A Small-scale compressed air energy storage system

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ME200 Project

May 9, 2025

Introduction

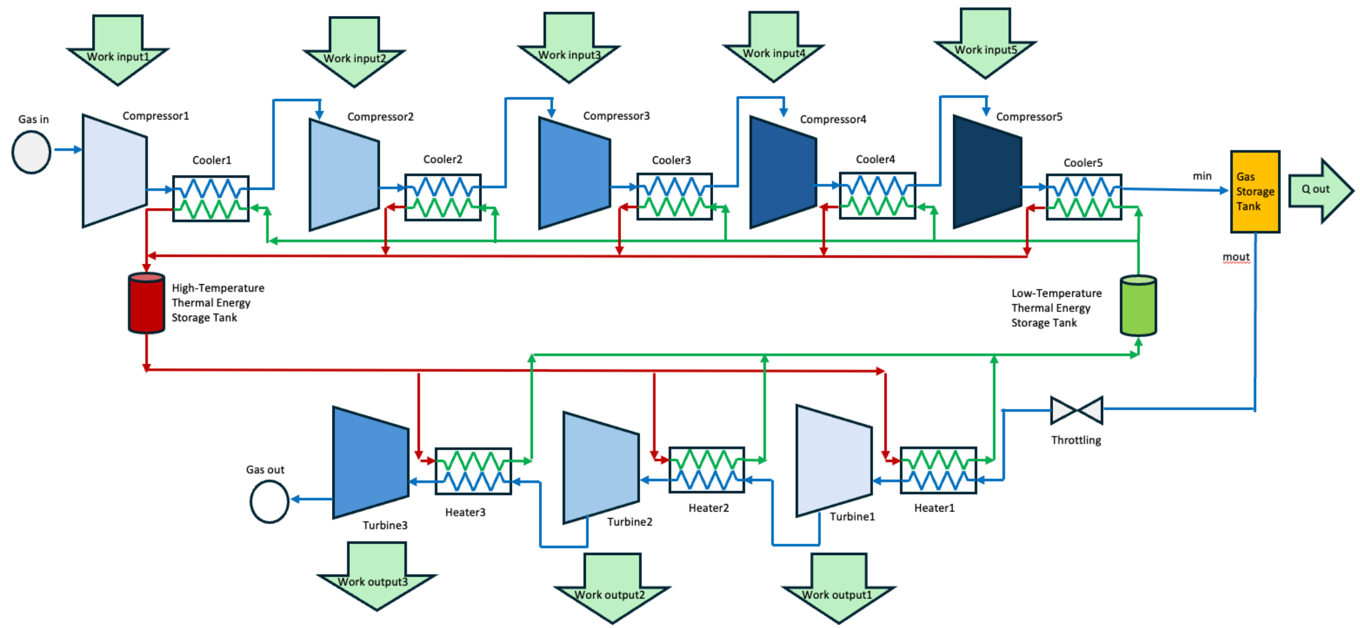

Figure: Schematic Diagram

Background

1. Small-scale compressed air energy storage systems eliminate the dependence on specific geographical conditions, offering broad applicability, low construction and operating costs. They are suitable for distributed energy systems and microgrids, providing reliable and stable power for remote areas, islands, or temporary facilities.

Simplification

1. We simplify the entire model into four parts. The first is the compressor (or expander). The second is the gas storage tank. The third is the cooler (or heater). The fourth is the throttling valve.
2. Our objective is to compute the overall efficiency of the energy storage system. Our approach is to determine the power consumption of the compressor, gas storage reservoir, and cooler. Then, identify the power output of the expander and heater. Finally, the efficiency is calculated by taking the ratio of output power to consumed power.
3. There's a simplification in that the compressor and expander, as well as the cooler and heater, are symmetric and equivalent. When analyzing, we only need to consider one of each pair and then account for the other during calculations. Here, we will focus on the compressor and cooler.

Assumption

1. R134a and air are regarded as ideal gases, whose properties satisfy the ideal gas state equation.
2. Heat dissipation in pipes and compressor is neglected as the main heat loss occurs in the gas storage tank and oil heat exchanger.
3. The ambient temperature is 25 degrees Celsius, and the atmospheric pressure is 1.014 bar.
4. Power consumption and work done during the start-up and shutdown of compressors and expanders are not considered.
5. During the entire operation, the overall average efficiency of each compressor and expander is used and remains constant. Because in the data analysis, you can see that the efficiency is only about (77.5-74.7)/77.47=3.614% difference. So we can all give them about 77% efficiency.
6. Heat loss in the thermal energy tank is not considered.Because compared with gas, liquid has almost the same specific heat, but the temperature change is almost 0, so $dertaQ$ almost 0.

Theorem

Step 1: Compressor Analysis

Given the symmetry and equivalence of compressors and expanders, we only discuss the compressor here.
First, under isentropic conditions, the formula shows that the temperature at the compressor outlet $T^{equals}_{comp,out}$ should satisfy:

$$T^{equals}_{comp,out} = T_{comp,in} \left(\beta_{comp}\right)^{\frac{k-1}{k}} \tag{1}$$

Where, $T_{comp,in}$ is the temperature of air entering the compressor, β_{comp} is the pressure ratio of the compressor, and k is the adiabatic index, taken as 1.4.
From the definition of isentropic efficiency (adiabatic efficiency) η_{comp},

$$\eta_{comp} = \frac{T^{equals}_{comp,in} - T_{comp,in}}{T^{real}_{comp,out} - T_{comp,in}} \tag{2}$$

The actual temperature at the compressor outlet $T^{real}_{comp,out}$ is obtained:

$$T^{real}_{comp,out} = \frac{1}{\eta_{comp}} T_{comp,in} \left(\left(\beta_{comp}\right)^{\frac{k-1}{k}} + \eta_{comp} - 1\right) \tag{3}$$

From the definition of β_{comp}, the pressure at the compressor outlet $p_{comp,out}$ is obtained:

$$p_{comp,out} = p_{comp,in}\beta_{comp} \tag{4}$$

Where, $p_{comp,in}$ is the import air pressure.
According to the first law of thermodynamics, the work done into the system is

$$U = Q - W \tag{4}$$

Because Q is assumpted to be 0
we can calculate the actual work consumed by a single compressor $W_{comp,i}$:

$$W_{comp,i} = \dot{m}_{comp} * c_p * \left(T^{real}_{comp,out} - T_{comp,in}\right) \tag{5}$$

Sum up to get the total work consumed by the compressor W_{comp}:

$$W_{comp} = \sum_{i=1}^{N} W_{comp,i} \tag{6}$$

Where, $\dot{m}_{comp}$ is the mass flow rate of the compressor, c_p is the specific heat capacity of air at constant pressure, N is the number of compressors.

Step 2: Gas Storage Tank Analysis

Given that the gas storage tank has a constant volume and pressure, the mass conservation equation for the storage tank is:

$$\frac{dm}{dt} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \tag{7}$$

Where, m is the air mass inside the storage tank, $\dot{m}_{\text{in}}$ and $\dot{m}_{\text{out}}$ are the air mass flow rates into and out of the storage tank.
From the definition of the heat transfer coefficient U_k between the storage tank and the environment, the heat exchange $\dot{Q}$ between the storage tank and the surrounding environment is obtained:

$$\dot{Q} = -U_k A(T - T_{\text{env}}) \tag{8}$$

Where, A is the surface area of the storage tank, T is the temperature inside the storage tank, and T_{env} is the ambient temperature.
From the first law of thermodynamics, the energy conservation equation inside the storage tank is:

$$\frac{d(mu)}{dt} = \dot{m}_{\text{in}}h_{\text{in}} - \dot{m}_{\text{out}}h_{\text{out}} + \dot{Q} \tag{9}$$

From the ideal gas state equation:

$$\frac{dm}{m} = \frac{dp}{p} + \frac{dV}{V} + \frac{dT}{T} \tag{10}$$

By rearrangement, the temperature and pressure changes within the storage tank $\frac{dT}{dt}$ and $\frac{dp}{dt}$ are obtained as:

$$\frac{dT}{dt} = \frac{1}{m}\left(\dot{m}_{\text{in}}\frac{c_p}{c_v}T_{\text{initial}} + \left[1 - \frac{c_p}{c_v}\right]\dot{m}_{\text{out}}T\right) - \frac{U_k A(T - T_{env})}{Vm}\frac{R}{c_v} \tag{11}$$

$$\frac{dp}{dt} = \frac{R}{V}\frac{c_p}{c_v}\left(\dot{m}_{\text{in}}T_{\text{initial}} - \dot{m}_{\text{out}}T\right) - \frac{U_k A(T - T_{env})}{V}\frac{R}{c_v} \tag{12}$$

Where: c_v is the specific heat capacity of air at constant volume, R is the specific gas constant for air, taken as 8314/28.97 J/(kg · K). T_{initial} is the initial temperature inside the storage tank,
For this microsystem, the pressure inside the tank affects the heat transfer coefficient between the storage tank and the environment, so this coefficient is not constant. The empirical formula is:

$$U_k = U_0 + \alpha \times (P - P_{\text{env}}) + \tau \times |\dot{m}_{\text{in}} - \dot{m}_{\text{out}}| \tag{13}$$

Where: U_0 is the heat transfer coefficient when $p = p_{\text{env}}$ and the air mass flow rate is zero; α is the coefficient reflecting the effect of pressure on convective heat transfer, and τ is the coefficient reflecting the effect of mass flow rate on convective heat transfer.

Theorem

Step 3: Heat Exchanger Analysis (Cooler and Heater)

Due to their symmetry and equivalence, only the cooler is discussed here.
In the heat exchanger, the heat exchanger effectiveness is a key indicator of performance, defined as:

$$\varepsilon = \frac{|t' - t''|_{\max}}{|t_1 - t_2|} \tag{14}$$

Where: $|t' - t''|_{\max}$ is the maximum temperature difference between the hot and cold fluids in the heat exchanger (K); $|t_1 - t_2|$ is the maximum possible temperature difference in an ideal state (K).
From the connection between the compressor and cooler, the outlet temperature of the i-th compressor equals the inlet temperature of the i-th heat exchanger, and the outlet temperature of the i-th heat exchanger is considered the inlet temperature of the $(i+1)$-th compressor. Thus, during the compression phase, the air temperature at the heat exchanger outlet is:

$$T^{i+1}_{air,comp,out} = \varepsilon_c T^i_{water,in} + (1 - \varepsilon_c) T^i_{air,comp,out} \tag{15}$$

$$T^{i+1}_{water,out} = \varepsilon_c T^i_{air,comp,in} + (1 - \varepsilon_c) T^i_{water,in} \tag{16}$$

Where: $T^{i+1}_{air,comp,out}$ is the air temperature at the outlet of the i-th heat exchanger (K), which is also the inlet temperature of the $(i+1)$-th compressor (K). ε_c is the effectiveness of the heat exchanger during compression. $T^i_{water,in}$ is the inlet temperature of the heat transfer medium in the i-th heat exchanger (K). $T^i_{air,comp,out}$ is the outlet temperature of the i-th compressor (K).

$\dot{m}_{\mathsf{cw},i}$ is the water mass flow rate into the i-th heat exchanger, in kg/s, calculated by:

$$\dot{m}_{\mathsf{cw},i} = \frac{c_{p,\mathsf{air}} \dot{m}_{\mathsf{air}} (T^i_{air,comp,out} - T^{i+1}_{air,comp,in})}{c_{p,\mathsf{water}} (T^i_{water,in} - T^i_{water,out})} \tag{17}$$

Where $c_{p,\mathsf{water}}$ is the specific heat capacity of water, in J/(kg · K). The total water mass flow rate in the heat exchanger is:

$$\dot{M}_{\mathsf{cw}} = \sum_{i=1}^{N_c} \dot{m}_{\mathsf{cw},i} \tag{18}$$

Where N_c is the number of compressor stages.
From the first law of thermodynamics, the heat transfer power from air to water in a single heat exchanger $\dot{Q}_{exchange,i}$ is calculated as:

$$\dot{Q}_{water,i} = (c_p \dot{m})_{\mathsf{air}} (T_{\mathsf{air,out},i} - T_{\mathsf{air,in},i+1}) \tag{19}$$

The total heat transfer power from air to water in all heat exchangers $\dot{Q}_{water}$ is:

$$\dot{Q}_{water} = \sum_{i=1}^{N_c} \dot{Q}_{c,i} \tag{20}$$

Step 4: Throttle Valve Analysis

The throttling process of air at the throttle valve is considered adiabatic, with the air enthalpy remaining equal before and after throttling:

$$h_{before} = h_{after} \tag{21}$$

Overall, the energy storage efficiency η is given by:

$$\eta = W_{turbine} / (W_{comp} + Q_{water}) \tag{22}$$

Data analysis example

The example below calculates the input power W of a compressor.
Compressor parameters

$T_{comp,in} = 25 + 273.15 = 298.15\,\mathrm{K}$	Inlet temperature
$\beta_{comp} = 3.5$	Pressure ratio
$\eta_{comp} = 0.744$	Isentropic efficiency
$p_{comp,in} = 0.1 \times 10^6 = 100000\,\mathrm{Pa}$	Inlet pressure
$\dot{m}_{comp} = 1692\,\mathrm{kg\,h^{-1}} = 1692/3600 \approx 0.47\,\mathrm{kg\,s^{-1}}$	Mass flow rate
$c_p = 1005\,\mathrm{J\,kg^{-1}\,K^{-1}}$	Specific heat capacity of air
$N = 1$	Number of compressors
$k = 1.4$	Adiabatic index

Calculation steps

1. Outlet temperature under isentropic conditions

$$T_{equals} = T_{comp,in} \times \beta_{comp}^{\frac{k-1}{k}} \tag{1}$$

$$\begin{aligned} T_{equals} &= 298.15 \times 3.5^{\frac{1.4-1}{1.4}} \\ &= 298.15 \times 3.5^{0.2857} \\ &\approx 298.15 \times 1.518 \\ &\approx 452.34\,\mathrm{K} \end{aligned}$$

2. Actual outlet temperature

$$T_{real,out} = \frac{1}{\eta_{comp}} \times T_{comp,in} \times \left(\beta_{comp}^{\frac{k-1}{k}} + \eta_{comp} - 1\right) \tag{2}$$

$$\begin{aligned} T_{real,out} &= \frac{1}{0.744} \times 298.15 \times \left(3.5^{0.2857} + 0.744 - 1\right) \\ &= 1.344 \times 298.15 \times (1.518 + 0.744 - 1) \\ &= 1.344 \times 298.15 \times 0.262 \\ &\approx 1.344 \times 78.08 \\ &\approx 104.92\,\mathrm{K} \\ &\approx 197.52\,^{\circ}\mathrm{C}\ \text{(Converted to Celsius)} \end{aligned}$$

3. Outlet pressure

$$p_{comp,out} = p_{comp,in} \times \beta_{comp} \tag{3}$$

$$\begin{aligned} p_{comp,out} &= 100000 \times 3.5 \\ &= 350000\,\mathrm{Pa} \end{aligned}$$

4. Work consumed by a single compressor

$$W_{comp,i} = \dot{m}_{comp} \times c_p \times (T_{real,out} - T_{comp,in}) \tag{4}$$

$$\begin{aligned} W_{comp,i} &= 0.47 \times 1005 \times (452.34 - 298.15) \\ &= 0.47 \times 1005 \times 154.19 \\ &= 0.47 \times 154965.95 \\ &\approx 72834.00\,\mathrm{W} \end{aligned}$$

Results

- Isentropic outlet temperature: Approximately $153.36\,^{\circ}\mathrm{C}$ ($452.34\,\mathrm{K}$)
- Actual outlet temperature: Approximately $197.52\,^{\circ}\mathrm{C}$ ($470.67\,\mathrm{K}$)
- Outlet pressure: $350\,000\,\mathrm{Pa}$
- Work consumed by a single compressor: Approximately $72\,834.00\,\mathrm{W}$
- Total work consumed by the compressor: Approximately $72\,834.00\,\mathrm{W}$

Data analysis

After calculating all the things above, we can finich the table as follows, all the outlet temp, inlet temp, power are calculated and listed, then after the following calculation, we finally get the η=66.48%.

Table: Air Compressor Parameters for Each Stage

Stage	Inlet Press	Outlet Press	Inlet Temp	Outlet Temp	Adiabatic Eff	Power	Flow Rate
1	0.1	0.35	25	153.0	74.4	76.7	1692
2	0.34	0.91	45	146.7	77.5	59.4	1692
3	0.89	2.40	45	147.6	80.5	57.8	1692
4	2.35	5.80	45	142.2	82.4	49.4	1692
5	5.72	11.23	45	109.1	83.0	36.3	1692

Table: Cooler Parameters for Each Stage

Cooler Stage	Inlet Air Temp	Outlet Air Temp	Inlet Water Temp	Outlet Water Temp	Water Flow Rate
1	153	45	35	120	484.6
2	146.7	45	35	120	456.3
3	146.7	45	35	120	460.4
4	142.2	45	35	120	436.2
5	109.1	45	35	60	991.7

Table: Design Parameters of Gas Storage

Storage Volume (m^3)	Storage Pressure Range (bar)
2×50	25-100

Data analysis

Table: Thermal Storage Tank Specifications

Water Tank	Operating Temp	Operating Press	Working Volume	Working Medium	Remarks
Ambient Water Tank	35	0.4	12	Boiler Water	
High-Temperature Water Tank	3120	0.4	12	Boiler Water	Insulated

Table: Parameters of Expanders for Each Stage

Stage of Expander	Inlet Press	Outlet Press	Inlet Temp	Outlet Temp	Adiabatic Efficiency	Power	Flow Rate
1	2.5	1.13	100	12	82.6	150.5	8892
2	1.12	0.4	100	13	81.0	185.5	8892
3	0.39	0.105	100	13	81.6	236.3	8892

Table: Heater Parameters for Each Stage

Heater Stage	Inlet Air Tempe	Outlet Air Temp	Inlet Water Temp	Outlet Water Tempe	Water Flow Rate
1	-15	100	120	35	2827.8
2	30	100	120	35	2163.9
3	25	100	120	35	2139.3

Table: Final Energy Storage Efficiency Calculation

Compressor Power Consumption	Turbine Output Power	Efficiency η
481.24	496.16	66.48

Impact and Summary

Impact

The first impact figure is the comparison between the isentropic efficiency and the work consumed by a single compressor.

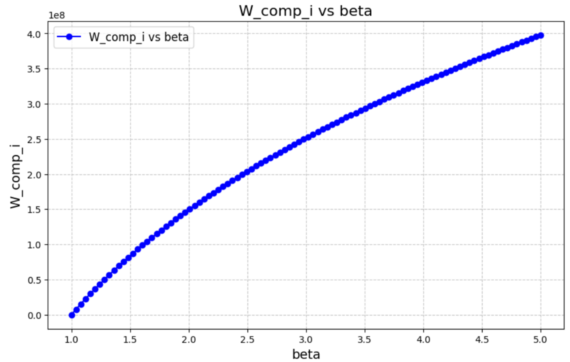

Figure: W_{comp_i} vs $beta$

The second impact figure is the comparison between the actual measured input and output flow rates of the gas storage tank and the calculated flow rates.

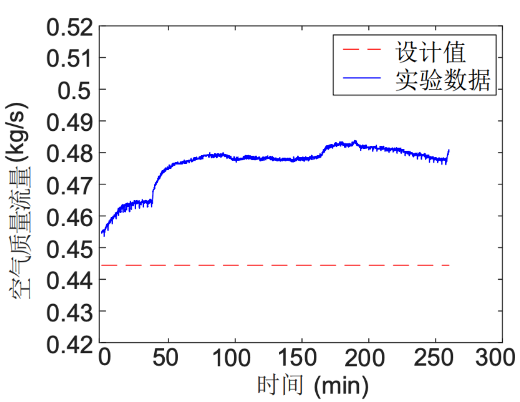

Figure: m_{in} of the gas storage tank

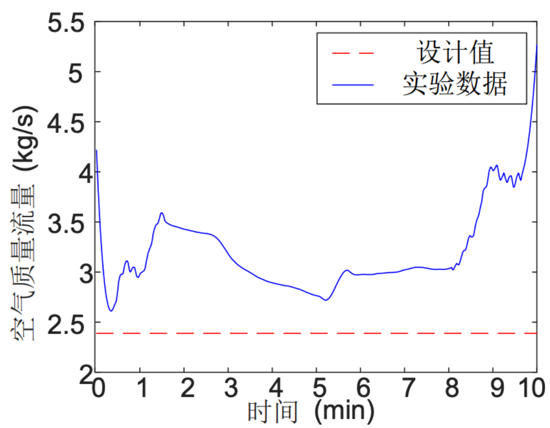

Figure: m_{out} of the gas storage tank

Summary

1. The small-scale compressed air energy storage system comprises five compressors and three expanders. During energy storage, multi-stage compression and cooling are employed, consuming electricity to collect compressive heat. In the energy release phase, multi-stage heating and expansion are utilized. While releasing the stored compressive heat, electricity is generated. The overall efficiency of this system is about 66.84%.

Future Study

1. According to our review of literature, numerous variable factors can be considered:
2. First, investigate the impact of the number of compressors and expanders on the final efficiency.
3. Second, examine how changing the storage tank medium from water to R134a or other refrigerants affects the final efficiency.
4. Third, analyze the influence of varying inlet pressures and compression ratios on the final efficiency.

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