A New Bifunctional Energy Storage Solution for Conventional and Renewable Energy Power Plants

Ahmad Arabkoohsar and Gorm B. Andresen

***Abstract*—In this work, a novel, simple and efficient power storage system that provides both heat and electricity in high efficiency is designed and proposed. The system works based on a gas turbine cycle in which the input energy is supplied by a thermal energy storage system instead of a combustion chamber. The other key point is that heat production is as important as power generation in this system. In order to prove the proficiency of the system, it is comprehensively analyzed thermodynamically in Denmark, regarding its power and heat market regulations and historical data as the case study of this work. The results show that an overall energy efficiency of almost 85% and the electricity efficiency of 35% can be expected for the system and it is demonstrated that the system is highly feasible economically even taking conservative electricity and heat prices into account.**

***Index Terms*—Energy storage, smart energy, thermal storage, wind power stabilization, energy market.**

I. Introduction

Battery storage is an efficient storage system for electrical energy. It emerged in the early 20th century and it is still widely used in the power production industry, especially in renewable energy source power plants [1]. The use of batteries is limited to short time scale applications because of their relatively high cost and some technical problems such as high rate of energy dissipation due to self-discharge losses [2]. Thus, alternative energy storage systems are of interest. In many of these systems, surplus electricity produced by power plants during low demand periods can be stored in different forms and it is later reclaimed in peak period to balance power supply and demand [3].

Even though some of the systems, especially compressed air energy storage and pumped hydroelectric energy storage systems, can be employed for large scale applications with relatively high efficiencies, the existence of new, reliable, cheaper and more efficient energy storage systems is still perceived, in particular, environmentally friendly systems that are not bound by geological constraints or limited natural resources [4]. Investigations indicate that an optimal energy storage system should enhance the reliability of renewable energy sources, improve the resilience of the grid and realize the benefits of smart grids [5]. In implementing an energy storage project, investigating the most appropriate type and size of the storage system, finding the most efficient operational strategy of the system and investigating the economic effectiveness of the storage system must be taken into account [6].

In this work, a novel, simple and efficient energy storage system is proposed and designed. This system is show-cased for Denmark where district heating is as important as power generation, and as a result, a considerable portion of the stored energy in low demand period is later, i.e. in generation mode of the energy storage system, used to support the local district heating system. Like conventional combined heat and power plants, the system is more efficient for locations with heating demand along with electrical energy deficit. In order to have a proper design, local power production of one of wind farms as well as the district heating system conditions of Aarhus city in Denmark have been taken into account as the case study of this work.

II. The Novel Energy Storage System

The schematic of this system is presented in Fig. 1. The concept is centered around a large thermal storage system, which is initially charged once up to the high temperature of 900 K. In the operational stage, the system may have charging or discharging process. In a charging process, surplus power output of e.g. wind turbines is simply used for heating the thermal energy storage medium up to higher temperatures using electrical coils or similar. The capacity of the storage should be so selected that a maximum temperature of no more than 950 K is reached over the year. In a discharging step, i.e. in case of power deficit in the system, the energy storage starts working to offset electricity ramps. In this case, the multistage turbine set, and consequently, the multistage compressor set that is coupled with it, is first driven by some prime-movers. The compressors then intake ambient air to produce hot and compressed air. This air then passes through a heat exchanger supported by hot air flow coming out from the thermal energy storage unit to be heated up to a temperature as high as possible. The heated compressed air system is expanded through the multistage turbines to produce rotational work that is utilized by an electricity generator for producing power. Considering the operational description given for the system so far, at first sight, it could be claimed that the general operational strategy of the system is based on the Bryton Cycle (gas turbine) and as it takes advantage of multistage turbines and compressors (isothermal processes instead of low efficiency adiabatic procedures) the system may look more like an Ericson Cycle. However, it should be noted that this system is practically different from both of them as it employs a thermal energy storage system (charged by surplus electricity of a power plant) as its energy supply source and it includes heat production, which is as important

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as power generation. In this system, the design is so that energy waste stands at its minimum level and efficiency is as high as possible.

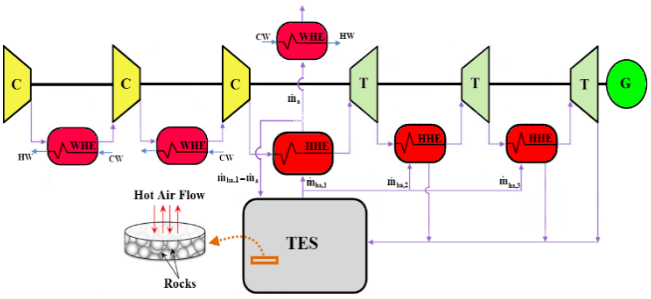

Fig. 1. Schematic of the proposed energy storage system; C: compressor, HE: heat exchanger, T: Turbine, G: electricity generator, TES: thermal energy storage system, CW: cold water, HW: how water.

Finally, it should be mentioned here that a rock storage has been chosen for the thermal storage part because: i) heat transfer between air and a rock bed is very good due to the very large heat transfer area and good heat conductivity of rocks, ii) its cost is relatively low in comparison with other large-scale, high-temperature storage methods, iii) rocks are nontoxic and abundant, iv) the long lifetime of rocks.

III. Thermodynamic Model of the Energy Storage

In this section, a detailed mathematical model of the proposed energy storage system is presented. In discharging step, the first operating component of the system is the compressors set. The total work of compressors set is calculated as:

$$\dot{W}_C = \sum_{j=1}^{n} (\dot{m}_a w_c) \tag{1}$$

In which, n, $\dot{m}_a$ and w_c are respectively the number of compressor stages, air mass flow rate through each compressor stage and its corresponding work. Considering an adiabatic process for each compressor stage, its specific work is then calculated using the first law of thermodynamics as:

$$w_c = c_{p,a}(T_{c,i} - T_{c,e}) \tag{2}$$

The second component of this configuration is the intercooling heat exchanger between different stages of compressors. The heat extracted from the air stream through these heat exchangers is utilized for district heating use. For these heat exchangers, the inlet air temperature ($T_{a,i}$) is equal to the same stage compressor outlet air temperature. The outlet air temperature of the intercooler heat exchanger ($T_{a,e}$) is calculated by [7]:

$$T_{a,e} = T_{a,i}(1 - \varepsilon) + \varepsilon\, T_{w,i} \tag{3}$$

In which, $T_{w,i}$ is the heat exchanger inlet water temperature and ε is the heat exchanger effectiveness. Thus, the heat rejected from the air stream that is absorbed by the water stream and injected to district heating system, is calculated by:

$$Q_{he} = \dot{m}_a c_{p,a}(T_{a,i} - T_{a,e}) \tag{4}$$

Note that the same formulation, but with their own specific conditions, applies for the other heat exchangers in the system. However, for heating heat exchangers before the turbines, there are some other factors that should be calculated. For these heat exchangers, the mass flow rate of hot air outgoing from the thermal energy storage ($\dot{m}_{ha}$) is given by:

$$\dot{m}_{ha} = \frac{\dot{m}_a(T_{a,e} - T_{a,i})}{(T_{ha,i} - T_{ha,e})} \tag{5}$$

In this equation, $T_{ha,i}$ is equal to the thermal energy storage temperature and $T_{ha,e}$ is the hot air outlet temperature from the heating heat exchangers calculable as [8]:

$$T_{ha,e} = T_{ha,i} - \frac{\varepsilon\, C_{min}(T_{ha,i} - T_{a,i})}{\dot{m}_{ha} c_{p,ha}} \tag{6}$$

where, C_{min} is the lower value of heat capacity between the two fluids through the heat exchanger and $c_{p,ha}$ is the specific thermal capacity of heating air in constant pressure.

For the turbines, the inlet air temperature is set on a constant and specific value. The heat required to increase the airflow temperature up to this value is provided with the heating heat exchangers placed before each stage of expansion. Considering an adiabatic process for each turbine, its outlet temperature is calculated as:

$$T_{t,e} = T_{t,i}\left[1 + \eta_{t,s}\left(r_t^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right)\right] \tag{7}$$

Each turbine network and the total work of turbine set are respectively calculated by:

$$w_t = h_{t,i} - h_{t,e} = c_{p,a}(T_{t,i} - T_{t,e}) \tag{8}$$

$$\dot{W}_t = \sum_{j=1}^{n} (\dot{m}_a w_t) \tag{9}$$

where, the mass flow rate of air through the turbines should be equal to:

$$\dot{m}_a = \frac{P_d}{w_t - w_c} \tag{10}$$

The total mass of reservoir is the summation of air and rocks masses and can be calculated by:

$$m_{tes} = m_r + m_a = \rho_r V_r + \frac{P_a V_a}{R T_a} \tag{11}$$

In this equation, m_r and ma are the mass of rock and air within the cavern. ρ_r, V_r, P_a, T_a, V_a and R are respectively the rocks density, the portion of cavern occupied by rocks, air pressure and temperature, the portion of cavern filled by air and the gas constant. Based on the first law of thermodynamics, the reservoir air temperature, in charging and discharging modes, is respectively computed by the following equations:

$$T_{tes}^{t+1} = T_{tes}^{t} + \frac{P_s}{m_{tes}^{t} c_{p,a}} \quad (12)$$

$$T_{tes}^{t+1} = \frac{\left[\begin{array}{c} E_{tes}^{t} - \sum \dot{m}_{ha,i}^{t} \left(h_{tes}^{t} - h_{ra,i}^{t}\right) \\ + \dot{m}_{ha,1}^{t} h_{l}^{t} - \dot{E}_{l}^{t} \end{array}\right]}{m_{tes}^{t+1} c_{v,a}} \quad (13)$$

where, P_s is the surplus power available used for heating the air within the thermal energy storage system. $h_{ra,i}$ is the enthalpy of returning air to the storage cavern after passing through the heating heat exchangers, h_l^t is the last stage turbine outlet enthalpy and $\dot{E}_l^t$ is the rate of heat losses from the storage unit.

In the end, 'the overall energy efficiency' of the system is defined as the ratio of the energy gained to that spent in the system (heat and electricity) while 'the electricity efficiency' is defined as the electricity output to the electricity input of the system.

IV. Results and Discussions

In this section, the results of simulations on the system are presented and discussed. Fig. 2 illustrates the trend of change in the overall energy efficiency of the energy storage system for various numbers of compression/expansion stages and various pressure ratios for both turbines and compressors.

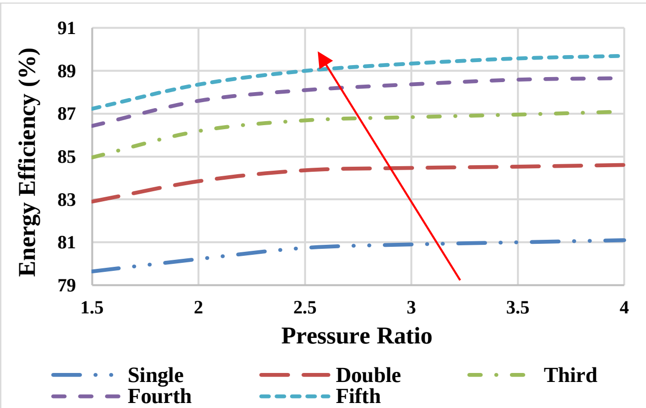

Fig. 2. Variation of overall energy efficiency of the energy storage system by increasing the pressure ratio for different numbers of compression/expansion stages.

As the figure shows, regardless the number of stages and even the pressure ratio, the overall energy efficiency of the system is high and this is why this system can be considered as a very efficient energy storage system. According to the figure, as the pressure ratio increases, energy efficiency of the system increases too, although the rate of enhancement, decreases for higher pressure ratios. For a fixed pressure ratio, increasing the number of stages picks up the energy efficiency. For higher compressor and turbine stage numbers, the efficiency improvement rate falls considerably. Overall, the maximum energy efficiency that can be expected from the system is about 90% for a pressure ratio equal to 4 and number of stages of 5.

Fig. 3 shows the trend of electricity efficiency change for increasing pressure ratios of compressors and turbines for different numbers of stages of compressor and turbine sets.

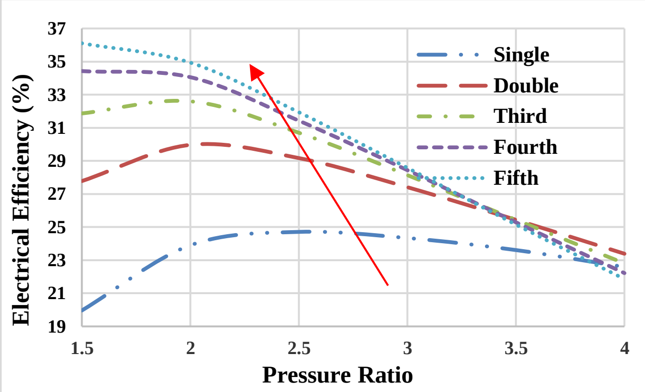

Fig. 3. Variation of electrical efficiency of the energy storage system by changing the pressure ratio for various numbers of compression/expansion stages.

According to this figure, increasing the number of stages improves the system electricity efficiency as well, regardless of the pressure ratio value. For variation of pressure ratio, in contrast with overall energy efficiency, electricity efficiency of the system decreases by increasing the pressure ratio, except for single and double stage systems for which the electricity efficiency falls when pressure ratio decreases from 2 to 1.5. As the figure shows, a maximum energy efficiency of almost 36% is expected for a system having 5 stages and pressure ratio of 1.5 and the lowest electricity efficiency of around 20% is obtained for pressure ratio of 4. For higher pressure ratios, the electricity efficiency decreases for all the cases. It should be noted that for a specific number of stages, lower pressure ratios limit the power output of the system and thus, opting higher pressure ratios may be inevitable sometimes. On the other hand, increasing the numbers of stages for a specific range of pressure and increasing the pressure ratio of the compressors and turbines for a specific number of stages increase the capital and O&M costs of the system.

In order to have a comprehensive analysis of the system, its performance is assessed for a whole day charging and discharging with sample values (10 MWe in charging and 5 MWe in discharging) and the results are presented hereunder. For having dynamic results, ambient temperature which is an effective parameter on the system performance is assumed to fluctuate sharply during this day.

Fig. 4 shows the storage cavern temperature and total internal energy variation over the charging and discharging processes. As can be seen, the storage capacity has been so chosen that it meets a peak temperature of 950 K by 12 h full load charging. This storage capacity is 4000 m^3. The graph shows that due to the low electrical efficiency of the system, all of the stored energy is discharged from the system after about 7.5 h.

Fig. 5 illustrates the total compressor and turbine work in the system over time. Naturally, this figure only includes information for the discharging step as these devices are in stand-by status while the cavern is being charged. Note that the amount of power produced by the system is equal to the network of the system (obtained from subtracting compressor consuming work from turbine output work which is going to be 5 MW here) multiplied by the generator efficiency (considered equal to 90% here).

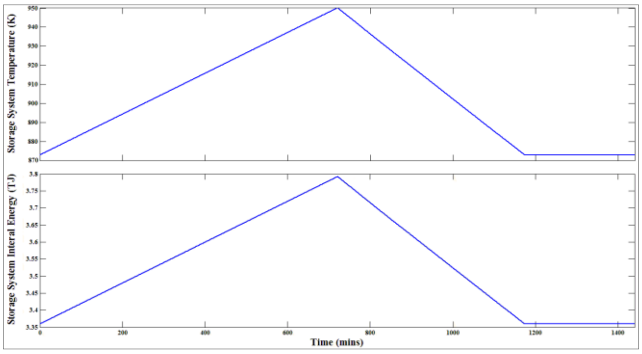

Fig. 4. Variation of energy storage cavern temperature and total energy over time.

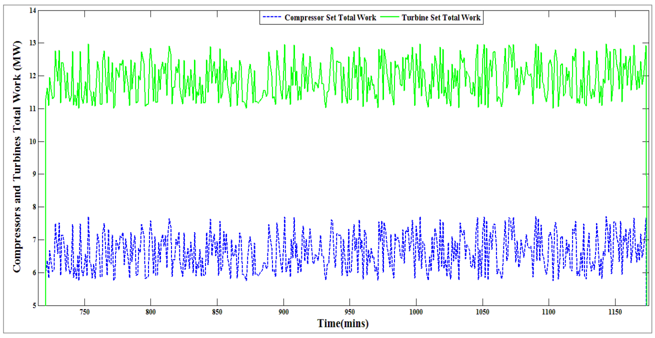

Fig. 5. Turbine and compressor sets total work during discharging process.

V. Conclusion

This study proposes a novel, creative and simple energy storage system suitable for locations with district heating networks as well as electricity distribution grids and therefore, providing centralized heat is as important as electricity. The system is cheap in comparison with other energy storage systems, can provide both electricity and heat in high efficiencies as high as around 80-90% and its electricity production efficiency is still in acceptable level of almost 25-35%. In this work, the concept was described in details and various effective parameters on the performance of the proposed system, such as the number of stages in compression and expansion parts, the pressure ratio of these devices, maximum temperature in the storage reservoir as well as ambient temperature variation, were considered and assessed and the most efficient configuration was designed taking into account the energy efficiency and economic outcomes. The performance of this sample designed system was simulated for a whole day charging and discharging cycle with a constant energy input of 10 MW and constant electricity output of 5 MW with random made ambient temperatures between -5 ^{o}C and 30 ^{o}C and the results were presented and discussed.

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