Thermodynamic Performance of Molten Salt Heat Storage System Used for Regulating Load and Supplying High Temperature Steam in Coal-Fired Cogeneration Power Plants

Haihua Luo1, Qiang Shen2, Yunfei Chen3*, Shien Sun1, Junguang Lin1, and Houlei Zhang3

1Zhejiang Zheneng Technology Institute, 311100 Hangzhou, China
2Zhejiang Zheneng Jiaxing Power Plant Co., Ltd., 314000 Jiaxing, China
3School of Energy and Power Engineering, Nanjing University of Science and Technology, 210094 Nanjing, China

Abstract. In order to accept more electricity from renewable energy, cogeneration power plants are considering to reduce electricity production, which affects the heat supply. Here we present a molten salt heat storage system for coal-fired cogeneration power plants, which can supply high temperature steam to users and decouple the heat and electricity production. The first and second law-based analytical models for the cycle and a real device are built. Two water input methods are taken into account. The results show that the high and low temperatures in the two molten salt tanks influence the design of the components and the entropy generation distribution significantly. The pinch temperature difference in the discharge duration limits the lowest molten salt temperature. The device with real heat exchangers produces higher entropy generation and lower second law efficiency. Environmental water input requires more heat and entropy generation for the same steam supply. Recommendations are provided for practical designs.

1 Introduction

Renewable energy is more and more widely utilized in power generation. Many coal-fired power plants in operation are considering (or required) to reduce their load output in order to accept more electricity from renewable energy [1, 2]. For cogeneration power plants with coupled electricity and heat outputs, there exist two problems. On one side, the electricity production cannot be dropped too much for ensuring the safety of the equipment and high efficiency. This limits the amount of the acceptable renewable electricity by the grid or the peak regulation capability. On the other side, when the electricity load decreases, the corresponding heat output may not be enough to fulfill the requirement of the heat users. To address these problems, some new operation strategies and new techniques have been explored and studied. Among them, heat storage becomes an attractive option.

Recently, Nuytten et al. [3] reported the flexibility of a combined heat and power system with a water reservoir heat storage for district heating. Garbrecht et al. [4] presented a case study by simulation for two fossil power plants: a heat-controlled waste incinerating facility and a power-controlled sub-critical lignite power plant. The simulations confirmed the technical feasibility. Li and Wang [5] studied a 600 MW supercritical coal-fired power plant with high temperature salt phase change heat storage. Their simulation results show that the design has faster dynamic response to the load demand changes and the grid frequency services. Zhao et al. [6] compared different flexibility enhancement methods for coal-fired combined heat and power plants and concluded that the design with heat storage has advantages over the heat-only boiler in coal-saving potential and economic performance. Wang et al. [7] investigated the peak shaving range of a heat storage-based unit based on Aspen Plus simulation. The case study of a 300 MW unit shows that the peak regulation capacity is increased significantly. Richter et al. [8] analyzed the effect of a steam accumulator on the load flexibility of coal-fired power plant. The simulation research shows that the load flexibility is improved. Cao et al. [9] proposed an approach for improving load flexibility of coal-fired power plants. High temperature heat storage through additional thermodynamic cycle is incorporated into the conventional system. A mixture of LiCl and KCl is selected as the storage material. The case study of a 600 MW supercritical coal-fired power plant shows that the approach is feasible. Up to know, most of the work was simulation-based studies, and high temperature steam supply was not involved.

In this paper, we focus on a cogeneration power plant with molten salt heat storage. Here, the heat is supplied to external industrial users which require higher pressure and temperature than normal building heating, which was...
not explored in literature. We first describe the configuration of the heat storage system, and then build thermodynamic models for the cycle and a real device. Based on the analytical results, we may determine the thermodynamic performance and identify the irreversibility distribution for the heat storage system which forms the basis of further design optimization.

2 Configuration of the heat storage system

Consider a sub-critical cogeneration power plant with industrial steam output to heat users (with pressure \( P \approx 1.5 \) MPa and temperature \( T \approx 250^\circ C \)). Figure 1 shows the scheme of the heat storage system for industrial steam supply. In normal mode without the heat storage, the steam at the outlet of the high pressure turbine (State 1) is extracted and its pressure and temperature are reduced through the component C to the values specified by the users (D). In this process, irreversibility (entropy generation, \( S_g \), or entropy generation rate \( \dot{S}_g \)) is unavoidable. When the electricity load of the power plant is lowered due to the consideration of more time-dependent renewable electricity, the heat load may not fulfill the need of the users. As a solution, the heat storage mode works.

The operation consists of two durations. During the charging duration with time \( t_c \), the steam at the outlet of the reheater (State 5) is extracted to heat the molten salt from \( T_L \) (State 7) to \( T_H \) (State 8) via the heat exchanger F. The thermal energy is then stored in the high temperature molten salt tank (HTMST). During the discharging duration (usually at night, with time \( t_d \)), the hot molten salt in HTMST is driven to heat the water coming from the deaerator (State 13) to the specified state (State 17) via three heat exchangers, i.e., preheater, evaporator and superheater.

Although from the viewpoint of the first law of thermodynamics, the heat storage method does not conserve energy theoretically, it lowers the minimum load of electricity production and ensures the heat supply simultaneously. Heat supply and electricity production are therefore decoupled, and flexible operation becomes available. Besides, the method does not involve the retrofit of the turbines and the boiler, which is an advantage for those old power plants.

Here we use a sub-critical cogeneration power plant as an example. The related parameters are listed in Table 1, where the subscripts correspond to the states in Figure 1.

3 Model

3.1 Thermodynamic cycle model

For the thermodynamic cycle analysis, we neglect the heat loss and the pressure drop of the components (the heat exchangers, molten salt tanks and connection pipes). Based on the first and second laws of thermodynamics, the model equations are obtained as follows.

\[
\dot{Q} = \dot{m}_i (h_i - h_o) = \dot{m}_i (h_i - h_o) \quad i = F, I, J, K
\]

\[
\dot{W}_j = \frac{\dot{m}_j \Delta P}{\rho \eta_D} \quad j = 1, 2, 3
\]

\[
Q_{tot} = Q_c + \dot{Q}_t_e = Q_2 = (\dot{Q}_k + \dot{Q}_j + \dot{Q}_I) t_d
\]

\[
Q_{out} = M_{wat} (h_i - h_o)
\]

\[
\eta = \frac{Q_i}{Q_i + (\dot{W}_{j2} + \dot{W}_{j3}) t_c + \dot{W}_{j2} t_c}
\]

\[
\dot{S}_g = \dot{m}_i (s_{i} - s_o) + \dot{m}_i (s_{co} - s_{o}) \quad i = F, I, J, K
\]

\[
\dot{S}_{gj} = \dot{m}_j (s_{j, out} - s_{j, in}) \quad j = 1, 2, 3
\]

\[
S_{gout} = S_{gj} + S_{gk} = \dot{S}_{gj} t_c + (\dot{S}_{gj} + \dot{S}_{gk} + \dot{S}_{gk}) t_d
\]
\[ \eta_{II} = 1 - \frac{T_{IC} S_{\text{total}}}{(\dot{E}x - \dot{E}x_0)_{T_1} + W_{P1} + (W_{P2} + W_{P3})_{T_d}} \]  

(9)

In Equations 1-9, \( \dot{Q} \), \( \dot{W} \) and \( \dot{E}x \) are heat transfer rate, pumping power and exergy rate, respectively; \( \dot{Q} \) is total heat transfer flux in one duration; \( \dot{m} \) is mass flow rate; \( M \) is mass; \( AP \) is pressure drop; \( h \) and \( s \) are specific enthalpy and entropy; \( \eta_1 = 0.7 \). \( \eta \) and \( \eta_{II} \) are pump efficiency, the first law efficiency and the second law efficiency (i.e., exergy efficiency), respectively; \( T_0 \) represents the surrounding temperature (298 K). Subscripts \( h \) and \( c \) represent hot and cold sides, subscripts \( i \) and \( o \) represent inlet and outlet, and other subscripts correspond to the states or components in Figure 1. In cycle analysis of Section 3.1, as we neglect the pressure drop of the components, \( \dot{W}_{P1} \) and \( \dot{W}_{P2} = 0 \).

In this paper, HITEC salt (53% KNO\(_3\)-40% NaNO\(_2\)-7%NaNO\(_3\), wt%) with melting temperature 142°C is selected as the heat storage material and the heat transfer fluid with temperature-dependent properties [10, 11]. An EES (Engineering Equation Solver) program is compiled to solve the above model. The properties of water are calculated by EES.

### 3.2 Steady device model

In this section, we analyse a case study with real heat exchangers under the specified conditions: \( T_{HI} = 420\, ^\circ\text{C} \), \( T_L = 190\, ^\circ\text{C} \) and total heat load \( Q_{\text{total}} = 350.5 \) MWh. The temperature at State 6 is kept the same as that in the cycle analysis. ASPEN EDR software [12] is used to design the thermodynamic performance with the pressure drop of the components taken into account.

### 4 Results and discussion

#### 4.1 Thermodynamic cycle analysis

We start the cycle analysis from \( T_{HI} = 420\, ^\circ\text{C} \) and \( T_L = 190\, ^\circ\text{C} \) for specified heating load \( Q_{\text{total}} = 350.5 \) MWh and fixed \( t_{H} = 18 \) h and \( t_d = 6 \) h. The cycle analysis results are shown in Table 2. We see that the evaporator J absorbs much more heat than I and K, and also produces more entropy generation rate. The second law efficiency \( \eta_{II} \) is significantly lower than the first law efficiency \( \eta_1 \).

#### 4.2 Molten salt pumps

The power consumption of the two molten salt pumps (not zero now) is calculated by Equation 2. Similar to Section 3.1, an EES program is compiled to calculate the

### Table 2. Cycle analysis results.

| Item | Unit | Value | Item | Unit | Value |
|------|------|-------|------|------|-------|
| \( T_{HI} \) | °C | 420 | \( Q_{\text{total}} \) | MW | 350.5 |
| \( T_L \) | °C | 190 | \( Q_{\text{total}} \) | MW | 4.61 |
| \( T_5 \) | °C | 289.76 | \( W_{P3} \) | kW | 35.24 |
| \( T_6 \) | °C | 205.7 | | | |
| \( T_7 \) | °C | 196.1 | \( S_{gf} \) | kW/K | 5.59 |
| \( m_5 \) | t/h | 130.1 | | | |
| \( m_7 \) | t/h | 152.7 | \( S_{gd} \) | kW/K | 2.67 |
| \( m_9 \) | t/h | 458.2 | \( S_{gd} \) | kW/K | 17.35 |
| \( m_{13} \) | t/h | 88.81 | | | |
| \( Q_{\text{rel}} \) | MW | 19.47 | \( S_{\text{total}} \) | MJ/K | 805.8 |
| \( Q_{\text{eff}} \) | MW | 5.58 | \( \eta_{II} \) | % | 65.92 |
| \( \eta \) | % | 99.94 | | | |

Figure 2 shows the effects of \( T_{HI} \) on the cycle performance. In Figure 2a, the two molten salt flow rates \( m_7 \) and \( m_9 \) decrease with the increase of \( T_{HI} \), which requires smaller molten salt pumps. Another advantage of higher \( T_{HI} \) is less molten salt mass \( (M_{\text{mol}}) \), which corresponds to larger heat storage density or smaller molten salt tank size (or investment). When \( T_{HI} \) increases, the average temperature difference of the heat exchanger F decreases and larger heat transfer area is required. Figure 2b documents the entropy generation of F, I, J, K and P3. With the increase of \( T_{HI} \), the entropy generation rate of F drops, while \( S_{gf} \) and \( S_{gd} \) increase very slightly. We also see that the irreversibility of the water pump is small. Figure 2c shows the variation of the irreversibility distribution between the charging duration and the discharging duration. To summarize, \( T_{HI} \) affects both the design of the components and the irreversibility distribution, but does not change \( \eta_1 \) and \( \eta_{II} \).
Figure 3 shows the effects of $T_L$ on the cycle performance. Seen in Figure 3a, with the decrease of $T_L$, the flow rates $\dot{m}_7$ and $\dot{m}_9$ and the molten salt mass $M_{\text{salt}}$ decrease, similar to the influence of increasing $T_H$. From Figure 3b, we see the variation of $S_{gF}$, $S_{gI}$, $S_{gJ}$ and $S_{gK}$ with $T_L$, while the total entropy generation rate $S_{\text{total}}$ keeps constant (Figure 3c). Same as $T_H$, $T_L$ does not influence $\eta_I$ and $\eta_{II}$. In brief, ($T_H - T_L$) affects the design or selection of the heat exchangers and the pumps, as well as the entropy generation distribution. Another point in selecting $T_L$ is the limitation of pinch temperature difference between State 11 and State 15 ($\Delta T_{11-15}$). One case is shown in Figure 4, where $\Delta T_{11-15} = 9.6^\circ\text{C}$. If we set $\Delta T_{11-15} = 5^\circ\text{C}$, the lowest $T_L$ will be 185.1$^\circ\text{C}$, 182.6$^\circ\text{C}$ and 180.1$^\circ\text{C}$ for $T_H = 420^\circ\text{C}$, 460$^\circ\text{C}$ and 500$^\circ\text{C}$, respectively. In real designs, trade-off or optimization is necessary considering both the cost of the heat exchangers and that of the molten salt tanks.
In this paper, we presented the thermodynamic performance of molten salt heat storage system for coal-

Table 5 shows a case study with real heat exchangers and assumed pressure drops of the tanks and connection pipes. Both $\eta_I$ and $\eta_{II}$ of the device are lower than that of the cycle without considering the pressure drops of the components. The total entropy generation of the device is 12.3% higher than that of the cycle. The entropy generation distribution has slight change compared with that in the cycle analysis. It should be noted that although we take the real heat exchangers into account, the heat loss of the components is neglected. The analysis for the device is still approximate.

Table 5. A case study with real heat exchangers.

| Item | Unit | Value | Item | Unit | Value |
|------|------|-------|------|------|-------|
| $T_{11}$ | °C | 400 | $Q_{total}$ | MWh | 430.5 |
| $T_{13}$ | °C | 395.8 | $S_{gd,pipe}$ | kW/K | 0.047 |
| $\dot{m}_{13}$ | t/h | 88.81 | $S_{gd}$ | MJ/K | 737.3 |
| $\dot{m}_{7}$ | t/h | 187.6 | $S_{gP3}$ | kW/K | 0.06 |
| $\dot{m}_{9}$ | t/h | 562.9 | $S_{gc}$ | MJ/K | 444.9 |
| $\dot{m}_{5}$ | t/h | 159.9 | $S_{gK}$ | kW/K | 10.66 |
| $\dot{m}_{7}$ | t/h | 131.56 | $S_{gd}$ | kW/K | 0.006 |
| $\dot{m}_{9}$ | t/h | 458.2 | $S_{gP3}$ | kW/K | 0.027 |
| $\dot{m}_{13}$ | t/h | 88.81 | $S_{gc,pipe}$ | kW/K | 0.047 |
| $\dot{W}_{P1}$ | kW | 3.23 | $S_{gd,pipe}$ | kW/K | 0.152 |
| $\dot{W}_{P2}$ | kW | 26.23 | $S_{gc}$ | MJ/K | 456.9 |
| $\dot{W}_{P3}$ | kW | 38.56 | $S_{gd}$ | MJ/K | 448.2 |
| $Q_F$ | MW | 19.47 | $S_{gtotal}$ | MJ/K | 905.1 |
| $\dot{Q}_I$ | MW | 5.55 | $\eta_{II}$ | % | 63.2 |
| $\eta_I$ | % | 99.93 | $\eta_II$ | % | 59.32 |
| $T_{14}$ | °C | 197.8 | $\dot{m}_{5}$ | t/h | 131.56 |
| $T_{16}$ | °C | 198.5 | $\dot{m}_{7}$ | t/h | 159.9 |
| $T_{15}$ | °C | 289.76 | $\dot{m}_{9}$ | t/h | 562.9 |
| $T_{10}$ | °C | 241.2 | $\dot{m}_{13}$ | t/h | 88.81 |
| $T_{11}$ | °C | 289.76 | $\dot{Q}_F$ | MW | 23.92 |
| $T_{12}$ | °C | 196.1 | $\dot{Q}_I$ | MW | 5.58 |

4.2 Steady device analysis

Section 4.1 does not involve the structure and size (or the design) of the heat exchangers. Here for specified heating load $Q_{total} = 350.5$ MWh, we select $T_{II} = 420^\circ C$ and $T_{I} = 190^\circ C$ in the device design to perform a case study. Usually, when the temperature is lower than $420^\circ C$, pipe 1 and pipe 2 are adopted as 0.2 MPa approximately.

Table 4. Design of heat exchangers.

| Heat exchanger | F | I | J | K |
|----------------|---|---|---|---|
| Type           | BEU | BEU | BEU | BEU |
| Material       | SS 347 | Carbon Steel | BEU | BEU |
| Shell OD/mm    | 1000 | 800 | 1800 | 800 |
| Tube OD/mm     | 25  | 25  | 25  | 25  |
| Tube length/m  | 3   | 1.5 | 6   | 4   |
| No. of tubes   | 1400| 900 | 2200| 400 |
| Tube arrangement | 30$^\circ$ | 30$^\circ$ | 30$^\circ$ | 30$^\circ$ |
| Baffle         | Single segmental | Single segmental | Single segmental | Single segmental |
| Baffle cut     | 25% | 25% | 25% | 25% |
| Baffle spacing/mm | 300 | 400 | 520 | 300 |
| Shells in series | 2 | 1 | 1 | 2 |
| No. of passes  | 2  | 2  | 4  | 4  |
| Area/m$^2$     | 695 | 115.6 | 1099.4 | 263.1 |

5 Conclusions

In this paper, we presented the thermodynamic performance of molten salt heat storage system for coal-

Table 3. Cycle analysis with environmental water input.

| Item          | Unit | Value | Item | Unit | Value |
|---------------|------|-------|------|------|-------|
| $T_{14}$      | °C   | 420   | $\dot{Q}_I$ | MW | 48.24 |
| $T_{15}$      | °C   | 190   | $\dot{Q}_J$ | MW | 17.95 |
| $T_{16}$      | °C   | 289.76| $Q_{total}$ | MWh | 430.6 |
| $T_{17}$      | °C   | 400   | $W_{P1}$ | kW | 51.51 |
| $T_{18}$      | °C   | 241.2 | $S_{gF}$ | kW/K | 6.87 |
| $T_{19}$      | °C   | 196.1 | $S_{gJ}$ | kW/K | 2.7  |
| $T_{20}$      | °C   | 196.1 | $S_{gK}$ | kW/K | 20.72 |
| $\dot{m}_{5}$ | t/h  | 159.9 | $S_{gK}$ | kW/K | 10.66 |
| $\dot{m}_{7}$ | t/h  | 187.6 | $S_{gP3}$ | kW/K | 0.06 |
| $\dot{m}_{9}$ | t/h  | 562.9 | $S_{gc}$ | MJ/K | 444.9 |
| $\dot{m}_{13}$| t/h  | 88.81 | $S_{gd}$ | MJ/K | 737.3 |
| $\dot{Q}_F$   | MW   | 23.92 | $S_{gtotal}$ | MJ/K | 1182 |
| $\dot{Q}_I$   | MW   | 5.58  | $\eta_{II}$ | % | 59.32 |
| $\eta_I$      | %    | 99.93 | $\eta_{II}$ | % | 59.32 |
fired cogeneration power plants, which can supply high temperature steam to users and decouple the heat and electricity production. The total entropy generation and the irreversibility distribution are documented. The two molten salt temperatures ($T_H$ and $T_L$) influence the design of the components and the entropy generation distribution, and proper parameters are recommended. The pinch temperature difference in the discharge duration is also discussed, which, as well as the solidification temperature of the molten salt, limits the lowest $T_L$. A case study with real heat exchangers is performed, which corresponds to higher entropy generation and lower second law efficiency. Instead of supplying water from the deaerator, environmental water input is also possible, which requires more heat and entropy generation for the same steam supply.

It should be reminded that the present work only focuses on the heat storage part itself, and does not evaluate the comprehensive influence of the incorporation of the heat storage on the overall power plant, which is the future work of us.

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