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The cold store for a pumped thermal energy storage system

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ABSTRACT

In recent years several proposals for thermodynamic cycles involving the compression and expansion of gas and thermal storage have been put forward as effective ways of storing energy. These include the work of Desrues [1] who proposed a thermal energy storage process for large scale electric applications, Isentropic Ltd [2] who were working on a pumped thermal energy storage system and Garvey who proposed storing wind energy using a wind driven thermal pumping system known as Wind-TP [3]. All these systems require a hot and a cold store capable of storing thermal energy which can later be used to generate electricity. The efficiency and ultimately the successful adoption of pumped thermal energy storage will depend on the effectiveness of the thermal stores. In this paper we compare the performance of a packed bed and a liquid thermocline as the cold store for an off-shore Wind-TP system. Simulations are used to compare the exergetic performance of the two options leading to the conclusion that a liquid thermocline has potential to be significantly more effective than a packed bed thermocline. An addition to a liquid store involving a sliding divider separating warm and cold fluid is proposed as a way of avoiding exergy losses associated with the smearing of a thermocline front.

1. The wind driven thermal pumping cycle

In recent years several pumped thermal energy storage (PTES) systems have been put forward as efficient methods to store energy [1,2,3]. The efficiency achieved will depend on the efficiency of the constituent thermo-mechanical components. In this paper we focus on the efficiency of the thermal stores and in particular the cold store. We use the example of a wind driven thermal pumping system to evaluate the relative performance of a liquid and a packed bed cold store. The thermal pumping cycle consists of a closed circuit of working gas passing through a compressor and an expander and exchanging heat with a hot store and a cold store as shown in Fig. 1.

The cycle can operate in various modes including the following principle modes

1. Direct power transmission (no heat or coolth being transferred to stores)
2. Charging mode (heat and coolth being stored, small electrical output compared to input power)
3. Discharge mode (heat and coolth being recovered, large electrical output compared to input power)

For illustrative purposes each mode is represented in the T-S diagrams shown in Figs. 2–4 . For simplicity the compressor and expander are assumed to be isentropic and the heat exchange to and from thermal stores perfect. The working gas in the main circuit is taken to be hydrogen [3] and the pressure ratio is 25:1 with a minimum pressure of 20 bar and maximum pressure of 500 bar. The minimum temperature of the working hydrogen is 120 K and the maximum is 753 K.

An off shore wind-TP system would have a large wind turbine with rated power of at least 5 MW to be economically viable (We assume 5 MW). An energy balance around the closed gas circuit reveals that in storage mode at rated power, 5 MW of heat will be pumped into the hot store and 2 MW will be pumped out of the cold store. The aim of a wind-TP system would be to offer a wind energy storage capability on the timescale of days thus making wind power a more flexible input to the grid. We now consider how big the thermal stores would need to be to operate in charge mode at rated power for three days.

We assume an isentropic efficiency of 100% for the primary compressor and expander so the pressure and temperature ratio across these devices is related by the following

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^a \text{ where } a = (\gamma - 1/\gamma) \quad (1)$$

From the Steady Flow Energy Equation the compressor power and expander power can be related to the temperature change

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Nomenclature

α	Thermal diffusivity of liquid
A	Open inlet area of thermal store
B	Exergy
D_{eff}	Effective diffusivity of a packed bed
ε	Void fraction of a packed bed
c	Thermal front velocity
C_p	Specific heat capacity
Cp_g	Specific heat capacity of gas
Cp_l	Specific heat capacity of liquid
Cp_s	Specific heat capacity of solid
γ	Ratio of specific heats
h	Heat transfer coefficient between gas and spheres in a packed bed thermocline
k	Thermal conductivity
l	Non-dimensional length scale
L	Height of thermal store
m	Mass flow
m_g	Mass flow of gas through the packed bed
m_l	Mass flow of liquid through liquid thermocline
M	Mass
M_c	Thermal mass of cold store
M_h	Thermal mass of hot store
μ	Gas viscosity
ρ_g	Density of gas
ρ_l	Density of liquid
ρ_s	Density of solid in packed bed
P	Pressure
Q	Heat flow rate
Q_b	Heat transfer rate from horizontal surfaces
Q_r	Heat transfer rate from vertical surfaces
Q_s	Heat transfer rate between gas and solid finite elements
r_p	Pressure ratio
R	Radius of the spheres within the packed bed
T	Temperature
T_c	Cold sink temperature
T_0	Reference temperature
T_g	Temperature of the gas in packed bed
T_s	Temperature of the solid in packed bed
τ	Non-dimensional time scale
t	Time
UA_r	Overall radial heat transfer coefficient
UA_b	Overall vertical heat transfer coefficient
v_s	Superficial velocity through empty packed bed
w	Insulation width
W	Mechanical power
z	Axial position in the packed bed or liquid thermocline

across them where m is the mass flow of gas through the machines and Cp is the specific heat capacity at constant pressure of the gas. Also note that in an ideal thermal pumping cycle operating in charge mode all of the power absorbed by the compressor is equivalent to the thermal power being stored in the hot store. Similarly all the power generated by the expander is equivalent to the rate of heat removal from the process gas, i.e.

$$W_{12} = Q_{23} = mCp(T_2 - T_1) \tag{2}$$

$$W_{34} = Q_{41} = mCp(T_3 - T_4) \tag{3}$$

or using Eq. (1) then we have:

$$Q_{23} = mCpT_1(r_p{}^a - 1) \tag{4}$$

$$Q_{41} = mCpT_3\left(1 - \left(\frac{1}{r_p}\right)^a\right) \tag{5}$$

As the mass flow is the same through both compressor and expander and we assume that the heat exchange in the hot and cold stores are effective enough that $T_1 = T_3$ then

$$\frac{Q_{23}}{Q_{41}} = r_p^a \tag{6}$$

Now consider the required thermal mass to store energy in both hot and cold store, M_h and M_c respectively, for a finite time, δt, of operating in charge mode.

$$M_h = \frac{Q_{23}\delta t}{(T_2 - T_3)} = \frac{Q_{23}\delta t}{T_1\left(r_p^a - 1\right)} \tag{7}$$

$$M_c = \frac{Q_{41}\delta t}{(T_1 - T_4)} = \frac{Q_{41}\delta t r_p^a}{T_1\left(r_p^a - 1\right)} \tag{8}$$

Dividing to find the ratio of the required thermal mass in the hot and cold store and using equation 6 we arrive at the result that the required thermal mass of the hot store and cold store is the same

$$\frac{M_h}{M_c} = 1 \tag{9}$$

This remains the case when real isentropic efficiencies of the compressor and expander are accounted for. However it is no longer valid if the thermal stores are less than perfect. None the less it remains a good starting point for the design of a PTES system. From Eq. (7) or (8) we see that to have three days of storage at rated power the thermal stores would need to have a size of (2 MW × 72 h)/(180 K) = 2.88 × 10^9 J/K. Based on the heat capacity and density of the candidate thermal storage media we assume a storage tank volume of 2525 m^3. For visualisation purposes this is the volume of a 5 m radius 32 m long cylinder. This is a similar size to the floats on some pilot floating wind turbine platforms and so the authors believe could realistically be housed within a wind turbine platform.

2. Exergetic efficiency of the cold store

The efficiency and ultimately the successful adoption of pumped thermal energy storage will depend on the isentropic efficiency of the compressor and expander and also the exergetic efficiency of the thermal stores. The change in exergy δB of a given mass, M, is related to the change in its thermal energy δQ and temperature T by the following expression where T_0 is a reference temperature that the mass is heated or cooled from.

$$\delta B = \left(1 - \frac{T_0}{T}\right)\delta Q \tag{10}$$

Integrating this expression as follows gives the stored exergy when the mass is changed in temperature from T_0 to T_1, i.e.

$$B_{01} = M\int_0^1 C_p\left(1 - \frac{T_0}{T}\right)dT \tag{11}$$