{ES,n}}}
\]
\[
T_{c4} = (1 - \epsilon)^3 T_0 \beta^a + \epsilon(1 - \epsilon)^2 T^i_{T_{ES,n}} \beta^2 + \epsilon(1 - \epsilon)T^i_{T_{ES,n}} \beta^3 + \epsilon T^i_{T_{ES,n}}
\]

(37)
a. Compressor work
\[
dW_c = c_p \left( \frac{\beta^a}{2} \right) \Sigma_{i=1}^{3} T_{ci} dm = \frac{c_pV}{R} \left( (1 - \epsilon)^2 \beta^a + \epsilon(1 - \epsilon) \frac{\tau^i_{T_{ES,n}}}{r_0} + 1 \right) \beta^2 + \epsilon \left( \frac{\tau^i_{T_{ES,n}}}{r_0} + 1 \right) - \left( 1 + 2\epsilon \frac{\tau^i_{T_{ES,n}}}{r_0} \right) \right) dp \Rightarrow
\]
\[
W_c = \frac{Vp_0}{a} \left( \frac{(1-\varepsilon)^2}{a+1} \left( \beta_2^{a+1} - \beta_1^{a+1} \right) + \frac{\varepsilon(1-\varepsilon)}{1+\frac{T_{TES}}{T_0}} \left( \beta_2^{\frac{2a}{3}+1} - \beta_1^{\frac{2a}{3}+1} \right) + \right.
\]
\[
\frac{\varepsilon \left( \frac{T_{TES}}{T_0} \right)^{\frac{a}{2a+1}}}{\left( \beta_2^{\frac{a}{3}+1} - \beta_1^{\frac{a}{3}+1} \right) - \left( \frac{2\varepsilon}{T_{TES}T_0} + 1 \right) \left( \beta_2 \beta_1 \right)} \right)
\]

(38)

b. Turbine work
\[
\begin{align*}
dW_t &= -c_p \left( T_{t1} - T_{t1} \beta_2^{-\frac{a}{3}} + T_2 - T_2 \beta_2^{-\frac{a}{3}} + T_3 - T_3^{-\frac{a}{3}} \right) dm \\
&= \frac{V}{R} dp
\end{align*}
\]
\[
W_t = -\frac{Vp_0}{aT_0} \left( T_{t1} + 2\varepsilon T_{TES}^{f} (\beta_2 - \beta_1) - \frac{\varepsilon (T_{t1}(1+\varepsilon))T_{TES}^{f}}{1-a} \left( \beta_2^{1-a} - \beta_1^{1-a} \right) + \right.
\]
\[
\frac{\varepsilon (1-a)T_{t1} - \varepsilon (1-\varepsilon)T_{TES}^{f}}{1-a} \left( \beta_2^{-\frac{2a}{3}} - \beta_1^{-\frac{2a}{3}} \right) - \left( \frac{1-\varepsilon}{1-a} T_{t1} \left( \beta_2^{1-a} - \beta_1^{1-a} \right) \right)
\]

(39)

c. Heat stored
\[
\begin{align*}
dQ_s &= c_p \left( T_{c2} - T_0 \beta_2^{-\frac{a}{3}} + T_{c3} - T_2 \beta_2^{-\frac{a}{3}} + T_{c4} - T_3 \beta_2^{-\frac{a}{3}} \right) dm \\
&= \frac{V}{R} dp \\
&= \frac{Vp_0T_0}{a} \left( \frac{(1-\varepsilon)^2}{a+1} \left( \beta_2^{a+1} - \beta_1^{a+1} \right) + \frac{\varepsilon \left( \frac{T_{TES}+1}{T_0} \right)}{\frac{2a+1}{a+1}} \left( \beta_2^{\frac{2a}{3}+1} - \beta_1^{\frac{2a}{3}+1} \right) - \right.
\]
\[
\frac{\varepsilon \left( \frac{T_{TES}+1}{T_0} \right)}{\left( \beta_2^{\frac{a}{3}+1} - \beta_1^{\frac{a}{3}+1} \right) + 3 \frac{T_{TES}T_0}{T_0} (\beta_2 - \beta_1) \right)
\]

(40)
d. Heat released

\[
dQ_r = -c_p \left( T_{t1} - T_0 + T_{t2} - T_{t1} \beta^{-\frac{a}{3}} + T_{t3} - T_{t2} \beta^{-\frac{a}{3}} \right) \frac{dm}{dm} = \frac{V}{RT_0} dp
\]

\[Q_r = - \frac{V}{aT_0} \int_{p_1}^{p_2} \left( T_{t1} - T_0 + (1 - \varepsilon)T_{t1} \beta^{-\frac{a}{3}} + \varepsilon T_{TES} - T_{t1} \beta^{-\frac{a}{3}} + (1 - \varepsilon)T_{t1} \beta^{-\frac{2a}{3}} + \varepsilon T_{TES} - (1 - \varepsilon)T_{t1} \beta^{-\frac{2a}{3}} + \varepsilon T_{TES} \beta^{-\frac{a}{3}} + \varepsilon T_{TES} \beta^{-\frac{2a}{3}} \right) dp \Rightarrow
\]

\[Q_r = - \frac{Vp_0}{ka} \left( T_{t1} - T_0 + 2\varepsilon T_{TES} f \right) (\beta_2 - \beta_1) - \frac{\varepsilon T_{t1} + \varepsilon^2 T_{TES} f}{1 - \frac{a}{3}} \left( \beta_2^{1 - \frac{a}{3}} - \beta_1^{1 - \frac{a}{3}} \right) + \frac{(\varepsilon - \varepsilon)T_{t1}}{1 - \frac{2a}{3}} \left( \beta_2^{1 - \frac{2a}{3}} - \beta_1^{1 - \frac{2a}{3}} \right) \]

6.3.4. AA-CAES with constant air storage chamber pressure and temperature (P, T model)

6.3.4.1. Thermodynamic basis

Charging process

In this model, as well as the temperature of the air storage chamber, its pressure is also kept constant, making the work exchanged with the environment \(d\dot{W} = p_{sc} dV\). During the charging process \(\dot{m}_{out} = 0\), we can relate the heat exchanged by the air storage chamber and the environment to the volume differential.

\[
\delta Q - p_s dV + c_p T_{in} dm = d \left( \frac{p_{sc} V}{RT} c_v T \right)
\]

\[d\dot{Q} = \frac{p_{sc} c_v}{R} dV + \frac{p_s R}{R} dV - c_p T_{in} dm = \left( \frac{p_{sc} c_p T_0}{R} T_0 - c_p T_{in} \frac{p_s}{RT_0} \right) dV = \frac{p_{sc} c_p (T_0 - T_{in})}{R_{g} T_0} dV
\]

Discharging process

The same equation can be applied as

\[
\delta Q = p_s dV - RT_0 dm = \frac{p_{sc}}{RT_0} dV
\]

Thermodynamic modelling of the AA-CAES process

In the P model, the air storage chamber has the following properties:

---


43 Idem.
• \( p_{sc} = p_2 \)
• \( T = T_0 = T_{sc,\text{out}} \)
• Initial volume = \( V_1 = \frac{p_1 V}{p_s} = \frac{p_1 V_2}{p_2} \)
• Maximum volume = \( V_2 = V \)

6.3.4.2. Equations according to the number of stages

1. For a 1 stage compression/expansion process (\( N=1 \))

\[
\varepsilon_1 = \frac{(m c_p)_{\text{air}} (T_{c,\text{in}} - T_{c,\text{out}})}{(m c_p)_{\text{min}} (T_{c,\text{in}} - T^i_{\text{TES,n}})} = \frac{T_0 \beta^a - T_2}{T_0 \beta^a - T^i_{\text{TES,n}}} \Rightarrow T_2 = (1 - \varepsilon)T_0 \beta^a + \varepsilon T^i_{\text{TES,n}} \tag{42}
\]

a. Compressor work

\[
\begin{align*}
\left\{ dW_c &= c_p \left[ \beta^a - 1 \right] \sum_{i=1}^{N} T_{c,i} dm = c_p (\beta^a - 1) T_{c,1} dm \\
& \Rightarrow dW_c = \frac{p_2}{a} \left[ \beta^a - 1 \right] \int_{V_1}^{V_2} dV \\
& \quad \text{dm} = \frac{p_2}{RT_0} dV
\end{align*}
\]

\[
W_c = \frac{p_2 p_0}{a} \frac{p_0}{p_0} (\beta_2^a - 1)V_2 \left( 1 - \frac{V_1}{V_2} \right) = \frac{p_2 p_0}{a} \left( \frac{p_0}{p_0} \right) (\beta_2^a - 1)V_2 \left( 1 - \frac{p_1}{p_2} \right) = \frac{p_0 V_2}{a} (\beta_2^a - 1)(\beta_2 - \beta_1)
\]

\[
W_c = \frac{p_0 V}{a} (\beta_2^a - 1)(\beta_2 - \beta_1) \tag{43}
\]

b. Expansion work

\[
\begin{align*}
\left\{ dW_t &= -c_p \left[ 1 - \beta^{-a} \right] T_{t,1} dm \\
& \Rightarrow W_t = -\frac{p_2 c_p T_{t,1}}{RT_0} \frac{T_0}{(1 - \beta^{-a})} \int dV \\
& \quad = -\frac{p_2 V_2 c_p T_{t,1}}{RT_0} \frac{T_0}{p_0} \left( 1 - \beta_2^{-a} \right) \left( 1 - \frac{p_1}{p_2} \right)
\end{align*}
\]

\[
W_t = -\frac{V p_0 T_{t,1}}{a} \frac{T_0}{T_0} \left( 1 - \beta_2^{-a} \right)(\beta_2 - \beta_1) \tag{44}
\]

c. Heat stored

\[
\begin{align*}
\left\{ dQ_s &= c_p (T_2 - T_0 \beta^a) dm \\
& \Rightarrow dQ_s = \frac{p_2}{a} \left( (1 - \varepsilon) \beta_2^a + \varepsilon - \beta_2^a \right) \int_{V_1}^{V_2} dV \Rightarrow
\end{align*}
\]
\[
\Rightarrow Q_s = \frac{\varepsilon p_0 V}{a} (1 - \beta^a_2)(\beta_2 - \beta_1)
\] (45)

d. Heat released

\[
\begin{align*}
\left\{ \begin{array}{l}
dQ_r = -c_p \varepsilon (T_{TES}^f - T_0) dm \\
dm = \frac{p_2}{RT_0} dV
\end{array} \right. \quad \Rightarrow Q_r = -\frac{p_2 \varepsilon}{a T_0} (T_{TES}^f - T_0)
\]

\[
\begin{align*}
&= -p_2 \varepsilon \int_{V_1}^{V_2} dV = -p_2 \varepsilon \frac{p_0}{a} \frac{T_{TES}^f}{T_0} (1) \frac{V_2}{p_2} (1 - \frac{p_1}{p_2}) \\
&\Rightarrow Q_r = -\frac{V p_0 \varepsilon}{a} \left( \frac{T_{TES}^f}{T_0} - 1 \right) (\beta_2 - \beta_1)
\end{align*}
\] (46)

2. For a 2-stage compression/expansion process (N=2)

\[
\varepsilon_2 = \frac{(mc_p)_{air} (T_{c,in} - T_{c,out})}{(mc_p)_{min} (T_{c,in} - T_{TES}^i)} = \frac{T_2 \beta^a - T_3^i}{T_2 \beta^a - T_{TES}^i} \Rightarrow
\]

\[
T_{c3} = (1 - \varepsilon)T_{c2} \beta^a + \varepsilon T_{TES}^i = (1 - \varepsilon)^2 T_0 \beta^a + \varepsilon (1 - \varepsilon) T_{TES}^i \beta^a + \varepsilon T_{TES}^i
\] (47)

a. Compressor work

\[
\begin{align*}
\left\{ \begin{array}{l}
dW_c = c_p \left[ \beta^a - 1 \right] \sum_{i=1}^{N} T_{c,i} dm = c_p \left( \beta^a - 1 \right) \left( T_0 + (1 - \varepsilon) T_0 \beta^a + \varepsilon T_{TES, n}^i \right) dm \\
dm = \frac{p_2}{RT_0} dV
\end{array} \right.
\]

\[
\Rightarrow W_c = \frac{p_2 c_p}{R} \left( (1 - \varepsilon) \beta^a + \varepsilon \left( \frac{T_{TES, n}^i}{T_0} + 1 \right) \beta^a \beta^a - \left( 1 + \varepsilon \frac{T_{TES, n}^i}{T_0} \right) \right) \int_{V_1}^{V_2} dV
\]

\[
\Rightarrow W_c = \frac{V_2 p_0}{a} \frac{p_0}{p_0} \left( (1 - \varepsilon) \beta^a + \varepsilon \left( \frac{T_{TES, n}^i}{T_0} + 1 \right) \beta^a \beta^a - \left( 1 + \varepsilon \frac{T_{TES, n}^i}{T_0} \right) \right) (p_2 - p_1)
\]

\[
W_c = \frac{p_0 V}{a} \left( (1 - \varepsilon) \beta^a + \varepsilon \left( \frac{T_{TES, n}^i}{T_0} + 1 \right) \beta^a \beta^a - \left( 1 + \varepsilon \frac{T_{TES, n}^i}{T_0} \right) \right) (\beta_2 - \beta_1)
\] (48)
b. **Expansion work**

\[
\begin{align*}
\left\{ dW_t &= -c_p \left( T_{t1} - T_{t1} \beta_2^a + T_{t2} - T_{t2} \beta_2^a \right) dm \\
\frac{dm}{dV} &= \frac{p_2}{RT_0} dV \\
W_t &= -\frac{p_2 c_p}{RT_0} \int (\varepsilon - 1) \beta_2^a - \varepsilon (T_{t1} + T_{TUES}^f) \beta_2^a + T_{t1} + \varepsilon T_{TUES}^f \right) (1 - \frac{p_1}{p_2}) \right) \\
W_t &= -\frac{Vp_0}{aT_0} \left( (\varepsilon - 1) T_{t1} \beta_2^a - \varepsilon (T_{t1} + T_{TUES}^f) \beta_2^a + T_{t1} + \varepsilon T_{TUES}^f \right) (\beta_2 - \beta_1) 
\end{align*}
\]

(49)

c. **Heat stored**

\[
\begin{align*}
\left\{ dQ_s &= c_p \left( T_{c2} - T_0 \beta_2^a + T_{c3} - T_{c2} \beta_2^a \right) dm \\
\frac{dm}{dV} &= \frac{p_2}{RT_0} dV \\
Q_s &= \int dQ_s = \frac{p_2}{p_0} \left( (\varepsilon - 1) \beta_2^a - \varepsilon (T_{TUES,n}^i + 1) \beta_2^a + 2\varepsilon T_{TUES,n}^i \right) (\beta_2 - \beta_1) 
\end{align*}
\]

(50)

d. **Heat released**

\[
\begin{align*}
\left\{ dQ_r &= -dmc_p(\varepsilon(T_{TUES}^f - T_0 + T_{TUES}^f - T_{t1} \beta_2^a)) dm = \frac{p_2}{RT_0} dV \\
Q_r &= -\frac{p_2}{aT_0} \left( 2T_{TUES}^f - T_0 \right) (2T_{TUES}^f - T_0 - T_{t1} \beta_2^a) V_2 \right) (1 - \frac{p_1}{p_2}) \right) \\
Q_r &= -\frac{Vp_0}{aT_0} (2T_{TUES}^f - T_0 - T_{t1} \beta_2^a) (\beta_2 - \beta_1) 
\end{align*}
\]

(51)

3. For a 3-stage compression/expansion process (N=3)

\[
\varepsilon_3 = \frac{(mc_p)_{air} (T_{c,in} - T_{c,out})}{(mc_p)_{min} (T_{c,in} - T_{TUES,n})} = \frac{T_{c3} \beta^a - T_{c4}}{T_{c3} \beta^a - T_{TUES,n}^i} \Rightarrow T_{c4} = (1 - \varepsilon)^3 T_0 \beta^a + \varepsilon (1 - \varepsilon)^2 T_{TUES,n}^i \beta_2^a + \varepsilon (1 - \varepsilon) T_{TUES,n}^i \beta_2^a + \varepsilon T_{TUES,n}^i 
\]

(52)
a. Compressor work

\[ dW_c = c_p \left( \beta^\alpha - 1 \right) \sum_{i=1}^{3} T_{ci} \, dm \]

\[ dm = \frac{p_2}{RT_0} \, dV \]

\[ \Rightarrow W_c = \frac{p_2 c_p}{R} \left( \beta^\alpha - 1 \right) \left( (1 - \varepsilon)^2 \beta_2^{\alpha - \frac{a}{3}} + (1 - \varepsilon) \left( 1 + \varepsilon \frac{T_{TES}^i}{T_0} \right) \beta_2^{\alpha - \frac{a}{3}} + 2 \varepsilon \frac{T_{TES}^i}{T_0} \right) \int_{V_1}^{V_2} dV \]

\[ \Rightarrow W_c = \frac{V_2 p_0}{a \cdot p_0} \left( (1 - \varepsilon)^2 \beta_2^\alpha + \varepsilon(1 - \varepsilon) \left( 1 + \frac{T_{TES}^i}{T_0} \right) \beta_2^{\alpha - \frac{a}{3}} + \varepsilon \left( \frac{T_{TES}^i}{T_0} + \frac{T_{TES}^i}{T_0} + 1 \right) \beta_2^{\alpha - \frac{a}{3}} \right. \]

\[ - \left. \left( 1 + 2 \varepsilon \frac{T_{TES}^i}{T_0} \right) (p_2 - p_1) \right) (\beta_2 - \beta_1) \]

\[ W_c = \frac{p_0 V}{a} \left( (1 - \varepsilon)^2 \beta_2^\alpha + \varepsilon (1 - \varepsilon) \left( 1 + \frac{T_{TES}^i}{T_0} \right) \beta_2^{\alpha - \frac{a}{3}} + \varepsilon \left( \frac{T_{TES}^i}{T_0} + \frac{T_{TES}^i}{T_0} + 1 \right) \beta_2^{\alpha - \frac{a}{3}} \right. \]

\[ + 2 \varepsilon \frac{T_{TES}^i}{T_0} \left( \beta_2 - \beta_1 \right) \] (53)

b. Expansion work

\[ dW_t = -dc_p \left( T_{t1} - T_{t2}^{\alpha - \frac{a}{3}} + T_{t2} - T_{t3}^{\alpha - \frac{a}{3}} + T_{t3} - T_{t3}^{\alpha - \frac{a}{3}} \right) \Rightarrow W_t \]

\[ dm = \frac{p_2}{RT_0} \, dV \]

\[ = -\frac{p_2 c_p}{RT_0} \left( -(1 - \varepsilon)^2 \beta_2^{-\alpha} + (\varepsilon(\varepsilon - 1)T_{t1} - \varepsilon(1 - \varepsilon)T_{TES}^f) \beta_2^{-\frac{2a}{3}} \right. \]

\[ - \left. (\varepsilon(\varepsilon + 1)T_{TES}^f + \varepsilon T_{t1}) \beta_2^{-\frac{a}{3}} + T_{t1} + 2T_{TES}^f \right) \int dV \]

\[ = -\frac{p_2 V_2 c_p}{R} \frac{p_0}{p_0} \left( -(1 - \varepsilon)^2 T_{t1} \beta_2^{-\alpha} + (\varepsilon(\varepsilon - 1)T_{t1} - \varepsilon(1 - \varepsilon)T_{TES}^f) \beta_2^{-\frac{2a}{3}} \right. \]

\[ - \left. (\varepsilon(\varepsilon + 1)T_{TES}^f + \varepsilon T_{t1}) \beta_2^{-\frac{a}{3}} + T_{t1} + 2\varepsilon T_{TES}^f \right) \left( 1 - \frac{p_1}{p_2} \right) \]

\[ W_t = -\frac{Vp_0}{aT_0} \left( -(1 - \varepsilon)^2 T_{t1} \beta_2^{-\alpha} + (\varepsilon(\varepsilon - 1)T_{t1} - \varepsilon(1 - \varepsilon)T_{TES}^f) \beta_2^{-\frac{2a}{3}} \right. \]

\[ - \left. (\varepsilon(\varepsilon + 1)T_{TES}^f + \varepsilon T_{t1}) \beta_2^{-\frac{a}{3}} + T_{t1} + 2\varepsilon T_{TES}^f \right) (\beta_2 - \beta_1) \] (54)
c. Heat stored

\[
\begin{align*}
\left\{ 
\begin{align*}
\text{\(dQ_s = c_p \left( T_{e2} - T_0 \beta_3^a + T_{e3} - T_{e2}^3 + T_{e4} - T_{e3} \beta_3^a \right) \right\} \, dm & \quad \Rightarrow \, dQ_s \\
\text{\(dm = \frac{p_2}{RT_0} \, dV \right\} \\
\text{\( = \frac{p_2 \epsilon \, p_0}{a \, p_0} \left( 3 \, T_{TES}^i \frac{T_{TES}^i}{T_0} \right) - (2 \epsilon \, T_{TES}^i \frac{T_{TES}^i}{T_0} + 1) \beta_2^a + \left( \epsilon \, T_{TES}^i \frac{T_{TES}^i}{T_0} + 1 \right) (\epsilon - 1) \beta_2^{2a} \right) \\
\text{\(- (1 - \epsilon)^2 \beta_2^a \) \right\}} \int_{V_1}^{V_2} dV \Rightarrow \\
\text{\(Q_s = \frac{p_0 \, V_2 \epsilon}{a} \left( -(1 - \epsilon)^2 \beta_2^a + \left( \epsilon \, T_{TES}^i \frac{T_{TES}^i}{T_0} + 1 \right) (\epsilon - 1) \beta_2^{2a} - \left( 2 \epsilon \, T_{TES}^i \frac{T_{TES}^i}{T_0} + 1 \right) \beta_2^{2a} + 3 \, T_{TES}^i \frac{T_{TES}^i}{T_0} \right) (\beta_2 - \beta_1) \right) \right) 
\end{align*}
\end{align*}
\]

\( (55) \)

d. Heat released

\[
\begin{align*}
\left\{ 
\begin{align*}
\text{\(dQ_r = -dmc_p \left( T_{t1} - T_0 + T_{t2} - T_{t3} \beta_2^a + T_{t3} - T_{t2} \beta_2^a \right) \right\} \, dm & \quad \Rightarrow \, Q_r \\
\text{\(dm = \frac{p_2}{RT_0} \, dV \right\} \\
\text{\( = \left( \frac{-p_2}{a \, T_0} \right) \left( T_{t1} - T_0 + 2 \epsilon \, T_{TES}^f - \epsilon \, T_{t1}^f + \epsilon \, T_{TES}^f \right) \beta_2^{2a} + \epsilon (\epsilon - 1) T_{t1} \beta_2^{2a} \right) \int_{V_1}^{V_2} dV \\
\text{\( = \left( \frac{-p_2}{a \, T_0} \right) \left( T_{t1} - T_0 + 2 \epsilon \, T_{TES}^f - \epsilon \, T_{t1}^f + \epsilon \, T_{TES}^f \right) \beta_2^{2a} \right) \int_{V_1}^{V_2} dV \Rightarrow \\
\text{\(Q_r = \frac{V_0 \, p_0}{a \, T_0} \left( T_{t1} - T_0 + 2 \epsilon \, T_{TES}^f - \epsilon \, T_{t1}^f \right) + \epsilon (\epsilon - 1) T_{t1} \beta_2^{2a} \right) (\beta_2 - \beta_1) \right) \right) 
\end{align*}
\end{align*}
\]

\( (56) \)

\[6.3.5. \text{Introduction of isentropic efficiencies in the compressors and turbines} \]

Up until this point, the equations found give the idea values for the work in the compressors and turbines. In real application however, although the turbomachinery is considered to be adiabatic, this is not the case, as the compression and expansion processes are not reversible and have losses due to factors such as friction in its components.
Therefore, isentropic efficiencies for both the compressor and the turbine must be introduced into the model. These values depend on the type of turbine or compressor used, and although currently higher values can be achieved, will initially be set to a conservative scenario of 80%.

The isentropic efficiency of the compressor, which for our model will be equal to 0.8, is defined as

$$\eta_{sc} = \frac{w_{c_{i=0}}}{w_{c_{q=0}}} = 0.8$$

The isentropic efficiency of a turbine, which for our model will be equal to 0.8, is defined as

$$\eta_{st} = \frac{w_{t_{q=0}}}{w_{t_{s=0}}} = 0.8$$

The introduction of these efficiencies implies that, due to losses and irreversibility, the work done by the compressors is actually higher, and the work extracted from the turbines is lower, provoking in the power capacity of the process and in its overall efficiency. The real values can be calculated simply by dividing $W_c$ and multiplying $W_t$ by their respective isentropic efficiencies.

**Compression work**

$$dW_c = \frac{dm c_p}{\eta_{sc}} \left( \frac{\alpha}{\beta N} - 1 \right) \sum_{i=1}^{N} T_i$$  \hspace{1cm} (57)

**Expansion work**

$$dW_t = -dm c_p \eta_{st} \sum_{i=1}^{N} T_i \left( 1 - \beta^{-\frac{\alpha}{N}} \right)$$  \hspace{1cm} (58)

With the results obtained, we can calculate both the work efficiency and the heat efficiency of the charging and discharging process.

The work efficiency is the energy generated by the expansion of the stored air in the turbines divided by the energy consumed in the compression of the air in the compressors.

$$\eta_W = \frac{W_t}{W_c}$$  \hspace{1cm} (59)

On the other hand, the thermal efficiency is the relationship between the heat released by the Thermal Energy Storage during the expansion and the heat stored in the Thermal Energy Storage during the compression.

$$\eta_Q = \frac{Q_r}{Q_s}$$  \hspace{1cm} (60)
The functioning cycle of the plant is considered as a regular distribution of 6 hours of charging during off peak demand hours, such as during the night, a 6-hour idle status, a 6-hour discharging period, and another 6-hour idle period. Therefore, in order to obtain the values for the power capacity of the plant, the turbine work is to be divided by the duration of the expansion process (6 hours). By defining the time, the mass flow rate of the air can also be calculated. A common parameter in traditional Compressed Air Energy Storage plants is the compression to expansion time ratio, which is the compression duration divided by that of the expansion. In this case, the compression/expansion ratio is 1. This parameter will allow the model’s results to be compared to those of existing plants further on in the project.
6.3.6. Results for the first cycle (n=1)

Before calculating the values for the work and heat variables expressed when the number of cycles increases, the modelling of the TES storage is needed to evaluate the effect of the heat losses on the system. Therefore, the values for the first cycle represented will be used to compare the results obtained for both models. For the first cycle, the TES is charged from ambient temperature, meaning $T_{TES,1}^i = T_0$ must be substituted in the equations for the compressor work and the heat stored. The turbine work and the heat released are non-dependant of this variable.

The non-dimensional work and heat values will be calculated for both models, as well as the TES storage temperature after the charging process to evaluate the possibility of using certain heat transfer fluids. The values can be multiplied by ambient pressure to obtain de energy density of the system, and then by the volume if the actual energy values are needed.

In order to calculate these values and that of the mass flow rate, maximum and minimum air storage pressure chamber are needed. The values will be based on the air storage chamber at the Huntorf CAES plant, which has maximum and minimum pressures of 66 bar and 46 bar and made from existing salt rock caverns, similar to the one used in this model. The maximum and minimum air cavern pressures in this model, $p_2$ and $p_1$, are therefore as 66 bar and 50 bar respectively\(^{44}\).

As can be seen in equation (17), the higher the difference between the maximum and minimum air storage temperatures, the higher the heat capacity needed of the cooling medium. This heat capacity is proportional to the mass flow rate needed to cool and heat the working fluid (the fluid that is compressed and expanded). With a difference of 20 bar, as proposed in the Huntorf CAES plant, the mass flow rates are very large and imply higher pumping energies, especially with 2 and 3 compression and expansion stages. Therefore, for this study, the difference between maximum and minimum air cavern pressures is set as 16 bar.

6.3.6.1. Comparison of both models

The following figures compare the values for the work and the heat variables of both air storage cavern models as well as their heat and thermal efficiencies.

---

Figure 35. Non-dimensional work used by the compressors according to the number of stages for both models, with isentropic efficiencies of 0.8, for fixed values of $\beta_2=66$ and $\beta_1=50$. Source: Own elaboration.

Figure 36. Non-dimensional work generated by the turbines according to the number of stages for both models, with isentropic efficiencies of 1, for fixed values of $\beta_2=66$ and $\beta_1=50$. Source: Own elaboration.
The work generated by the turbines is the most important value, as from it the power capacity of the system can be determined. As we can see, the values for the P, T chamber model are larger than those for the V, T model, indicating a higher energy (and power) density, allowing to generate more power with an equal air storage chamber volume. The values decrease considerably with the number of stages.

**Figure 37.** Non-dimensional heat stored in the TES during the compression process according to the number of stages for both models, with isentropic efficiencies of 1, for fixed values of $\beta_2=66$ and $\beta_1=50$. **Source:** Own elaboration.

**Figure 38.** Non-dimensional heat released during the expansion process according to the number of stages for both models, with isentropic efficiencies of 1, for fixed values of $\beta_2=66$ and $\beta_1=50$. **Source:** Own elaboration.
Figure 39. Work efficiency according to the number of stages, for fixed values of \( \beta_2=66 \) and \( \beta_1=50 \), for the first cycle. Source: Own elaboration.

Figure 40. Thermal efficiency according to the number of stages, for fixed values of \( \beta_2=66 \) and \( \beta_1=50 \), for the first cycle. Source: Own elaboration.
6.3.6.2. Result analysis and conclusions

By expressing non-dimensional values for the energy and the heat, these values can be referred to as energy and work densities due to the proportionality of both values (with a constant of $p_0$). This also introduces the possibility of easily and effectively comparing the values obtained to those of other energy sources.

The total, non-dimensional work consumed during compression decreases with the number of compression/expansion stages used, regardless of the model used. The biggest variation is in the addition of the second stage, in which the work needed to power the compressors is lowered by 20.3% for the P, T model and 19.7% for the V, T option. In terms of the values, for the constant pressure and temperature air storage chamber, the work consumed around 5% higher than that of the V, T model for all alternatives for the number of stages.

When it comes to the non-dimensional work output of the expansion stages, it decreases with the addition of stages, especially in the increase from one to two expansion stages, as in the compressors, in which the reduction is of 10.0% and 9.5% for the P, T and V, T storage chambers respectively. In both models, the change when adding a third expansion stage is much smaller.

The turbine energy density is around 5% higher with the P, T model in all three cases, therefore giving a higher work output and power capacity than the V, T chamber under the same pressure and volume conditions.

Combining both relations, the work efficiency of the system increases with the number of stages, as the reduction in the compression work is around double of that of the turbine work. In this case, the V, T model reaches higher values, and the evolution of the work efficiencies is very similar in both models in relation to the number of stages is very similar.

The total heat stored during the charging process decreases with the number of stages for both air storage chamber alternatives, with the isobaric and isothermal obviously giving higher values due to the higher compression work which implies higher air temperatures, although their difference is very slight. For both models, increasing the number of stages reduces the heat stored: by 9.7% with a second stage and by another 3.8% with a third one.

On the other hand, the heat released back into the air increases with the number of stages, again giving higher results for the P, T model. For the two air storage chamber alternatives, the rise in heat release with the number of stages is of barely 1.5%.

The thermal efficiency is very similar in both models and greatly increases with the number of stages due to the considerable reduction in the heat stored and the elevation of the heat released. The V, T model also possesses a higher thermal efficiency than the isobaric one.
In general terms, the V, T model gives higher values for round-trip and heat efficiency, whereas the P, T storage chamber gives higher turbine work outputs and therefore has potential to have higher power capacities.
6.4. Modelling of the Thermal Energy Storage

Having modelled the air storage chamber and calculated the compressor and turbine work, as well as the heat stored and released in each cycle, the Thermal Energy Storage will be designed and dimensioned. The design of the TES is one of the main challenges faced in the implementation of an AA-CAES power plant as it needs to be able to contain enough thermal energy to heat the air before the turbines, and its losses must be minimised in order to make the system efficient. Other challenges faced in the development of TES tanks include the need for them to be resistant to the high pressures reached throughout the process\(^45\).

Several heat storage media have been considered during the development of this project. Water was initially chosen as the medium based on its great thermal properties, such as its high specific heat capacity of around 4180 J/kg\(\cdot\)K, which allows it to store great amounts of heat without a very large increase in its temperature. However, its operating temperature range is small. At 100\(^\circ\)C it becomes vapour, losing its advantageous heat properties and causes problems in the installation such as humidity and instability. After calculating the temperatures reached throughout the system and seeing as the temperature of 100\(^\circ\)C is surpassed greatly multiple times during the process, this possibility was discarded.

Other liquid sensible heat storage materials were studied, such as liquid sodium or lithium. These have relatively high specific heat capacities, making them useful for their role in the system. Their high boiling points remove temperature restrictions and allow them to heat the working air to very high temperatures before its expansion, producing a higher work output and therefore power. However, the challenge with using liquid metals as active storage is their high melting points. This implies a minimum temperature in the TES to ensure their liquid state (the melting points of sodium and lithium are 97.9\(^\circ\)C and 180.5\(^\circ\)C respectively) which is considerably higher than ambient temperature. Therefore, maintaining the metals in a liquid state would need an external heat source programmed to prevent the temperature dropping below their melting points, which is both complicated and expensive. The higher temperatures would also lead to an increase in the TES’s thermal losses.

The most viable solution was therefore determined to be the use of a passive heat storage with air as the heat transfer fluid. The heat storage is modelled as a large cylindrical tank, filled with a loosely packed bed of rocks that absorb and store heat from the air flowing through the tank. In other

words, during the charging process, the air used as heat transfer fluid heats up in the heat exchanger and then travels through the TES from top to bottom, transferring this heat to the rocks, which reach a temperature of $T_{TES}$. During the discharging phase, cold air enters the storage from the bottom and leaves at $T_{TES}$. The charging and discharging air flows are in opposite directions for a more efficient heat transfer and better thermal stratification\textsuperscript{46}, as seen in the following figure.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure41.png}
\caption{Schematic showing the flow direction of the air during the charging and discharging processes. Source: [19]}
\end{figure}

The bed will be composed of small granite rocks, which possess the highest specific heat capacity and density amongst the other options (limestone, sandstone, etc.). This provides the TES with better thermal properties, as well as a smaller storage volume. The porosity of the bed, $\phi$, defined as the volume of air relative to the total volume, is set to $\phi = \frac{V_{\text{air}}}{V_{\text{total}}} = 0.35$.

The packed bed of rocks is a very appropriate solution for Advanced Adiabatic Compressed Air Energy Storage systems for various reasons. Firstly, a packed bed of rocks is more economically feasible and simple solution than more complex materials which have better properties but require a large inversion and maintenance costs. In addition, it has a wide temperature range and provides direct heat transfer between the heat transfer fluid and the passive storage material\textsuperscript{47}.

In pursuance of a reduction of heat losses during the process, the TES is enclosed by a thermal insulator, glass wool, with a thickness of 0.2m. Glass wool is a great fibrous thermal insulator as a

\textsuperscript{46} Jung-Wook Park, Dohyun Park, Dong-Woo Ryu, Byung-Hee Choi, Eui-Seob Park. Analysis on heat transfer and heat loss characteristics of rock cavern thermal energy storage.

\textsuperscript{47} \textit{wtr.}
result of its low thermal conductivity (around $0.04 \frac{W}{mK}$), which is achieved during its production from glass fibres which results in trapped air that is very effective in retaining heat. An additional interesting property is its acoustic insulation.

The TES tank together with the insulation would have the following cross-section:

![Cross-section of the rock bed Thermal Energy Storage modelled. Source: Own elaboration.](image)

**6.4.1. Dimensions**

Having already considered the heat losses the heat exchangers by means of its efficacity, which has been set to a value lower than the average found in current day similar equipment, the heat transfer in the Thermal Energy Storage is considered ideal. That is, the hot air is flowing for a long enough time and with an efficient direct heat transfer through the packed bed that, at the end of the process, the TES will have the temperature of the hot air leaving the heat exchanger, and the air will be at the initial temperature of the TES, which is, in our case, ambient temperature. This is equivalent to removing the second air circuit and using the compressed air directly as the heat transfer fluid by making it flow directly into the TES and then into the next compression stage, with a heat transfer efficiency of 0.7.
For the temperature difference to be the same in the heat exchanger fluid and the storage, both must have an equal heat capacity. This heat capacity has already been calculated when modelling the air storage chambers and is equal to that of the air travelling through the compressors and turbines.

The following mathematical reasoning will be applied to AA-CAES systems with a 300000 $m^3$ air storage chamber, with maximum and minimum storage chamber pressures of 66 bar and 50 bar respectively, considering a cycle of 6 hours of charging, 6 hours of an idle state followed by 6 hours of discharging and another 6 idle hours. A 3-stage system has a higher power and heat efficiency but needs a heat transfer fluid (air in this case) mass flow that is not mechanically feasible, as can be seen in the table, which has been calculated with the following equation. However, it will be included in the analysis considering that it could be possibly applied if another Thermal Energy Storage medium is used, or a different charging mechanism is applied, for example by directly introducing the working fluid into the TES without the use of a heat transfer fluid, and for the sake of comparing its results with those for systems with less stages.

$$\dot{m}_N = \frac{(mc_p)}{c_p t_{charge}} \tag{61}$$

<table>
<thead>
<tr>
<th>Number of stages, N</th>
<th>Heat transfer fluid (air) mass flow rate, $\dot{m}$ [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>257.72</td>
</tr>
<tr>
<td>2</td>
<td>515.44</td>
</tr>
<tr>
<td>3</td>
<td>773.16</td>
</tr>
</tbody>
</table>

Table 10. Necessary mass flow rate of the Heat Transfer Fluid (in this project, air) according to the number of stages. **Source:** Own elaboration.

The mass flow rates are proportional for the reason previously explained which states that the heat transfer fluid heat capacity in each heat exchanger is equal to that of the working fluid, meaning that doubling or tripling the number of stages and therefore heat exchangers does the same to the mass flow rate. In order to continue with the condition of the working fluid and the heat transfer fluid having the same heat capacity, the Thermal Energy Storage must have an equivalent $(mc_p)$ value. The total necessary volume of the TES, depending on the number of stages, for this can be calculated using the densities and relative volumes of the rock bed and air the form it.

$$\frac{(mc_p)}{N} = (mc_p)_{TES} V_{TES} c_{pTES} = \rho_{rocks} V_{rocks} c_{procks} + \rho_{air} V_{air} c_{p_{air}}$$

$$= \phi V_{TES} \rho_{air} c_{p_{air}} + (1 - \phi)V_{TES} \rho_{rocks} c_{p_{rocks}} \Rightarrow$$
\[ V_{TESN} = \frac{(mc_p)_N}{\phi \rho_{air} c_p air + (1 - \phi) \rho_{rocks} c_{rocks}} \]

With this expression, the necessary TES volume can be calculated. Because the denominator of the equation is constant for the different number of stages, the mathematical relationship between the volumes is the same as that of the heat capacities—they are proportional with a constant of N times the value for a single-stage process. The common relation between the height \( h \) and radius \( r \) of the TES was between 4 and 5 in the consulted bibliography. Toward minimising the outer lateral surface of the cylinder, considering that it is majorly through this surface that the heat losses take place, the relation is set to 5. Using the following simultaneous equations, the radius and height of the Thermal Energy Storage tank can be determined.

\[
\begin{aligned}
\frac{h_{TES}}{r_{TES}} &= 5 \\
\frac{r_{TES}}{V_{TES}} &= \frac{1}{\sqrt{\frac{5\pi}{(mc_p)_N}}} \\
V_{TES} &= \frac{\phi \rho_{air} c_p air + (1 - \phi) \rho_{rocks} c_{rocks}}{(mc_p)_N}
\end{aligned}
\]

The dimensions obtained for the TES according to the number of compression/expansion stages are

<table>
<thead>
<tr>
<th>Number of stages</th>
<th>( V_{TES}(m^3) )</th>
<th>( r_{TES}(m) )</th>
<th>( h(m) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3595.3</td>
<td>6.1</td>
<td>30.6</td>
</tr>
<tr>
<td>2</td>
<td>7190.7</td>
<td>7.7</td>
<td>38.5</td>
</tr>
<tr>
<td>3</td>
<td>10786.0</td>
<td>8.8</td>
<td>44.1</td>
</tr>
</tbody>
</table>

| Table 11. Dimensions of TES according to the number of compression stages. Source: Own elaboration.

### 6.4.2. Heat losses

For the modelling of the system, especially before the Thermal Energy Storage temperature variation reaches a permanent regime, it is necessary to calculate the temperature of the rock bed storage at the end of a full charge and discharge cycle. This temperature will be equal to the starting temperature for the TES in the next cycle, which initially is equal to ambient temperature as has been stated earlier on.
As a consequence of the TES tank obtaining the high temperatures reached during the process and the direct contact of the outer walls with the environment at ambient temperature, there is bound to be an outwards heat flow from the tank. Thermal insulation is therefore needed to prevent as much heat as possible from leaving the TES.

The final TES temperature can be calculated by means of a simple energy balance: the heat variation in the tank is equal to the heat rate withdrawn during the discharging process and that lost to the environment subtracted from the heat rate entering during the charging process.

\[
\left(\rho V c_p\right)_{TES} \frac{dT_{TES}}{dt} = \frac{Q_s}{t_{charge}} - \frac{Q_r}{t_{discharge}} - \dot{Q}_{loss}
\]  

(63)

The stored and released heat from the Thermal Energy Storage have already been calculated during the air storage chamber modelling (see 4.3), leaving the heat losses as the only unknown left. The heat losses will be calculated considering the conductive and convective heat transfers during the charging and discharging process, making the total time of the process 12 hours, consisting of 6 for the charging phase and 6 for the discharging phase.

The equation for the heat flow through a wall between two regions, which in this case are the TES tank and the environment, is the following

\[
\dot{Q}_{loss} = UA_{outer} \Delta T
\]  

(64)

In which \(A_{outer}\) is the contact surface of the regions between which the heat transfer is taking place, in \(m^2\), \(\Delta T\) is the temperature difference between both regions, in K, and \(U\) is global heat transfer coefficient, in \(\frac{W}{m^2K}\), which is the inverse of the thermal resistance, and for cylinders is calculated by the inverse of the sum of all convective and conductive heat transfer coefficients multiplied by their surrounding surface areas

\[
UA = \frac{1}{\frac{1}{A_i h_i} + \sum \frac{1}{2\pi L k_i h_i} + \frac{1}{A_e h_e}}
\]  

(65)

The equation shows two convective terms, which account for the convection of the moving air in the interior of the tank, \(h_i\), and the convection on the exterior of the tank with the environment, \(h_e\), as well as a summation of the conductive terms of all the existing insulating layers, \(i\), between both regions.
6.4.2.1. Calculation of global heat transfer coefficient, $U$

![Diagram of heat transfer coefficients](image)

**Figure 43.** Heat transfer coefficients involved in the TES heat loss. *Source:* Own elaboration.

The unpredictable and random movement of fluids, especially when travelling in turbulent regime, makes the calculation of the exact values of the convective coefficient almost impossible. In most cases, these coefficients are estimates using established non-dimensional number correlations based on the nature of the flow, the heat transfer conditions and the geometry of the conduct through which the flow is taking place. These correlations imply a base error of their precision, which is around ±15%. This must be taken into consideration when carrying out the mathematical development and calculations, as trying to reach exact values using these coefficients would be inefficient and senseless.

**Interior convective coefficient, $h_i$**

During the charging and discharging processes in which air is going through the rock bed at high speeds, the convection is considered as forced, and the corresponding correlations must be used. The cylindrical geometry of the tank is also a factor considered during the estimation of the coefficient.

Firstly, in order to determine whether the air flow is in laminar or turbulent regime, the non-dimensional Reynolds number value $Re$, dependant on the following values taken at average TES temperature during the cycle.
• the equivalent diameter\(^{48}\) of the granite rocks in the packed bed, \(D_r = 0.15\), in \([m]\)
• the density of the fluid used \(\rho\) in \([kg/m^3]\)
• the fluid travelling speed, \(v\) in \([m/s]\)
• and the dynamic viscosity of the fluid, \(\mu\) in \([kg/(m\cdot s)\)]

is calculated as follows

\[ Re = \frac{\rho v D_r}{\mu} \] \hspace{1cm} (66)

In the case of this model, which consists of cycles with 6 hours of charging, 6 hours of idle state, 6 hours of discharging, and another 6 hours of idle state, the velocity of the air flow through the TES tank is unknown, though can be calculated through the mass flow rate

\[ \dot{m} = A_t \cdot \rho \cdot v \Rightarrow v = \frac{\dot{m}}{A_t \rho} \] \hspace{1cm} (67)

in which \(A_t\) represents the cross-sectional area of the storage tank.

The mass flow rate can be calculated taking into consideration that the heat capacity of the heat transfer fluid is equal to that of the working air (as seen in 4.3.1.) with the equation (61).

Introducing equations (61) and (63) into equation (66) the Reynolds number is determined. Once the value of the Reynolds number and the flow type have been determined, it is possible to calculate the convective coefficient via the following method

1. Determination of the non-dimensional Prandtl number, \(Pr = \frac{\mu C_p}{\lambda}\)

2. Input of this value together with that obtained for \(Re\) into an approximate correlation calculated numerically that was developed specifically for rock bed models, and that is dependent on the thermal conductivity of the heat transfer fluid, the equivalent diameter of the granite pebbles, and the \(Re\) and \(Pr\) non-dimensional numbers.

\[ h_i = \left(\frac{\lambda_{air}}{D_r}\right) \left(2.576Re^{\frac{1}{3}}Pr^{\frac{1}{3}} + 0.0936Re^{0.8}Pr^{0.4}\right) \] \hspace{1cm} (50)

\(^{48}\) The equivalent diameter of the rocks is by definition the diameter of a sphere that would have the same volume as the rocks.

\(^{49}\) Jung-Wook Park, Dohyun Park, Dong-Woo Ryu, Byung-Hee Choi, Eui-Seob Park. Analysis on heat transfer and heat loss characteristics of rock cavern thermal energy storage.

The value obtained for this convective term must be between the range of values for these coefficients for forced convection in gases. That is, the value should be between 15 and $250 \frac{W}{m^2K}$.

**Conductive term**

The insulating layer used is filled with glass wool. In this case, it is simple to determine the value of the conductive contribution to the global heat transfer coefficient as all the needed values

- The thickness of the insulator, $d_{ins} = 0.2m = r_e - r_i$
- The thermal conductivity of the glass wool, $k_{ins} = 0.04 \frac{W}{mK}$

have already defined.

**Exterior convective coefficient, $h_e$**

The exterior heat transfer coefficient for ambient air, $h_e$, varies between typical values of 1 and 25 $\frac{W}{m^2K}$. Considering that the power plant is installed near a wind farm in order to take advantage of this energy at lower losses and transport costs, the area around the plant is predicted to be windy. Accordingly, the exterior heat transfer coefficient is higher than that in non-windy areas and is set as $20 \frac{W}{m^2K}$ simulating very unfavourable conditions for the minimisation of heat losses.

6.4.2.2. Results.

<table>
<thead>
<tr>
<th>Number of stages</th>
<th>Reynolds number</th>
<th>Flow type</th>
<th>$h_i \left[\frac{W}{m^2K}\right]$</th>
<th>$h_e \left[\frac{W}{m^2K}\right]$</th>
<th>$UA \left[\frac{W}{K}\right]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>97248,87</td>
<td>Turbulent</td>
<td>202.59</td>
<td>20</td>
<td>159.40</td>
</tr>
<tr>
<td>2</td>
<td>122525,90</td>
<td>Turbulent</td>
<td>240,80</td>
<td>20</td>
<td>251.83</td>
</tr>
<tr>
<td>3</td>
<td>140257,15</td>
<td>Turbulent</td>
<td>266,41</td>
<td>20</td>
<td>549,26</td>
</tr>
</tbody>
</table>

Table 12: Global heat transfer coefficients for the Thermal Energy Storage in AA-CAES systems. **Source:** Own elaboration.

The results show that the correlations used are coherent, as the value for the interior convective heat transfer coefficient is in the range of values for forced convection in gases.

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52 [http://www.farm.net/~mason/materials/thermal_conductivity.html](http://www.farm.net/~mason/materials/thermal_conductivity.html)
With these results the affirmation can be made that the TES tank is well isolated. In both cases, the value of the global heat exchange coefficient $U$ is around $13 \frac{W}{m^2K}$. To calculate the heat losses, the value for UA must be multiplied by the existing temperature difference between the tank and the environment

$$\dot{Q}_{\text{loss}} = UA(T_{\text{TES}} - T_0)$$

Once the heat losses have been calculated, the total change in temperature in the TES system can be determined and with it the final temperature of the TES for the cycle, which is equivalent to the TES temperature at the start of the next cycle.

For the isobaric and isothermal air storage chamber model (V, T model), the heat losses in the Thermal Energy Storage are shown in the following figure:

**Figure 44.** Heat losses in the V, T model according to the number of stages (N) and the number of charging/discharging cycles (n). Source: Own elaboration.

The figure shows that as the number of compression/expansion stages increases, the heat losses are reduced considerably. By adding a second compressor and turbine stage, the heat losses are reduced by 25.7%, and by adding a third one, although this adds technical difficulties due to the extremely high mass flow rates needed in the process, reduces them by 28.2%. The main reason for this is that the
temperatures reached in the TES decrease as the number of stages increases, reducing the heat flow towards the environment as is expressed in equation (64).

Regarding the number of cycles, an increase in these means an increase in the TES heat losses, especially in the first cycles after the process starts. Another detail is that, as the number of stages increases, the number of cycles the TES goes through before its heat losses are made constant and the evolution enters a stationary regime increases. For a single stage process, the heat losses stabilise after 4 full cycles, whereas the two and three stage processes need 7 and 8 full functioning cycles to reach a stationary regime.

For the P, T air storage chamber model, the evolution is as follows:

![Heat losses in the TES according to the number of cycles](image)

**Figure 45** Heat losses in the P, T model according to the number of stages (N) and the number of charging/discharging cycles (n). **Source:** Own elaboration.

In this case, the reduction in the heat losses as the number of stages increases is even more pronounced: increasing to a two-stage compressor and turbine system reduces the heat losses by 42.7%, and with a third one the values are 44.7% lower. This is because the TES temperature at the end of the charging process is a function of the heat stored during charging, which decreases with the number of cycles except for in a 1 stage process in which it is constant, as can be seen in equation (45). This causes TES temperatures to reach very high values compared to the other cases-a maximum of
967K for N=1 instead of 558K and 473K for N=2 and N=3 respectively-inducing large heat flow values from the TES to the environment.

As with the V, T model, the heat losses increase as the number of cycles increases, more outstandingly for the first few cycles, before reaching a maximum and stabilising themselves at that value. The cycles needed to reach a stationary regime in the constant pressure and temperature air storage chamber model are 4, after which, regardless of the number of compression/expansion stages used, the heat losses are quasi-constant.

6.5. Results of the model and evolution with the number of stages

Once the Thermal Energy Storage unit and its heat losses have been modelled, the final results of the model can be determined taking into consideration the air storage chamber model, the number of stages and the evolution of the variables with the number of cycles. The three-stage process is again included for comparison purposes and for the possibility of its use with a different heat transfer fluid than the one proposed but is mechanically unworkable because of the extremely high heat transfer fluid mass flow rates it uses.

Because the heat losses have been modelled for a full charging and discharging process, the final temperature of the TES in cycle \( n \) is equal to the initial temperature of the TES for cycle \( n + 1 \).

All the results have been obtained through the programming of a Microsoft Excel document in which these variables can be studied (see Appendix 2). The variables represented include the maximum and minimum air storage temperatures, which are the same as the maximum and minimum heat transfer fluid temperatures and can be important in order to decide on the implementation of the use of other heat transfer fluids if necessary, and must also be lower than the value that would according to equation (23) make the working fluid reach a maximum turbine input temperature, \( T_{t,\text{in}} \), of 1550K\(^5\), which marks the current technological and material limit for turbine input temperatures.

The turbine non-dimensional work, which is a measure of the energy density of the system, the work efficiency of the system, and the total output power capacity, this being an indicator of the number of households or industries that can be supplied by the system and a point of comparison with other alternatives for power sources, will also be represented.

The power capacity of the model is calculated as follows

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\(^5\)Sammy Houssainy, Mohammad Janbozorgi, Peggy Ip, Pirouz Kavehpour. (2017). *Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system*
\[ P_t(W) = \frac{W_t}{\rho V} \cdot p_0 \cdot V \cdot \frac{\dot{m}_{\text{discharge}}}{t_{\text{discharge}} (s)} \]  

(68)

in which:

- the ambient pressure, \( p_0 \), is set as 100000MPa (1 bar) and the air storage chamber volume, \( V \), has a value of 300000 m\(^3\)
- the discharge time of the plant, which has been set to 6 hours in the model developed, and is equal to the charge time, based on the values of other studies\(^5\).\(^4\).

**6.5.1. For the V, T model**

**TES temperatures**

The minimum TES temperature, which is equal to the TES temperature at the end of the discharging cycle and to that at the beginning of the next charging cycle, and the maximum TES temperature, which is reached at the end of the charging process, evolve as follows with the number of stages and of cycles:

![Maximum and minimum TES temperature according to the number of cycles](image)

Figure 46. Minimum and maximum TES temperatures in the V, T model according to the number of stages (N) and the number of cycles (n). Source: Own elaboration.

\(^{54}\) Sammy Houssainy, Mohamad Janbozorgi, Peggy Ip, Pirouz Kavehpour. (2017). *Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system*
In the figure, each value for the number of stages N has two curves: the top one corresponding to the maximum temperatures reached and the bottom one for the minimum.

The graph shows that as the number of stages increases, the maximum and minimum temperatures of the TES decrease. The variation is especially appreciable when the number of stages is changed from one to two, with a decrease in the minimum temperature of an average of 18.9% when the temperatures are stabilised, and a fall of 30.9% in the values for the maximum temperatures.

The difference between a two and a three-stage process is barely significant for the minimum temperatures, with a difference of 5.88%. Regarding the maximum temperatures, increasing the number of stages to three means a 15.1% drop in the values.

The number of cycles needed for the TES temperatures to reach a stationary regime is higher as the number of stages increases, with the single stage process needing only 4 stages and the other two needing 7 full charging/discharging cycles to reach constant values.

In relation to the variation with the number of cycles, as it increases, the TES temperatures do as well, with the highest variation taking place in the first and second charging cycles which correspond to the starting up of the plant from its stationary state at ambient temperature.

With these values, the use of heat transfer fluids with a boiling point lower than the temperatures shown must be discarded as it would be evaporated, leading to a very important fall in its thermal properties and to problems in the pumps, pipes and other components with the constant change in phase of the fluid during the process. With the model proposed, water, whose thermal and mechanical properties are very advantageous, is therefore inadequate to use.

The use of other heat transfer fluids proposed in some sources, such as molten metals like lithium or sodium, has to also be initially discarded due to their melting points being higher than the minimum temperatures reached and them being in a solid state.
Turbine work and power capacity

The evolution of the non-dimensional turbine work and of the power capacity (which show the same evolution due to their proportionality) is presented in these figures:

Figure 47. Non-dimensional work output for the V, T model depending on the number of stages (N) and cycles (n). Source: Own elaboration.

Figure 48. Power capacity of the turbines in the V, T model according to the number of stages (N) and cycles (n). Source: Own elaboration.
From these figures, real values for the power of a potential AA-CAES plant with the model developed are be finally determined.

As the number of expansion stages increases, the figure shows that the power capacity of the model is reduced. When the values reach a permanent regime, the difference between the power generated in a one stage process model and a two stage one is of 6.6%, with a drop of another 3.7% when adding a third turbine stage. These differences show that despite the single stage process reaching a 6.6% higher power capacity, the temperatures reached during the process are much higher and the heat losses experienced by the system are up to 42.7% higher. The temperatures reached in any case would not suppose any technical problem for the turbines, as they are considerably lower than their input temperature upper limit.

As the number of stages increase, the power capacity of the expansion process increases as well, with the largest increment occurring in the first few process cycles after the activation of the plant. The values for the power output increase steadily regardless of the number of stages until a maximum value is reached and the process reaches a stationary regime. As with the rest of the variables explained above, the single stage process reaches this stationary phase in 6 cycles, whereas the two and three stage alternatives need 9 and 10 full cycles respectively.

Work efficiency

![Graph showing work efficiency according to the number of cycles](image)

Figure 49. Work efficiency of the V, T model depending on the stages (N) and cycles (n). Source: Own elaboration.

Regarding the efficiency of the process, the evolution is the opposite of that of the other variables with the number of stages. As N increases, the work efficiency does as well. In other words,
the addition of more turbines makes the process more efficient reduces the system’s power capacity but increases its round-trip efficiency.

By increasing the number of stages from one to two, the work efficiency increases by 8.19%, with an additional increase of 5.93% when a third stage is considered. This rise in the efficiency by the addition of stages is due to the lower heat losses associated with the system working in lower ranges of temperatures, which allows the storage and release of heat into the system and the cooling and heating of the working fluid to be more effective.

The variation of the efficiencies with the number of cycles $n$ is less significant, but its rising tendency is clear. As $n$ increases, so do the roundtrip efficiencies. The variations are much more appreciable for a 1-stage process, as its strong temperature variations in the first few stages induce the same effect on its efficiency.

The values can be seen to reach their maximum much faster than any of the other variables. This is a result of the variations in the compressor and work being negligible compared to their actual values, thus provoking very slight and unappreciable differences in the efficiency.
6.5.2. For the P, T model

TES temperatures
The maximum and minimum TES temperatures in this case vary as shown in the figure:

In the graph, each colour represents the values for the temperatures of process with a certain number of stages. For each colour, the upper curve indicates maximum temperature values and the lower one indicates minimum temperatures.

For this model, the variation of the temperatures with the number of cycles and stages is very similar to that of the previous model: both the maximum and minimum temperatures decrease with the number of stages, with the largest variation occurring in the addition of a second stage, which supposes a 42.3% drop in the maximum temperatures reached and a 26.8% fall in that of the minimum ones.

In this model, the values reached for both ends of the temperature range are higher than in the V, T model. This makes the use of water impossible, due to the constant phase changes it would suffer. Other heat transfer fluids that are liquid in the TES temperature range could however be used. The use of molten lithium, whose heat properties are acceptable despite them being worse than those of water, could be studied in the single stage process in the P, T model as the heat transfer fluid, as its melting point of 371 K is surpassed in the permanent regime of this process.

![Diagram of TES temperatures](image)

**Figure 50.** Minimum and maximum TES temperatures in the P, T model according to the number of stages (N) and the number of cycles (n). **Source:** Own elaboration.
The temperatures increase as do number of cycles for which the process has been functioning, with the highest variations again during the first cycles in which the process is started up. The variations are also more pronounced as the temperatures are higher, with the most significant variation therefore being in the first cycles of the maximum temperature of a single stage process.

In the case of the P, T model, contrary to the constant volume and temperature one, the number of cycles needed for the temperatures to reach a constant level, which is 6, is independent of the number of stages.

**Turbine work and power capacity**

For the P, T model, the non-dimensional turbine work and the power capacity of the turbines as a function of the number of charging discharging cycles is shown in the following figures. As in the V, T model, they are both values are proportionate ad therefore show the same evolution.

![Figure 51. Non-dimensional work output for the P, T model depending on the number of stages (N) and cycles (n).](image)

*Source: Own elaboration.*
From the figures, it can be concluded that the turbine work output and the power capacity for the P, T model

Again, as with all the other variables and similarly to the other model, an increase in the number of stages, the figure shows that the power capacity of the model is reduced. Once the system enters a permanent regime, the difference between the power generated in a one stage process model and a two stage one is of 20.0%, a much larger difference than in the V, T model. However, when increasing the number of stages from two to three, the fall in the power capacity is less important, with a value of 3.78%.

With the isobaric and isothermal air storage chamber, the very high temperatures reached in the one compression and expansion stage process, which imply a 25.7% increase in the heat losses in comparison with the two-stage process, also account for a 20.0% increase in the power capacity and therefore is a possibility worth considering. However, higher temperatures also imply more technical difficulties and probably higher investments costs in components such as pipes and turbines.

The variation of the power capacity in the P, T model with the number of cycles is similar to that explained for the V, T alternative: it increases with the number of cycles, particularly for the first few ones. In this case though, the time taken for a stationary regime to be entered is higher and
independent of the number of stages, with all the options taking 10 full cycles to reach constant values for the power generated.

**Work efficiency**

The last variable studied, the work efficiency, has the following relation with the number of cycles in the P, T air storage chamber model.

**Figure 53.** Work efficiency of the V, T model depending on the stages (N) and cycles (n). *Source: Own elaboration.*

The evolution is very different in this model. Whereas the efficiencies for the two and three stages processes show a variation that coincides with that of the other model, the single-stage process shows a unique tendency.

With this alternative, the one-stage process has the lowest efficiency in the first stage after which it passes the two-stage process. After the 4th cycle, it passes the three-stage process and is the one with the highest round-trip efficiency. This strong increment is a result of the compressor work input being constant and independent of the TES temperatures and therefore of the number of cycles.
(equation (43)), and the turbine work output increasing as the TES temperatures rise with the number of stages, as revealed by in Figure 31.

The process with \( N=2 \) has the lowest efficiency after the first full charging and discharging cycle and is always at values below those corresponding to the \( N=3 \) process, which is the most efficient for the first 4 cycles. These curves show a smoother variation, as the compressor and turbine work both vary with the TES temperatures, which in turn are lower and show a smaller range of values in comparison to the ones in \( N=1 \).

In regard to the evolution with the number of cycles, it is the same as that of all other variables, showing a steady increase until a maximum value is reached, corresponding to the system entering its permanent regime. The change in the efficiency for the one-stage process is especially appreciable, as can be seen in the figure with its steep slope for the first 4 cycles.

The processes with \( N=1 \) and \( N=2 \) reach their permanent regimes after 3 full charging and discharging cycles as opposed to the single-stage alternative which in this model takes up to 6 cycles to reach its maximum value.

### 6.5.3. Comparison of the power output to other sources

As a final result, it is important to put the power capacity of the model into perspective by means of a comparison with other renewable power sources and real plants. To do so, the values for the AA-CAES process used will be those once the permanent regime has been reached.

The plants chosen as reference are:

- the Olmedilla Photovoltaic Park\(^{55}\), the solar power plant with the highest power capacity in Spain, which was commissioned in 2008.
- the Alcantara Dam, the hydroelectric plant with the second highest power output in Spain\(^{56}\).
- the Huntorf CAES power plant\(^{57}\), powered by traditional CAES, located in Germany.

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\(^{56}\) "".

the McIntosh CAES power plant\textsuperscript{58}, which is also powered by traditional CAES and located in the USA.

As shown in the figure, for the model studied, the power capacity is larger than that for the Olmedilla Photovoltaic Park, regardless of the number of stages and the air storage chamber model chosen.

The P, T model air storage chamber with a single compressor and turbine stage also shows a similar power output than that of the McIntosh CAES plant, which also uses compressed air storage but without the Thermal Energy Storage unit, extracting heat from the burning of a non-renewable and polluting source. The other alternatives of the AA-CAES, that appear in yellow, are a maximum of 23.6% lower (in the case of the V, T model with three stages) than the output of the McIntosh plant. This shows that it is possible to achieve similar values with the AA-CAES than those of the existing and less ecologically sustainable diabatic CAES. On the other hand, the Huntorf CAES plant has a much larger power output, partly as a result of its two separate air storage chambers and its very fast discharging phase, which takes place in 3 hours and has a turbine mass flow rate of 417 kg/s\textsuperscript{59}.

The values are however limited by the power input from wind farms or from the attached power source and therefore are very inferior to those produced in hydroelectric plants, such as the

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure54.png}
\caption{Comparison of the power capacity of an AA-CAES installation (yellow) with other renewable source power plants. \textit{Source:} Own elaboration.}
\end{figure}

\textsuperscript{59} Idem.
Alcantara Dam in Spain, which has a power capacity of over 10 times that of the lowest AA-CAES result.

To give another perspective of the possibilities of the process, the number of people that can take advantage of the power produced can be calculated. In Spain, a person consumes an average power of 550 W of electricity per day\(^6\).

It is supposed that there is a full charging and discharging cycle every 24 hours (with the 6 hours of charging, 6 of idle state, 6 hours of discharging and another 6 hours of idle state), and therefore depending on the air storage chamber used and the number of stages, the number of people that can be supplied by the system is:

<table>
<thead>
<tr>
<th>Model</th>
<th>Number of people supplied</th>
</tr>
</thead>
<tbody>
<tr>
<td>V, T ; N=1</td>
<td>169024</td>
</tr>
<tr>
<td>V, T ; N=2</td>
<td>157865</td>
</tr>
<tr>
<td>V, T ; N=3</td>
<td>152176</td>
</tr>
<tr>
<td>P, T ; N=1</td>
<td>206184</td>
</tr>
<tr>
<td>P, T ; N=2</td>
<td>164862</td>
</tr>
<tr>
<td>P, T ; N=3</td>
<td>158622</td>
</tr>
</tbody>
</table>

Table 13. The number of people that can be provided by the different AA-CAES models. Source: Own elaboration.

7. CONCLUSIONS

In general terms, the modelling of the AA-CAES process in this project, which has its obvious limitations, serves as only the first step required in the developing and investigating of this technology, for which many more variables and factors that are beyond the scope of this text must be considered. There are still many fields on which many organisations are currently working to make this a reality and give an extra value to the renewable energy sector and the sustainable storage and supply of energy, which will help solve one of the largest problems that will be faced by the world in the near future, the energetic one.

7.1. Analysis of results

From the results, several clear conclusions can be reached. Firstly, that regardless of the air storage chamber model used, the evolution of the variables studied as a function of both the number compression and expansion stages and of the number of cycles is similar. That is, as the number of stages decrease and or the number of completed process cycles increases, the TES initial and final temperatures increase, the work input and output increase, the heat losses increase and the efficiency decreases (except for the single-stage process for the P, T air storage chamber model).

In addition, it can be concluded that with an isobaric air storage chamber, the power capacity of the plant and its efficiency is higher than the corresponding V, T value for the same number of stages considered. It is therefore clear that this system is thermodynamically better and more advantageous when it comes to supplying power. While this is true, the installation and maintenance of an isobaric air storage chamber is much more challenging (no existing plants have one), as well as its preparation and construction being more difficult and expensive, although these factors are not studied in this investigation.

When it comes to deciding the number of stages chosen for the potential plant, the most logic solution would be to choose two. Although a one stage process has more power, its efficiency is generally lower and is more unstable, and although the three-stage process has the highest efficiency, its power output is lower and the high mass flow rates needed to cool down and reheat the air in three stages are too high. The two-stage process has a relatively high capacity and efficiency and more realistic mass flow rates.

The utilisation of air as the heat transfer fluid and a packed bed of rocks as the Thermal Energy Storage is a viable option. The volumes of the tank are relatively big as the thermal properties of both the air and the rocks are not very suitable, but the simplicity of their collection and instalment is very
advantageous. Although water has advantageous thermal properties and was initially thought of as the active heat storage medium (which both transports and stores the heat), the temperatures reached in the process make this option impossible with the imposed conditions.

Finally, although the values of the power capacity are much lower than other alternatives such as hydroelectric, one of the traditional CAES plants, and other obvious sources such as nuclear or gas power stations, the results are acceptable. The range of power capacity of the different models allows for power to be supplied to at least 152000 people in Spain, with a V, T storage chamber and three-stages of compression/expansion, to up to 206000 for the P, T model with N=1.

7.2. Limitations

Throughout the investigation, many different scenarios and circumstances have tried to be included, but some important limitations, that affect the model and its mechanical and economic viability have surged due to either lack of information or of them being outside of the reach of the project despite their relevance.

In the model proposed, before reaching the results obtained, several values must be fixed. While it is true that this reduces the universality of the results to a single situation, different possibilities can be evaluated by changing the values in the Excel document developed (see Appendix 8.2) if necessary.

The maximum and minimum air storage pressures, which influence all the variables studied either directly in their equation, indirectly through their influence on the heat capacity of the fluids employed or both directly and indirectly, are two of these values. Their optimisation depends on a number of factors that have been considered like the maximum possible mass flow rate in compressors, turbines and pumps, the number of stages, and the power capacity expected of the process, and others that have not been taken into account, like the cost they induce on the project or the challenges of their construction, implementation and maintenance. In an attempt to reduce this uncertainty, they have been chosen according to similar real existing models, such as real CAES plants.

Another hypothesis that affects the results and that can be varied is the relationship between the heat capacities of both fluids that take part in the system: the working fluid and the heat transfer fluid. They have been set as equal, as seen in equation (16)(15). This is then used in the equations for the heat exchangers and therefore influences the temperatures during the process and the work and heat variables. However, this can simply be changed during the calculations by simply multiplying the heat capacity of the heat transfer fluid by the factor desired.
7.3. Possibility of implantation in Europe

It is also of importance to mention the possibility of making this kind of energy storage and source a reality given the present circumstances in Europe.

The geographical and climatic conditions necessary for the development of such a project have already been seen to exist in Europe, and more specifically in Spain as well (with Spain having the second highest installed wind power capacity in Europe): large areas of salt deposits that are available for their emptying and use as air storage chambers with existing or potential wind farm locations nearby for the powering of the compressors during the charging process. In fact, the first of only two existing traditional CAES plants is located in Huntorf, Germany. However, the wind farm used must have a minimum power capacity of at least the values obtained for the compressor work input of the AA-CAES (obtained by dividing the turbine work output by the round-trip efficiency) which is of at least 160 MW depending on the model chosen, adding another restriction to the potential regions for its implantation, especially in Europe.

In addition, the European Union with policies such as the 2030 Energy Strategy has set several sustainable development and ecologic aims in which the share of renewable energy must be increased by at least 27% and greenhouse emissions must be reduced to 60% of the 1990 levels. To achieve this, the energetic challenge needs new solutions and methods to substitute traditional sources, one of which will certainly be the storage of renewable energy sources for which AA-CAES is a possible answer. Hereupon, the commitment of the European authorities is a factor that also increases the possibility of such a power plant being opened in the future.

However, there are two main obstacles that are preventing the opening of an AA-CAES power plant, as can be seen in the Annexes for the two actual AA-CAES projects (RICAS and ADELE-ING): the very high mechanical and thermodynamic demands on the components used, some of which exceed the limits of modern-day equipment, and the low financial feasibility of the project.

Of the two, the economic problem supposes a larger impediment. Most European countries have either recently gone through or still find themselves in an economic crisis implying that projects such as these in which the investment costs are high, the results and viability are not currently guaranteed, and the power capacities are still relatively low compared to more affordable alternatives, are difficult to carry out. Once the ongoing investigations and the characteristics of the turbomachinery

---

and Thermal Energy Storage systems are more advanced, the probability of the building and operating a first AA-CAES plant would increase greatly.

In summary, although an AA-CAES installation is currently improbable, the opening of one of these plants would be a great demonstration that the project can be carried out and an incentive to other organisations for the developing of similar energy-producing stations, thus increasing the importance of this source and of wind energy on the share of energetic consumption.
8. POSSIBLE FUTURE IMPROVEMENTS

8.1. Addition of a High Temperature Thermal Energy Storage (HHTES AA-CAES model)

8.1.1. Explanation

As has been mentioned in section 5.1.3. High Temperature Hybrid CAES (HTH-CAES), one of the improvements being studied with respect to AA-CAES is the addition of a second Thermal Energy Storage able to reach much higher temperatures by using an economical TES medium and removing the need of a heat transfer fluid\(^{62}\). This High Temperature Thermal Energy Storage (HTTES) is heated with energy from a source external to the whole process, therefore reducing its independence and possibly increasing its environmental impact but at the same time increasing its power capacity greatly and therefore the project’s viability.

In this section, the TES that is charged by heat transfer during the compression process is called the Low Temperature Thermal Energy Storage. According to the model studied, during the expansion process, the heat transfer fluid that exchanges heat with the air entering the turbines has an input temperature into the heat exchanger of, \(T_{LTTES}^f\). The turbines’ work output depends on this value. If \(T_{LTTES}^f\) can be increased without severely modifying other parameters that may affect the necessary work input during compression, the efficiency of the CAES system can be increased together with the work output, achieving a higher power capacity in a simple manner.

Increasing \(T_{LTTES}\) can also be used to achieve the same work output while reducing other parameters such as the air storage chamber volume or the compressors’ work demand. This allows the AA-CAES system to be implanted to many more locations, as the volume of the air cavern to achieve a reasonable output is considerably reduced\(^{63}\) and so is the power demanded from the nearby wind farms.

As the working fluid flows directly through the HTTES after having already gained heat from the LTTES (Figure 23), it can elevate its temperature to \(T_{HTTES}\), and, as we can see from the following figure, the work output after the working fluid is expanded in the turbines increases:

\(^{62}\) Sammy Houssainy, Mohammad Janbozorgi, Peggy Ip, Pirouz Kavehpour. (2017). Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system

\(^{63}\) Zhiwei Yang, Zhe Wang, Peng Ran, Zheng Li, Weidou Ni. (2014). Thermodynamic analysis of a hybrid thermal-compressed air energy storage system for the integration of wind power.
In a hypothetical isentropic (and adiabatic) process, as in this T-s diagram, the work produced during expansion can be calculated as follows, in which $T_{10} = T_{HTTES}$ and $T_{8} = T_{LTTES}$:

$$\begin{cases} h = w + q \\ q = 0 \end{cases} \Rightarrow w_{HTTES} = \Delta h_{HTTES} = c_p(T_{10} - T_{11}) > w_{AA-CAES} = \Delta h_{LTTES} = c_p(T_{8} - T_{8'})$$

indicating that the addition of a HTTES can cause a great rise in the work output without the modification of any conditions.

In this process, a new variable is introduced: the external heat used to raise the temperature of the HTTES, $Q_{ext}$. To reinforce the process’ sustainability and continue with the objective of not utilising fossil fuels during the process this heat can ideally either come from:

- An electrical resistance, which in the best-case scenario can be powered by surplus electricity generated by the wind farm, if its power capacity is higher than the needed work input for the compression and is still generating electricity after the LTTES is fully charged.
- Other sources not involving combustion or emission of greenhouse gases, for example, an exothermic chemical reaction or electricity obtained from any other renewable energy source.

For the charging process, the equations remain the same as in the case with only one TES, and the final temperature of the LTTES, if the conditions are maintained equivalent to those of the
investigation, is equal to the final temperature of the TES of the normal AA-CAES. After the charging process and right before the start of the expansion to reduce idle state heat losses, $Q_{ext}$ is received by the HTTES.

$$Q_{ext} = (mc_p)_{HTTES} \cdot (T_{HTTES} - T_i)$$

To reduce heat losses in the exchange and transport of heat, and to cut down on the investment costs implied in adding another TES with its corresponding heat exchangers, pipes and heat transfer fluid, the working fluid travels directly into the TES, which also maximises air-heat storage medium contact surface area and makes the heat transfer more efficient.

With the addition of the extra heat input, the equation for the round-trip efficiency is modified. If the work needed to create this heat through, for example the Joule effect, is equal to the heat emitted by the source and entering the system (neglecting heat losses), then the efficiency of the system would be defined as

$$\eta_{w_{HTTES}} = \frac{w_t}{w_c + w(q_{ext})} = \frac{w_t}{w_c + q_{ext}}$$

This variation of the AA-CAES can be key in making it more attractive to develop, due to the investment costs and the necessary technology being the same as that needed before and the power capacity of the process can be increased greatly or maintained while reducing the total volume of the installation. It also adds more possible locations to consider: by lowering the compressors’ use of work for an equal value of turbine work output, the AA-CAES plant can be supplied by a higher number of potential wind farms.
9. ECONOMIC AND TEMPORAL ANALYSIS

In order to efficiently complete a project, it is important to have a clear and concise temporal planning to follow, in which all the different stages of the product are well defined, and their duration specified. Providing an approximate cost of the project is also a decisive factor in deciding whether or not to carry it out.

9.1. Temporal analysis

For the temporal analysis, there are two main tools that help its planification: The Project Breakdown Schedule (PBS) and the Gantt diagram. In the PBS, the main phases of the project are determined, organised and put into categories, all of this while establishing a hierarchy between the different tasks. As this project has been done as a part of a degree at the Universidad Politécnica de Madrid, with a part developed in the INSA Lyon university, its dedication is to be of around 300 total hours.

9.1.1. Project Breakdown Structure

![Figure 56. PBS for the AA-CAES modelling project. Source: Own elaboration.](Image)
9.1.2. Gantt diagram

The Gantt diagram is a planification tool that reflects the different tasks shown in the Project Breakdown Schedule, together with their duration and their dependence with one another. In this section, together with the diagram, built with the free software *Gantt Project*, the duration of each task will be shown in a table.

<table>
<thead>
<tr>
<th>Task number</th>
<th>Task name</th>
<th>Start date</th>
<th>End date</th>
<th>Duration (days)</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>Documentation and reading reports</td>
<td>01/03/2018</td>
<td>20/04/2018</td>
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<td>1.1</td>
<td>Thermodynamics</td>
<td>01/03/2018</td>
<td>08/03/2018</td>
<td>6</td>
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<tr>
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<td>12/03/2018</td>
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<td>05/10/2018</td>
<td>15</td>
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<tr>
<td>4.1</td>
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<td>17/09/2018</td>
<td>24/09/2018</td>
<td>6</td>
</tr>
<tr>
<td>4.2</td>
<td>Heat losses</td>
<td>28/09/2018</td>
<td>05/10/2018</td>
<td>6</td>
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<td>5</td>
<td>Results and conclusions</td>
<td>23/10/2018</td>
<td>30/10/2018</td>
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<td>Report development</td>
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<td>05/11/2018</td>
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<td>23/10/2018</td>
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<td>30/10/2018</td>
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<tr>
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<td>Spanish summary</td>
<td>30/10/2018</td>
<td>05/11/2018</td>
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Table 14: Temporal analysis of the project. Source: Own elaboration.
Figure 57. Gantt diagram of the project. Source: Own elaboration.
9.2. Budget

9.2.1. Cost of labour

The first cost taken into consideration is the salary of the worker, in this case a recently graduated industrial engineer. Other engineers have been involved in the project, such as the tutors and other teachers that have helped with a specific part of the project, both in France and in Spain (places in which the project was developed. It is considered that their wage is of 30 €/h, and that they have dedicated a total of 20 hours to the project.

Considering the average salary of a recently graduated industrial engineer is around 16 €/h, the following table shows the total cost of the worker.

<table>
<thead>
<tr>
<th>Worker</th>
<th>Salary (€/h)</th>
<th>Time worked (h)</th>
<th>Total cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Recently graduated industrial engineer</td>
<td>16,00</td>
<td>320</td>
<td>5120,00</td>
</tr>
<tr>
<td>Experienced engineers (tutors)</td>
<td>30,00</td>
<td>20</td>
<td>600,00</td>
</tr>
</tbody>
</table>

Table 15. Cost of labour for of the project. Source: Own elaboration.

9.2.2. Equipment and licences

When it comes to the equipment needed for this project, the costs are as follows.

Firstly, the cost of the laptop used to carry out the entirety of the investigation must be estimated. It consists of a Lenovo Ideapad, with an approximate cost of 550,00 €. Its life expectancy is of 5 years, leaving an amortisation cost of 110,00 €.

Secondly, during the stay in France, a shared office was given to the student in the university. As it was shared with 2 other students, the cost of the rent must be divided by three. The rent of the office could be estimated to around 100,00 €/month in total, which is 33,30 € per month per student. As the office was used during a period of 4 months (March to June, both included), this leaves a total rent cost of 133,30 €.

In addition, a lot of scientific articles were consulted during the realisation of the investigation. The majority of them were obtained from the Science Direct website, which demands a licence in order to access the full articles. As the cost of this license depends on the institution, it cannot be included. The licenses for Microsoft Office, which is the main software used, are also supplied free of charge by the university and therefore are not included.
## 9.2.3. Total budget

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost (€)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Workers</td>
<td>5,720,00</td>
</tr>
<tr>
<td>Laptop amortisation</td>
<td>110,00</td>
</tr>
<tr>
<td>Rent</td>
<td>133,30</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>5,963,30</strong></td>
</tr>
</tbody>
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Table 16. Total budget for the project. Source: Own elaboration.
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### 12. APPENDICES

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<thead>
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<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<td>( \eta )</td>
<td>Efficiency</td>
<td>-</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density</td>
<td>( \frac{kg}{m^3} )</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>Heat exchanger efficiency</td>
<td>-</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>Mass flow</td>
<td>( \frac{kg}{s} )</td>
</tr>
<tr>
<td>( a )</td>
<td>Parameter that expresses ( \frac{\gamma-1}{\gamma} )</td>
<td>-</td>
</tr>
<tr>
<td>( p )</td>
<td>Pressure</td>
<td>( Pa )</td>
</tr>
<tr>
<td>( \beta )</td>
<td>Pressure ratio, ( \beta_i = \frac{p_i}{p_o} )</td>
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</tr>
<tr>
<td>( c_p )</td>
<td>Specific heat capacity</td>
<td>( \frac{J}{kg K} )</td>
</tr>
<tr>
<td>( T )</td>
<td>Temperature</td>
<td>( K )</td>
</tr>
<tr>
<td>( t )</td>
<td>Time</td>
<td>( s )</td>
</tr>
<tr>
<td>( V )</td>
<td>Volume</td>
<td>( m^3 )</td>
</tr>
<tr>
<td>( h )</td>
<td>Specific enthalpy</td>
<td>( \frac{J}{kg} )</td>
</tr>
<tr>
<td>( u )</td>
<td>Specific internal energy</td>
<td>( \frac{J}{kg} )</td>
</tr>
<tr>
<td>( R )</td>
<td>Universal gas constant</td>
<td>( \frac{J}{kg K} )</td>
</tr>
<tr>
<td>( W )</td>
<td>Work</td>
<td>( J )</td>
</tr>
<tr>
<td>( Q )</td>
<td>Heat</td>
<td>( J )</td>
</tr>
</tbody>
</table>
### List of subscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$i$</td>
<td>Compression or expansion stage $i$, varies from 1 to 3</td>
</tr>
<tr>
<td>$N$</td>
<td>Total number of compression/expansion stages</td>
</tr>
<tr>
<td>$n$</td>
<td>Charge/discharge cycle number</td>
</tr>
<tr>
<td>$c$</td>
<td>compressor</td>
</tr>
<tr>
<td>$sc$</td>
<td>Air storage chamber</td>
</tr>
<tr>
<td>$ex$</td>
<td>Heat exchanger</td>
</tr>
<tr>
<td>WF</td>
<td>Working fluid</td>
</tr>
<tr>
<td>HTF</td>
<td>Heat transfer fluid</td>
</tr>
<tr>
<td>$in$</td>
<td>Fluid entering a certain part of the process</td>
</tr>
<tr>
<td>$out$</td>
<td>Fluid leaving a certain part of the process</td>
</tr>
<tr>
<td>$s$</td>
<td>Stored in Thermal Energy Storage</td>
</tr>
<tr>
<td>$r$</td>
<td>Released from Thermal Energy Storage</td>
</tr>
<tr>
<td>$t$</td>
<td>Turbine</td>
</tr>
<tr>
<td>$c$</td>
<td>Compressor</td>
</tr>
<tr>
<td>$cm$</td>
<td>Cooling medium-equivalent to heat transfer fluid</td>
</tr>
<tr>
<td>$p_1$</td>
<td>Minimum air storage chamber pressure (Pa)</td>
</tr>
<tr>
<td>$p_2$</td>
<td>Maximum air storage chamber pressure (Pa)</td>
</tr>
<tr>
<td>TES</td>
<td>Thermal Energy Storage</td>
</tr>
<tr>
<td>HTTES</td>
<td>High Temperature Thermal Energy Storage</td>
</tr>
<tr>
<td>LTTES</td>
<td>Low Temperature Thermal Energy Storage</td>
</tr>
<tr>
<td>$0$</td>
<td>ambient conditions</td>
</tr>
</tbody>
</table>

### List of superscripts

<table>
<thead>
<tr>
<th>Superscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$i$</td>
<td>Initial state of a cycle</td>
</tr>
<tr>
<td>$f$</td>
<td>Final state after a cycle</td>
</tr>
</tbody>
</table>
12.2. APPENDIX 2: Guide to Excel document for the calculations and graphs

Simultaneously to the modelling of the AA-CAES system, all the variables and equations were introduced into an Excel document in order to be able to reach the results required and to plot them on graphs.

All the variables and parameters are interrelated, allowing the user to change parameters such as the exchanger efficiency, the air storage chamber volume, the maximum and minimum pressures, the conductivity of the TES insulator, etc. and see the variation in the results immediately.

First of all, the different parameters are given a value (the fluid $\gamma$, the maximum and minimum air storage chamber pressures and volume, the isentropic efficiencies of the turbine and compressor, and the exchanger efficiency).

![Figure 58](image)

Figure 58. Entry of the values of air storage chamber and turbomachinery parameters in the Excel document.

Next, the air storage chamber model is selected: the V, T model, which is in yellow, or the P, T model, which is in red. The document, in which all the interrelated equations have been introduced, calculates the values of the compression and expansion work and the stored and released heat, the TES heat capacity and temperature after charging, and the work and thermal efficiency for the first cycle of the process ($n=1$).

![Figure 59](image)

Figure 59. Calculation of the variables for the first cycle by the Excel document.

Once this is done, the values for the properties in the units of the International System of the TES are introduced: the density and the specific heat capacity of the rocks (which depends on the
material chosen, in this project granite), the porosity of the TES, and the density, specific heat capacity and dynamic viscosity of the heat transfer fluid (in this project air).

<table>
<thead>
<tr>
<th>rocks</th>
<th>air (T=400K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>density</td>
<td>2.750</td>
</tr>
<tr>
<td>cp</td>
<td>882</td>
</tr>
<tr>
<td>Dr</td>
<td>0.15</td>
</tr>
<tr>
<td>porosity</td>
<td>0.25</td>
</tr>
</tbody>
</table>

Figure 60. Entry of the values of the properties of Thermal Energy Storage media in the Excel document.

The document calculates the heat losses and final temperature of the TES, work consumed and generated and power capacity for all the cycles until all variables reach a stationary regime (i.e. become constant), for both the V, T model and the P, T model and for up to 3 compression/expansion stages.

As a sample, the results for the two stage (N=2) V, T model process (yellow) and the three stage (N=3) P, T model process (red) are shown. Some variables in the table, such as the TES temperature at the end of the compression process TTES, or the input temperature of the first turbine Tt1, are not shown in the results but are needed for the calculation of them, and therefore must be calculated and included in the Excel document.

<table>
<thead>
<tr>
<th>N</th>
<th>TTES</th>
<th>Tt1</th>
<th>Qcond(V,J)</th>
<th>Qcond(T,J)</th>
<th>Glaze</th>
<th>Q</th>
<th>TTES</th>
<th>Wcond(J,J)</th>
<th>Wm(J,J)</th>
<th>work_net</th>
<th>power (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>457</td>
<td>436</td>
<td>78.077</td>
<td>51.811</td>
<td>36.193</td>
<td>782</td>
<td>5.0</td>
<td>127.378</td>
<td>87.493</td>
<td>0.088</td>
<td>80.076</td>
</tr>
<tr>
<td>2</td>
<td>520</td>
<td>495</td>
<td>73.677</td>
<td>68.145</td>
<td>11.588</td>
<td>7552</td>
<td>71.2</td>
<td>128.887</td>
<td>60.595</td>
<td>0.247</td>
<td>81.681</td>
</tr>
<tr>
<td>3</td>
<td>540</td>
<td>460</td>
<td>72.985</td>
<td>70.367</td>
<td>19.912</td>
<td>7552</td>
<td>31.8</td>
<td>130.634</td>
<td>62.215</td>
<td>0.247</td>
<td>81.125</td>
</tr>
<tr>
<td>4</td>
<td>543</td>
<td>463</td>
<td>73.312</td>
<td>71.983</td>
<td>12.815</td>
<td>7552</td>
<td>31.8</td>
<td>131.127</td>
<td>62.395</td>
<td>0.247</td>
<td>81.758</td>
</tr>
<tr>
<td>5</td>
<td>546</td>
<td>463</td>
<td>71.966</td>
<td>71.712</td>
<td>12.250</td>
<td>7552</td>
<td>31.8</td>
<td>131.367</td>
<td>62.964</td>
<td>0.247</td>
<td>81.755</td>
</tr>
<tr>
<td>6</td>
<td>545</td>
<td>465</td>
<td>71.023</td>
<td>71.416</td>
<td>12.200</td>
<td>7552</td>
<td>31.8</td>
<td>131.317</td>
<td>62.499</td>
<td>0.247</td>
<td>81.631</td>
</tr>
<tr>
<td>7</td>
<td>545</td>
<td>465</td>
<td>71.020</td>
<td>71.416</td>
<td>12.200</td>
<td>7552</td>
<td>31.8</td>
<td>131.317</td>
<td>62.499</td>
<td>0.247</td>
<td>81.631</td>
</tr>
<tr>
<td>8</td>
<td>545</td>
<td>465</td>
<td>71.020</td>
<td>71.416</td>
<td>12.200</td>
<td>7552</td>
<td>31.8</td>
<td>131.317</td>
<td>62.499</td>
<td>0.247</td>
<td>81.631</td>
</tr>
<tr>
<td>9</td>
<td>546</td>
<td>466</td>
<td>71.020</td>
<td>71.416</td>
<td>12.200</td>
<td>7552</td>
<td>31.8</td>
<td>131.317</td>
<td>62.499</td>
<td>0.247</td>
<td>81.631</td>
</tr>
<tr>
<td>10</td>
<td>546</td>
<td>466</td>
<td>71.020</td>
<td>71.416</td>
<td>12.200</td>
<td>7552</td>
<td>31.8</td>
<td>131.317</td>
<td>62.499</td>
<td>0.247</td>
<td>81.631</td>
</tr>
</tbody>
</table>

Figure 61. Examples of the calculation of results by the Excel document (for and N=2 V, T model system and an N=3 P, T model system).

In addition, the document calculates the Reynolds and Prandtl number for the heat transfer fluid, its Nusselt number, the global heat transfer coefficient of the TES and the volume of the Thermal Energy Storage tank as a function of the number of stages. These results are presented in chapter 6.4.

Modelling of the Thermal Energy Storage.

<table>
<thead>
<tr>
<th>Number of stages</th>
<th>Volume</th>
<th>radius(m)</th>
<th>radius(m)</th>
<th>height</th>
<th>Pe</th>
<th>Pr</th>
<th>Nu</th>
<th>h</th>
<th>UA</th>
<th>Qlosses</th>
<th>Q</th>
<th>TTES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3000.5</td>
<td>6.1</td>
<td>6.3</td>
<td>30.6</td>
<td>372.45</td>
<td>971</td>
<td>0.58</td>
<td>196.16</td>
<td>230.55</td>
<td>20</td>
<td>257.24</td>
<td>25407.23</td>
</tr>
<tr>
<td>2</td>
<td>1800.7</td>
<td>7.7</td>
<td>7.5</td>
<td>30.5</td>
<td>372.45</td>
<td>971</td>
<td>0.58</td>
<td>220.11</td>
<td>240.00</td>
<td>20</td>
<td>389.76</td>
<td>38821.55</td>
</tr>
<tr>
<td>3</td>
<td>10730.0</td>
<td>0.0</td>
<td>0.5</td>
<td>44.1</td>
<td>372.45</td>
<td>971</td>
<td>0.58</td>
<td>256.41</td>
<td>250.00</td>
<td>20</td>
<td>545.28</td>
<td>5407.26</td>
</tr>
</tbody>
</table>

Figure 62. Calculation of TES thermal properties and heat losses by the Excel document.
12.3. APPENDIX 3: Other real values for the properties of the process

Although in the model studied, properties such as the air storage volume and pressures, heat exchanger efficiency, charge and discharge time, etc. have been fixed to certain values, other possible alternatives found in different sources are shown below.

1. Charge/discharge times | 6h / 6h
---|---
Air storage chamber volume | 300000 m³
Heat exchanger efficiency | 80%
P₁ | 46 MPa
P₂ | 66 MPa

Table 17. Specific values for a real-life application of AA-CAES. Source: [3]

2. Charge/discharge times | 7.8h / 7.8h
---|---
Air storage chamber volume | 300000 m³
Heat exchanger efficiency | 70%
P₁ | 46 MPa
P₂ | 66 MPa

Table 18. Specific AA-CAES values in a real application. Source: [7]

3. Charge/discharge times | 41.6h / 26h
---|---
Air storage chambers volume | 310000 x2 m³
Heat exchanger efficiency | -
P₁ | 48 MPa
P₂ | 66 MPa

Table 19. Parameter values for the Huntorf operating CAES plant. Source: [28]

4. Charge/discharge times | 12h / 3h
---|---
Air storage chamber volume | 560000 m³
Heat exchanger efficiency | -
P₁ | 45 MPa
P₂ | 74 MPa

Table 20. Parameter values for the McIntosh operating CAES plant. Source: [28]
12.4. APPENDIX 4: AA-CAES existing projects

There are currently two projects aiming to install Advanced Adiabatic CAES plants in progress.

12.4.1. ADELE or ADELE-ING project

The ADELE project, launched in 2009 by RWE power (Germany’s biggest power producer) in cooperation with other energy dedicated organisations, aimed to give a solution to the intermittent nature of wind and solar power, which by 2020 is expected to account for 30% of Germany’s generation of power. The distribution proposed by the Adele project is the following. As is shown in the figure and is explained before, it is advantageous to place the power plant near wind farms in order to reduce energy losses in the transport of the electricity generated in the wind turbines, as well as the cost of the needed devices for this transport.

![Figure 63. Layout of a possible AA-CAES power plant proposed in the ADELE project. Source: [24]](image-url)
RWE assures that the installation and operation of an AA-CAES system is feasible, as has been confirmed by the European Union and through a study carried out by General Electric together with RWE in 2008. The main problem with making the theory a reality is the economic feasibility: components are very specific, difficult and long preparation and construction processes are required, and the materials needed for components such as the TES and the air storage chamber support need unique thermodynamic and mechanical properties.

The AA-CAES system means that the equipment used suffers a wide range of heavy mechanical and thermal demands. In the compressors, constantly changing conditions implicit in the cyclical character of the process during charging and discharging, high temperatures and very high pressures are unprecedented and therefore have unknown effects when acting together.

In the case of the air storage cavern, the storage process in salt caverns for natural gas is familiar to engineers and energy companies but has very different conditions to those needed in AA-CAES. The nature of the process in which air is stored and removed in short periods (daily) with its consequent pressure variations as opposed to natural gas, implies that new factors must be taken into consideration, such as the adequate size and design, possible humidity issues and the reliability and durability of the chamber. It is also important to study the wind behaviour and wind power potential near possible air storage chamber locations, together with the salt cavern or aquifer availability near existing wind farms when choosing the site for the air storage chamber, as both factors are key for the AA-CAES. Investigators at ADELE also included in their possible sites the studies of existing unused caverns as this would greatly reduce the initial cost of the project.

The turbines suffer similar problems to the compressors: very high pressures, higher than maximum allowed inlet pressures today if the aim is to achieve considerably high turbine powers. Another challenge faced in the project is where to put the exceeding energy produced in the turbines if it ends up not being needed.

For the Thermal Energy Storage, the main challenge faced in the ADELE project is the choice of and adequate heat storage medium according to cost, thermal properties, behaviour in cyclical applications and durability. The chosen medium in ADELE is the use of packed rocks, which reduce the exergy losses and the overall cost of the Thermal Energy Storage.
The other challenge faced by the TES is the need for it to resist high pressures, which is achieved through a pressure vessel adapted to the working conditions specified by the system. In the ADELE system, a pre-stressed container made of concrete was added to the structure of the Thermal Energy Storage in order to withstand the large pressure values and cyclic variations. The pre-stressing elements added a real size model that was used by Züblin to simulate the real behaviour in working conditions are shown in the following figure.

![Figure 64](image1.png)

**Figure 64.** Alternatives in the ADELE project for the shapes and materials of the packed rock Thermal Energy Storage.  
*Source: [25]*

![Figure 65](image2.png)

**Figure 65.** On the left, cross section of the pre-stressing structure of the TES to withstand high pressures. On the right, part of a 1:1 sized TES developed by Züblin for experimental validation.  
*Source: [25]*
Züblin, the company that partly developed and designed the heat storage shown above on the right, is together with RWE power, one of the companies involved in the project, which also include\(^6^4\):

- General electric for the compressors, turbines and overall integration.
- Züblin and Ooms-Ittner-Hof for the heat-storage materials, components, devices and the TES thermal insulation.
- The German Aerospace Centre, DLR, for the heat storage tank concept and design, and shape, together with the study of high-temperature insulation.
- The Fraunhofer Institute of Optronics, System Technologies and Image Exploitation and the University of Magdeburg for the economic study and the modelling of the electrical grid\(^6^5\).

The project had the aim of developing the power station until the final stage to present for an economic investment for its funding. In other words, this initial phase of the project dealt with the study of its concept, feasibility, and the development of its components\(^6^6\): after a study of many different system layouts and setups and their cost optimisation, it was concluded that the most advantageous option was with 2 compression-expansion stage systems with low temperature Thermal Energy Storages\(^6^7\).

After the first phase finished in 2013, the study carried on in what is known as the ADELE-ING project, with a more engineering focus being applied, such as the preparation and durability studies for the TES and air storage chamber, the heat storage medium and its arrangement, the fluid dynamics, etc., as well as the analysis of new system variants.

As of today, the project has reached a considerably advanced position, with all the components having solutions available for their operation in the system, a cost optimisation for several variants, high round trip efficiencies (much higher when compared to those of the CAES plants in Huntorf or McIntosh), and with capital costs that are equivalent\(^6^8\) to those of already operating pumped hydroelectric energy storage plants, with values of around 1300 €/kW, therefore making it a reasonable alternative.

However, the high initial investment costs needed together with the uncertainty in the economic situation in many countries make the projects have low economic viability at the moment.

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\(^6^6\) Idem.

\(^6^7\) Idem.

\(^6^8\) Idem.
All the information above has been taken from the following official sources, which can be visited for more information:

- The official ADELE project brochure from the RWE power website.
  

- A presentation explained at the Swiss Competence Centre for Energy Research Symposium in May 5th, 2015 by an investigator of the German Aerospace Center (DLR), one of the organisations involved in the development of the ADELE project.

12.4.2. RICAS2020 project

Similar to the Germany-based ADELE project, the RICAS2020 project is a study for the design and development of an Advanced Adiabatic Compressed Air Energy Storage system, in this case in an underground research infrastructure in Austria which can be used to evaluate the performance of such systems\(^6\). In this case, the main objective of the project is to eliminate the dependence of the system on certain locations near wind farms or with existing underground salt caverns and developing the possibility of the AA-CAES system to be independent of geographical and geological conditions, allowing for it to be located wherever high electricity demands exist: near industrial areas, large cities, etc.

![Figure 66. Schematic of the process developed in the RICAS project, in which the air storage chamber support materials can be seen. Source: http://www.ricas2020.eu/project/project-objectives/design-concept/](http://www.ricas2020.eu/project/project-objectives/design-concept/)

As well as researching into possible Thermal Energy Storage configurations and materials and into possible alternatives to the design of the air storage chamber, as was done in ADELE, the project

also aims to research into more modern and safer drilling and underground constructing technologies, allowing to install the whole system underground, therefore not occupying any constructible floor space, in the most competitive price possible\(^{70}\), by, for example, taking advantage of all the excess material after drilling for the storage of heat in packed beds or for the mechanical support of the air storage chamber.

For the air storage chamber, the challenges faced by the project are the high pressures reached, the structural elements needed to withstand the fast rate of pressure variations, both of which are also studied in ADELE, together with the liability condition that the storage chamber must have a lifetime of at least 25 years. In order to achieve this, the structure is supported by 3 layers of different materials, sand as grouting and concrete and reinforced with pre-stressed steel for mechanical support. Lastly, a sealing membrane is added to keep the other layers together.

Figure 67. Cross section of the air storage chamber model in the RICAS project. Source: http://www.ricas2020.eu/research-areas/cavern-sealing-solutions/

In the case of the Thermal Energy Storage, the RICAS project concluded that rock bed storage shows similar thermodynamic performance compared to ceramics (with very good properties) at a much lower cost\(^{71}\), meaning they are the most viable option as heat storage medium when the objective is to minimise material costs and maximise exergy efficiency.

The main innovation with respect to the ADELE project comes with the condition that the whole system is underground, implying very modern and effective drilling and tunnelling techniques.


\(^{71}\) RICAS. Design study for the European underground infrastructure related to Advanced Adiabatic Compressed Air Energy Storage. THERMAL ENERGY STORAGE. Link: http://www.ricas2020.eu/research-areas/thermal-energy-storage/
and machinery. To achieve this, RICAS2020 project suggests the use of laser-assisted mechanical cutting, which provides reduced wear on the drilling tools, higher effectiveness in drilling and manipulating hard rock, and a lower power use during the tunnelling and drilling processes, as the laser beam prepares the material to be drilled thus reducing the force necessary to cut it\textsuperscript{72}.

![Figure 68. Schematic showing the drilling process and tool proposed in the RICAS project. Source: http://www.ricas2020.eu/research-areas/innovative-excavation-methods/](http://www.ricas2020.eu/research-areas/innovative-excavation-methods/)

The main objectives set by the organisation for the project are the creation of an infrastructure to allow for the investigation of an AA-CAES system in real conditions, to provide common ground for the search of economically viable solutions to the underground structures of the CAES and the Thermal Energy Storage in excavated materials, and to add to the planning and the research of the process the idea of the system life-cycle and the possible decommissioning of the plant in the future\textsuperscript{73}. The idea of the Life-Cycle Assessment according to the RICAS2020 project can be seen in the figure below. In this analysis, the social and environmental impacts of developing, building, operating and dismantling the project are all taken into consideration.

\textsuperscript{72} RICAS. Design study for the European underground infrastructure related to Advanced Adiabatic Compressed Air Energy Storage. INNOVATIVE EXCAVATION METHODS. Link: http://www.ricas2020.eu/research-areas/innovative-excavation-methods/

\textsuperscript{73} Idem. PROJECT OBJECTIVES http://www.ricas2020.eu/project/project-objectives/
Together with the University of Leoben, which has the role of project coordination and management (legal, institutional and financially wise), the following organisations, which are all very experienced in their corresponding research areas, are also involved in the project, contributing to its feasible and economically affordable development:

- **SINTEF**, an independent investigation organisation, for the sealing of the air storage chamber and investigation of material combinations resistant to high pressures and temperatures.
- **ETH Zurich University**, for the design and optimisation of the Thermal Energy Storage by investigating alternatives such as packed bed of rocks, concrete, and other Thermal Energy Storage media.
- **HBI Haerter**, an engineer consulting firm for underground facilities and tunnels, for the safety and security guidelines when during the preparation and construction of the AA-CAES plant.
- **Bayerisches Laserzentrum**, a research and development organisation.
- **General Electric**, for the turbomachinery and the engineering concepts

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• LEITAT, a Spanish Research and Development organisation, which will focus on the environmental impact of the development and construction of the project, as well as its impact on society.

• The European Union will finance the project through its Horizon 2020 research and development programme, with a budget of 1.4 million €

The numerical results, as well as the conclusions so far on the TES material are available publicly online on the official webpage of the RICAS project, as well as a report on the risk and security evaluation of the project.

All the above information has been extracted from official sources, which can be visited for more information if needed:

• RICAS2020 explanatory project video:
  https://www.youtube.com/watch?v=ry3pSaJsVBo

• The official website of the RICAS 2020 project

• The official website of SINTEF, one of the organisations cooperating in the study and research of the RICAS project