Exergetic analysis of a steam-flashing thermal storage system

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ABSTRACT

Thermal energy storage is attractive in the design of concentrator solar thermal systems because of its ability to increase turbine capacity factor and to facilitate dispatchable, if not continuous, power output from a solar field. At the right cost, a storage system can improve overall economics of a solar energy system. Presented here is a simulation study of the performance of a cycle that uses large-scale thermal energy storage via hot compressed liquid water. Such a cycle is potentially interesting because of its ability to allow collector field, thermal storage, and power cycle to all work with the same fluid, thereby eliminating losses associated with heat exchangers working between different fluids. An exergetic analysis demonstrates that most of the exergy destruction that results from integrating this type of storage is due to the steam flashing process. Analysis identifies that a variable-pressure accumulator sized with a volumetric capacity for peak-load is the most effective method of implementing a steam flashing storage system. Although high efficiencies are possible it seems likely that the high cost of high pressure storage ultimately makes the concept nonviable.

Keywords: Exergy, modelling, steam flashing, thermal storage

INTRODUCTION

As solar thermal technology is still in its infancy compared to more traditional power generation types, the power generation capabilities have been small comparatively. In addition, thermal storage systems have generally, until recently, only been implemented in solar-powered generation facilities on the comparatively small scale, as storage is not required in traditional fossil fuel power plants. Compressed-liquid systems are just one of the various means of thermal storage that have been implemented in the past. The Eurelios plant in Sicily used two small storage systems, a steam accumulator containing compressed-water, and a molten salt system. The compressed-water system delivered vapour ranging from 19 bar to 7 bar and combined with the molten salt storage system could only support the plant for 30 minutes at reduced electrical output (Gretz 1987). There was also a 5 MW solar thermal plant in the Crimea, USSR built in 1985 utilizing a compressed-water storage, however little information is available for specific details (Mills 2004).

The PS10 system uses a compressed-water system working at a maximum pressure of 40 bar with a temperature of 250°C to the minimum pressure allowed by the system to run the turbine at a 50% partial load which lasts 50 minutes (Solúcar 2006). For this system there is a reported solar-energy-to-electricity efficiency of the plant of about 17.5%, a storage efficiency of about 92.4% and a Rankine cycle energetic efficiency of 30.75% (Medrano *et al.* 2010). The PS20 system has also been commissioned in the past year, however exact storage details at this stage are scarce.

Development of thermal storage systems has been evolving rapidly with recent projects by Andasol, Abengoa and Torresol with plants in Spain (Medrano *et al.* 2010). Other than recent projects, thermal storage systems thus far integrated and in operation in solar power plants have either have been experimental storage systems, or have been only capable of buffering cloudy transitions, and would still be regarded as solar-only plants meaning that they have no capabilities outside sunny conditions, although cost benefits are still evident. Currently, there are many solar power plants that have been announced for construction or already in construction and many of these may contain thermal storage (California Energy Commission 2009). As a result, more research and implementation of storage systems will occur.

The method for incorporating a thermal storage system into a concentrating solar power (CSP) plant can vary greatly, and this is demonstrated in past instances. Several factors that are important for maximising efficiency include the storage media, working fluid, working and storage temperatures and pressures, as well as economic factors and scale of the power plant. Rosen (2004) presents some design methods and features for stratified tanks filled with water while Steinmann (2006) presents various methods of integrating an accumulator into a solar thermal plant, some of which include directly injecting steam, use of a heat-transfer fluid or through usage of secondary storage materials.

DYNAMIC STORAGE MODELLING

Storage systems can be easily analysed as control volumes and, as a result, the relations fundamental to the models in this article are the Conservation of Mass, the First Law of Thermodynamics and the Temperature-Entropy (Tds) equation. The models were implemented in Matlab (Mathworks 2007) and fluid properties were calculated from the open-source software X Steam for Matlab (Holmgren 2006).

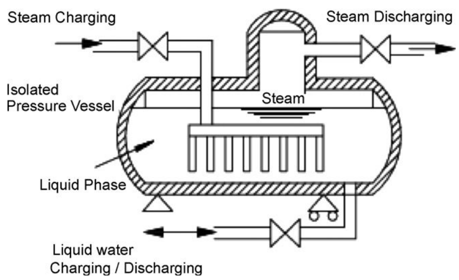

Fig. 1: Scheme of variable-pressure accumulator (Steinmann & Eck 2006)

A model was constructed for a variable-pressure accumulator, also known as Ruth's accumulator, shown in Figure 1, consisting of a tank almost completely of full of water with a small volume of steam above it. The model assumes a well-mixed fluid in the tank and that the latent heat energy for the vapourisation phase change is entirely provided for by the liquid phase during internal flashing process. The system is constrained by the total volume of the tank and starting from this constant volume assumption, we obtain Equation 3.

$$\frac{dV}{dt} = \frac{d(vm)}{dt} = 0 \qquad \textit{Eq. 1}$$

$$m\frac{dv}{dt} + v\frac{dm}{dt} = 0 \qquad \textit{Eq. 2}$$

$$\frac{dv}{dt} = -\frac{v}{m}\frac{dm}{dt} \qquad \textit{Eq. 3}$$

During discharge, steam is drawn off the top which results in a decrease in the pressure of the vessel and consequently causes some of the water to flash to steam (Ter-Gazarian 1994). During charging, superheated steam is injected into the water, and the heat of condensation of the injected steam causes an increase the temperature and pressure with only a small variation in mass (Steinmann & Eck 2006). The amount of steam that can be drawn off from the variable-pressure accumulator during discharge is in a direct relationship with the pressure drop that occurs from the start to end of the flashing procedure.

Steinmann (2006) applies the First Law of Thermodynamics with the assumptions of no work, heat transfer or potential or kinetic energy changes, and the following sets of equations are derived:

$$dE = \dot{m}_e h_e \qquad \textit{Eq. 4}$$

$$d(m_{vessel} u_{vessel}) = h_e dm_{vessel} \qquad \textit{Eq. 5}$$

$$u_{vessel} dm_{vessel} + m_{vessel} du_{vessel} = h_g(p_{vessel})\, dm_{vessel} \qquad \textit{Eq. 6}$$

$$m_{steam} \Delta h_{fg,m} = m_{liquid} Cp_{liquid,m}(T_{sat}(p_{start}) - T_{sat}(p_{end})) \qquad \textit{Eq. 7}$$

This final equation is based on several assumptions:
• Complete heat of evaporation is provided by the liquid phase.
• Specific heat capacity of liquid water and specific heat of vapourisation is approximated by the average value $Cp_{liquid,m}$ and $\Delta h_{fg,m}$ for saturated liquid water at average pressure $p_m = (p_{start} + p_{end})$
• The change in liquid mass m_{liquid} during depressurization is neglected.

The use of the Equation 6 depends on the phase of the fluid within the system. For subcritical temperatures, where the fluid is in the two-phase liquid/vapour region, rearrangement of Equation 7, with the incorporation of heat loss gives the following equation:

$$T_{sat}(p_{end}) = T_{sat}(p_{start}) + \frac{dm(h-u) - Q_{loss}}{m_{vessel} Cp_{vessel}} \qquad \textit{Eq. 8}$$

A simple Euler method for solving ordinary differential equations has been implemented in this model. Since the model is implemented using a time-stepping method, the effects of the last two assumptions stated above are minimised greatly, as, for a small enough time step, there is little difference between the fluid properties within the duration of the time-step.

The pressure change can be determined here based purely on the saturation temperature found. With the updated value of pressure, all remaining fluid properties can be determined because total volume is constant. In the supercritical temperature region, integration of Equation 6 gives the following equation, which, in combination with the calculated specific volume from Equation 3, allows determination of all other fluid properties.

$$h_{end} = h_{start} + \frac{dm(h-u) - Q_{loss}}{m_{vessel}} \qquad \textit{Eq. 9}$$

An important factor to consider in use of thermal storage is heat loss from the storage tank to the surrounding environment and, consequently, heat loss calculations have been added to this model to gain a realistic insight into the storage and cost analyses.

STORAGE EVALUATION

Isolating the storage system allows verification of the model and an indication of the dynamic nature of the vessel, as well as the effects of differing modes of operation and boundary conditions. A variable-pressure accumulator can be used for a fluid at both subcritical and supercritical temperatures, and, as long as the storage vessel is able to withstand the pressures and temperatures, it can be used to alternate between these two states. For heat-transfer purposes, the storage tank has been modelled as a horizontal cylinder allowing the maximum possible area for steam flashing to occur. In simulation, steam at constant temperature was injected into the vessel, and then the vessel was discharged until a minimum pressure was reached. This process was then repeated several times. The storage model was simulated with 0.5 m of insulation to minimise heat loss so that the effect of heat loss on the overall efficiency is small. The storage vessel needs to be able to withstand higher pressures than those for which steam accumulators are normally designed. Without these high pressures, obtaining high temperature steam out of the storage vessel would be impossible.

In Figure 2 the grey lines indicate the storage vessel limited to a subcritical pressure of 200 bar, whereas the black dashed lines show a simulation limited to a supercritical pressure of 300 bar. As seen in the left diagram in Figure 2, the addition of high-temperature steam followed by the extraction of saturated (flashed) vapour is not a cyclic process, and gradually leads to an increase in the internal energy of the vessel contents. This is more noticeable once the system becomes supercritical, as each cycle causes an overall increase in entropy in the system, however this occurs in both cycles. As a result, the specific volume of the fluid in the storage increases, thus reducing the amount of mass that can be stored at the same pressure. The difference in energy in the mass flows causes a different rate of pressure change per unit mass and, consequently, there is always more mass removed than injected. Overcoming this issue was achieved by a recharge of the storage tank with liquid water at the end of the cycle. This not only overcomes the problem with a reducing mass but also allows the storage to return to close to its initial state as the cold injected water condenses the majority of the low pressure inoperable steam back to liquid. The diagram on the right in Figure 2 shows the difference that the recharge with cold liquid water has on the overall storage cycle. This consequently results in the same cycle each time. This process was created for model verification only and may not be done in practice.

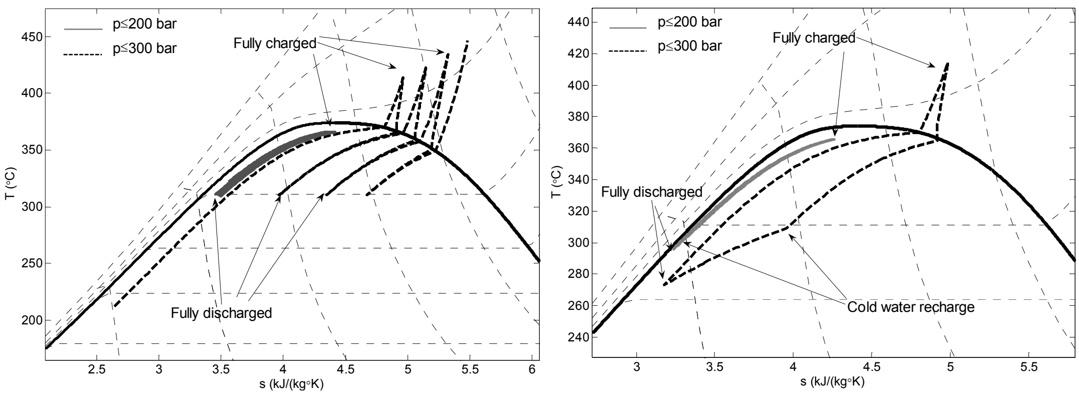

Fig. 2: T-s diagram of the accumulator charging/discharging process for super-critical and sub-critical pressure limited storage vessels (a) without (b) with cold water recharge for super-critical and sub-critical pressure limited storage vessels.

Integration with time of the charging/discharging cycle allows determination of the round-trip efficiency of the storage. The variable-pressure accumulator gives energetic and exergetic efficiencies of 99.6% and 92.3% respectively. The loss in exergetic efficiency results from the properties of the stored fluid varying considerably across the cycle, primarily because of the change in pressure that occurs during the internal flashing process and the cold water injection. As a result, the extracted steam can never reach the state of the injected fluid for any useful length of time. The loss of 0.4% in energetic efficiency is associated with thermal losses from the storage.

TIME DEPENDENT ANALYSIS

The integrated storage system under analysis in this study looks at two scenarios, a base-load demand and a peak-matching load demand, in addition to its overall efficiencies, to determine if there is a definitive difference or effective use in both cases. Based on the scope of this study, a dynamic analysis has only been conducted on the storage system. No dynamic analysis was conducted on other processes, such that immediate response to changing conditions with no transient effects occurs. As a result, processes other than the storage system were implemented at a quasi-steady state.

Model Assumptions

Conditions for the cycle have been designed for a typical turbine used in solar power plants. Although well in excess in capacity, the Siemens SST-700 steam turbine is often used for solar thermal powered plants and has been designed to for this analysis (Siemens 2009) and allows the system to continue to run a maximum capacity based on the solar radiation. The systems requirements include a minimum turbine exhaust of 90% steam quality and minimum daytime output of 10 MW and the solar collector and storage will be sized against this during the winter period for the base-load regime and during the summer period for peak-load regime encompassing a supply for several hours into the night to utilise high electricity prices during summer evenings and buffering of the system during cloudy periods.

Isentropic turbine and pump efficiencies of 87% and 80% respectively are assumed (Coventry & Pye 2009) and the effects of pipe friction and heat transfer on items other than the storage vessel have not been included in the model. Typical Meteorological Year (TMY) data for Alice Springs was used to simulate real solar conditions (Energy Partners *et al.* 2005). It is assumed the solar collector is mass-flow-controlled, such that with the changing solar radiation throughout the day, the temperature and pressure of the fluid through the collector remain constant. In order to achieve this, the mass flow through the collector varies in response to the solar radiation. The model assumes that all electricity generated is sent to the grid and uses a time interval of 5 minutes but achieves numerical stability up to 20 minutes.

Storage analysis and electricity load investigations

Initially, in order to gain an understanding of the full system, a steady-state model was conducted with no dynamic effects. This allowed an understanding of the operating points and a first-base estimate of efficiency and necessary improvements. Transient modelling of the system was achieved using an enhanced model of this steady state model integrated with the dynamic storage model from the previous section and is shown in Figure 3. Both the subcritical and supercritical systems were incorporated into a unified model.

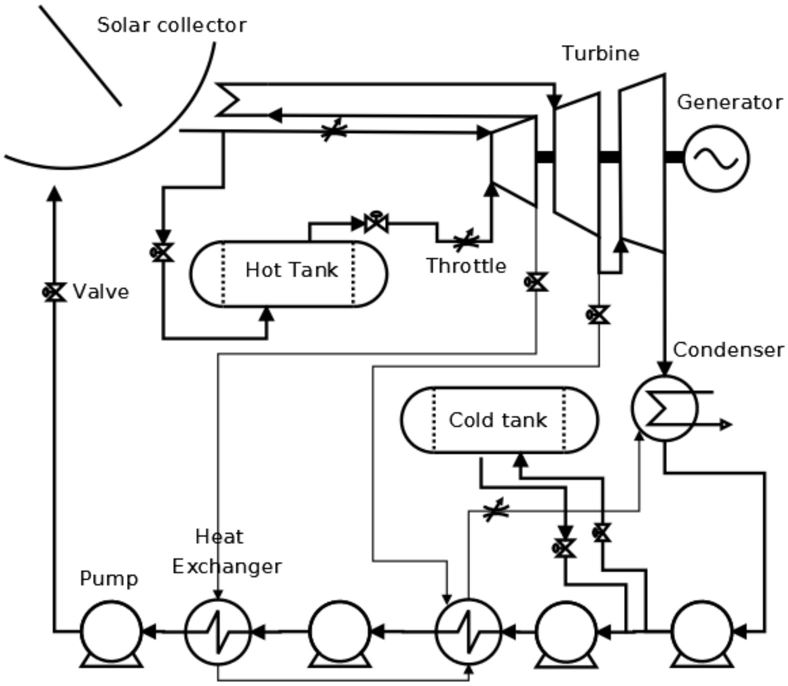

Fig. 3: System layout for dynamic model of a two-tank storage system utilising compressed liquid.

The results of a four-day simulation of a base load of 10 MW during a winter period are shown in Figure 4. The simulation starts during the night with useful stored energy with an overall storage volume of 1500 m³. The schematic for this system requires several days to become fully charged at the end of a day and the fluid properties have stabilised into the appropriate cycle and the results shown start once this has been reached. An initial 3-day period of start-up has been excluded from the results.

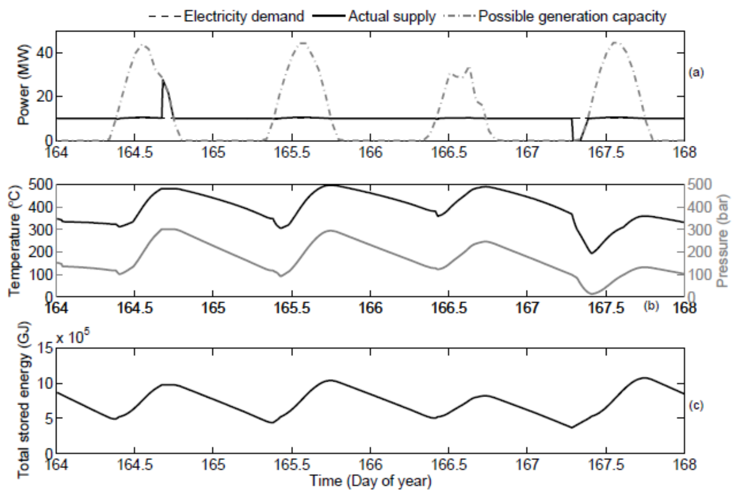

Fig. 4: Four-day simulation of winter period (13-16 June 1997) in Alice Springs. (a) power generated for a constant electricity demand, (b) storage temperature and pressure and (c) total stored energy.

This system is able to supply power during usual solar conditions but cannot supply base-load power continuously from storage after cloudy daytime conditions as seen in Figure 4 on the third day. The sudden drops in temperature and pressure seen at either the start of the next charging cycle or when the storage is empty, results from the cold water recharge as the steam condenses within the storage vessel.

Figure 5 shows the results of a five day simulation of a peak load of 10 MW during periods throughout the year. The schematic shown here starts during the night with no useful stored energy, with an overall storage volume of 300 m³ and a thermal capacity of 69 MWh. This volume was based on the ability to supply 100% of the required power for about four hours during the typical summer conditions and is almost double the ratio of thermal capacity to storage volume than the base-load scenario despite similar efficiencies.

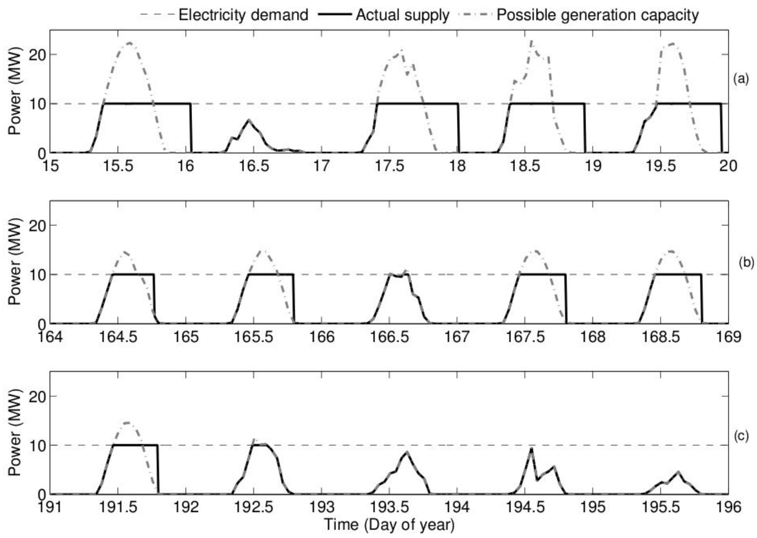

Fig. 5: Five-day peak-load simulation in Alice Springs showing power generated against a constant electricity demand. (a) summer days (15-19 January 1978) with a single cloudy day, (b) a winter period (13-17 June 1997) and (c) consecutive cloudy winter days (10-14 July 1980).

A peak-load regime has the capacity to produce sufficient energy from storage for several hours during summer, however, during winter, the storage can only function as a buffer system, if at all, as there is insufficient solar energy collected for both normal operation during the day and for storage.

The main issue found with a storage vessel of this type is the low quality steam that results at the exhaust of the turbine during the period that the storage is discharging. During normal operation using steam directly from the solar collectors attains high steam quality (>90%) at the turbine exhaust, steam flashed from storage exits at between 75% and 80% steam. This results from lack of capability to superheat and reheat the saturated steam after extraction from the storage vessel. Consequently saturated steam or slightly superheated steam, depending on current conditions in storage tank, is fed into the turbine and quickly starts to condense. Turbines exist that accept lower-quality steam, and might be more appropriate, but were not considered in the scope of this study.

A potential issue with applying a variable-pressure accumulator storage system is the effect of the drop in pressure in the storage vessel on the specific power output. As expected, there is a decreasing power output with pressure with a sudden decrease as the storage vessel becomes a liquid/vapour mix and for a pressure drop from 300 bar to 100 bar there is a decrease in the specific power output of 8.2% across the cycle.

CYCLE EFFICIENCIES AND EXERGY DESTRUCTION

The cycle efficiencies, both energetic and exergetic can be seen in Table 1. It should be noted also, aside from the losses caused by the storage, the general daytime cycle efficiency should improve based upon the cycle improvements implemented including, higher collector temperature, reheats and feedwater regeneration. Energetically, the results were very similar to the steady-state investigation, which was conducted as a first-base estimate, with quite high round-trip efficiencies seen. This is partly a result of the overall system, as no reheat and superheating is done using the storage cycle, and the consequences are seen in the low quality steam that results from the turbine exhaust. With respect to the exergetic efficiencies, compared to the efficiencies found from the steady-state model, there seems to be only a reduction of about 10% due to the dynamic nature of the storage. As a basis of comparison for this system, the PS10 system utilising a similar storage concept, as outlined in the Introduction, has a Rankine-cycle energetic efficiency of 30.75% and storage efficiency of 92.4%. Both of these figures are very close to those calculated in this study, with a Rankine cycle energetic efficiency of 31.6% and storage efficiency of 90.9%, but with a scale that allows 3-4 hours of operation after sunset.

Tab. 1: Energetic and exergetic efficiency of integrated storage systems

Efficiency	Energetic	Exergetic
Overall	32.9%	67.5%
Day cycle (no storage interaction)	31.6%	73.7%
Stored energy/exergy to power	30.3%	63.5%
Storage round-trip[1]	90.9%	

Table 2 shows the distribution of exergy with respect to the various thermodynamic processes. The most interesting and important result here is the exergy destruction that occurs within the storage vessel and during the exterior steam flashing. Comparison of the storage with the value of 92.3% found in Section 3 is important to this discussion. The variable-pressure accumulator, when isolated has an exergetic efficiency of 92.3%. This is in comparison to the losses in the storage vessel found in the whole system showing only a 0.2% difference. There are the extra losses due to flashing with this system, because at the high pressures that the storage operates, there is a need to flash the steam so the turbine limits are not exceeded.

Tab. 2: Exergy account of integrated storage system

	Process	Exergy account	Process	Exergy account
Outputs	Net power out	67.5%	Cooling water losses	4.2%
Exergy destruction	Turbine	8.7%	Various heat exchangers	2.4%
	Pump	0.4%	Storage vessel	7.5%
	Condenser	5.2%	External throttling	4.1%
Total	Input heat	100%		

[1] The storage round-trip is based upon the ratio between the power generated from storage to the power that could have been generated from the standard cycle operating had no storage occurred.

COST AND SIZING OF VESSELS

Through optimising the shape of the cylinder on the desire to minimise cost relative to the cost of steel and insulation, it was found that long thin cylinders allowed lowest weight of steel required for the pressure vessel with a radius of 1.5 m for the peak-load simulation and 2.3 m for the base-load simulation. However, these values are dependent on the cost of steel. Consequently, due to the large wall thickness required to store at these large pressures, the cost of storage becomes quite large. This capital cost is the main drawback of this type of storage, although containing a very low-cost storage media. However, if this concept is applied to an underground sealed cavern it may become one of the most cost-effective means of storage (Copeland & Ullman 1983).

CONCLUSIONS AND FURTHER WORK

This project involves the investigation of steam flashing thermal storage system and the effects of integrating this into a solar power plant. In particular the dynamic nature of the storage and the exergy destruction related to integrating the storage into the solar power plant is examined. The storage vessels can become large, especially for a base-load case. Nonetheless, compared to most other storage systems already constructed, the modelled peak-load system shows a good ratio of thermal capacity to storage volume. However pressurization of the storage vessel requires a significant quantity of steel and consequently the cost of storage becomes substantial compared to the total cost of the other plant equipment and therefore limits the viability of this type of storage concept. Based on the results in this project, variable-pressure accumulators show good efficiencies and when implemented for short term storage to cover peak loads into the night this appears to be the most cost effective method, which agrees with the general consensus that this type of storage system is very useful as a buffer system for cloudy transitions. The potential for a system to generate electricity from either type of storage decreases slightly over time as discharging of the storage occurs.

The internal steam flashing process results in a significant loss in exergy although relatively high round-trip efficiencies can be still be achieved. Turbines that can operate at high pressures relative to the storage pressure would also be beneficial to the efficiency of the system. Implementation of the full system, utilising the variable-pressure accumulator, may lead to problems with high moisture rates at the turbine exhaust. This problem would need to be addressed before this system could be fully implemented.

Further investigation could be done by analysing a composite storage system employing a steam accumulator, which produces saturated steam and which entails the majority of the energy, with a secondary storage for superheating. A comparatively small storage unit would be needed for superheating as less energy is required for superheating than for supplying the latent heat of vapourisation for steam.

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NOMENCLATURE

Cp	Specific heat capacity
E	Energy

h	Enthalpy
$h_g(p_{vessel})$	Enthalpy of steam at pressure p_{vessel}
m	Mass
p	Pressure
Q_{loss}	Heat loss
$T_{(sat)}(p_{start})$, $T_{(sat)}(p_{end})$	Saturation temperature at pressure p at start or end of time step
u	Internal energy
v	Specific volume
V	Volume
Δh_{fg}	Specific heat of vapourisation
Subscripts	
e	Exit
Liquid	State of variable for liquid phase in vessel
Start/end	State of variable at start/end of time step
Steam	State of steam added/removed during charging/discharging process
Vessel	State of variable for the fluid in vessel

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