

Article

Numerical Simulation of Natural Gas Waste Heat Recovery Through Hydrated Salt Particle Desorption in a Full-Size Moving Bed

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Abstract

To achieve energy conservation, emission reduction, and green low-carbon goals for gas storage facilities, it is crucial to efficiently recover and utilize waste heat during gas injection while maintaining natural gas cooling rates. However, existing sensible and latent heat storage technologies cannot sustain long-term thermal storage or seasonal utilization of waste heat. Thermal chemical energy storage, with its high energy density and low thermal loss during prolonged storage, offers an effective solution for efficient recovery and long-term storage of waste heat in gas storage facilities. This study proposes a novel heat recovery method by combining a moving bed with mixed hydrated salts ($\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$). By constructing both small-scale and full-scale three-dimensional models in Fluent, which couple the desorption and endothermic processes of hydrated salts, the study analyzes the temperature and flow fields within the moving bed during heat exchange, thereby verifying the feasibility of this approach. Furthermore, the effects of key parameters, including the inlet temperatures of hydrated salt particles and natural gas, flow velocity, and mass flow ratio on critical performance indicators such as the outlet temperatures of natural gas and hydrated salts, the overall heat transfer coefficient, the waste heat recovery efficiency, and the mass fraction of hydrated salt desorption are systematically investigated. The results indicate that in the small-scale model ($1164 \times 312 \times 49 \text{ mm}$) the outlet temperatures of natural gas and mixed hydrated salts are 79.8°C and 49.3°C , respectively, with a waste heat recovery efficiency of only 33.6%. This low recovery rate is primarily due to the insufficient residence time of high-velocity natural gas ($10.5 \text{ m}\cdot\text{s}^{-1}$) and hydrated salt particles ($2 \text{ mm}\cdot\text{s}^{-1}$) in the moving bed, which limits heat exchange efficiency. In contrast, the full-scale moving bed ($3000 \times 1500 \times 90 \text{ mm}$) not only accounts for variations in natural gas inlet temperature during the three-stage compression process but also allows for optimized operational adjustments. These optimizations ensure a natural gas outlet temperature of 41.3°C , a hydrated salt outlet temperature of 82.5°C , a significantly improved waste heat recovery efficiency of 94.2%, and a hydrated salt desorption mass fraction of 69.2%. This configuration enhances the safety of the gas injection system while maximizing both natural gas waste heat recovery and the efficient utilization of mixed hydrated salts. These findings provide essential theoretical guidance and data support for the effective recovery and seasonal utilization of waste heat in gas storage reservoirs.

Keywords: hydrated salts; moving bed; seasonal heat storage; natural gas reservoir



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1. Introduction

Underground gas storage facilities are essential infrastructure in the natural gas industry, primarily used to store and regulate supply in response to fluctuating demand. In northern China, for instance, natural gas consumption in winter can exceed summer levels by more than threefold [1]. Due to their large capacity, cost-effectiveness, and operational safety, underground storage facilities help balance seasonal supply and demand. By 2023, China had established 33 operational underground gas storage sites, reaching a peak-shaving capacity of 23 billion cubic meters, playing a critical role in national energy security [2]. These facilities typically operate on an annual cycle—injecting gas in summer and autumn and extracting it in winter and spring. During injection, natural gas is compressed in multiple stages, raising its temperature to around 100 °C and generating substantial low-grade waste heat. Currently, to ensure compression efficiency and system safety, air coolers are commonly used to reduce the gas temperature below 40 °C, with no heat recovery implemented [3]. However, air cooling systems are costly, noisy, and occupy significant space. To support energy conservation and carbon reduction goals, there is an urgent need to recover this waste heat efficiently without compromising gas cooling requirements.

Conventional methods for low-grade heat recovery include heating, cooling, and power generation [4–6]. However, gas storage facilities are often located in remote areas far from urban centers, making it difficult to utilize waste heat during the summer injection season through traditional means. In contrast, there is significant demand for low-grade heat in winter—for preheating gas prior to throttling (to prevent pipeline blockage) and for residential heating. Therefore, storing the heat recovered during injection for use in the extraction season is key to unlocking the full value of seasonal waste heat. Traditional sensible and latent heat storage technologies are generally unsuitable for long-term storage due to limited duration [7–9]. In contrast, thermochemical heat storage based on hydrated salt desorption offers high energy density, minimal thermal losses, and low cost—making it ideal for long-duration applications [10,11]. For instance, $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$ dehydrates gradually when heated to 30–100 °C, forming $\text{MgSO}_4 \cdot 2\text{H}_2\text{O}$ and absorbing a significant amount of heat (about 1100 J/g—five times the latent heat of paraffin) [12,13]. Likewise, $\text{MgSO}_4 \cdot 2\text{H}_2\text{O}$ reacts with water vapor to form $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$, releasing substantial heat, with exothermic temperatures between 55 °C and 70 °C [14,15].

Research on thermochemical storage using hydrated salts has primarily focused on developing high-performance composite materials by incorporating porous substrates such as silica gel, diatomite, activated carbon, vermiculite, and expanded graphite [16]. For example, Liu et al. [17] developed a shale- CaCl_2 composite with a theoretical energy density of 75.6 kWh/m³, which maintained stable performance over 20 adsorption/desorption cycles. Similarly, Li [18] et al. selected different types of zeolites as mass transfer enhancement and structural stabilization materials to prepare a series of zeolite/ MgSO_4 composite materials. Zhang et al. [19] prepared composite heat storage materials by adding hydrophilic modified expanded graphite, which showed excellent comprehensive performance. Most system-level studies have adopted packed-bed designs, where flue gas or air directly contacts the salt composite for heat exchange. Michel et al. [20], for instance, evaluated the isothermal adsorption behavior of SrBr_2 composites in a packed bed and found that increasing particle size improved vapor transport and adsorption rate. Hao et al. [21] designed a multi-module cylindrical packed bed reactor and made a numerical comparison of the performance of the top peripheral air inlet and the bottom central air inlet scheme. The bottom central air inlet scheme has obvious advantages, such as uniform reaction rate, short reaction time, and small resistance loss. In numerical studies, Rui et al. [22] used a 3D model to analyze the effects of inlet temperature, flow rate, salt porosity, and bed geometry on desorption

performance, finding that raising inlet temperature from 80 °C to 92 °C reduced desorption time by 35% and increased the desorbed mass fraction by 15%.

However, compared to conventional flue gas systems, the waste heat from natural gas injection differs significantly in process conditions—direct contact between natural gas and hydrated salts is not feasible. To maintain the required gas cooling rate, this study proposes replacing the traditional packed bed with a moving bed to recover waste heat from natural gas. A three-dimensional numerical model coupling the desorption and heat absorption of hydrated salts is developed to simulate the temperature and flow fields within the moving bed and validate the feasibility of the proposed approach. The effects of key parameters—including salt inlet temperature, gas inlet temperature, flow velocity, and mass flow ratio—on salt outlet temperature, gas outlet temperature, overall heat transfer coefficient, heat recovery efficiency, and salt desorption fraction are systematically investigated. The results aim to provide theoretical guidance and data support for the seasonal utilization of waste heat in underground gas storage systems.

2. Methodology

2.1. Physical Model

A small-scale physical model of the moving bed heat exchanger is shown in Figure 1a. The overall height of the model is 1164 mm, with a thickness of 49 mm and a width of 312 mm. The hydrated salt enters from a top inlet that is 125 mm wide and exits through a bottom outlet 75 mm wide. Internally, three columns of serpentine heat exchange tubes are arranged in a staggered layout. Each tube has an outer diameter of 9 mm. To reduce mesh count, the wall thickness of the tubes is neglected in the geometric model; instead, a virtual wall thickness of 1 mm is defined in Fluent boundary conditions to account for thermal resistance. The transverse spacing between columns is 12.5 mm, while the longitudinal spacing between straight sections of tubes in a single column is 50 mm. Each straight section has a length of 240 mm.

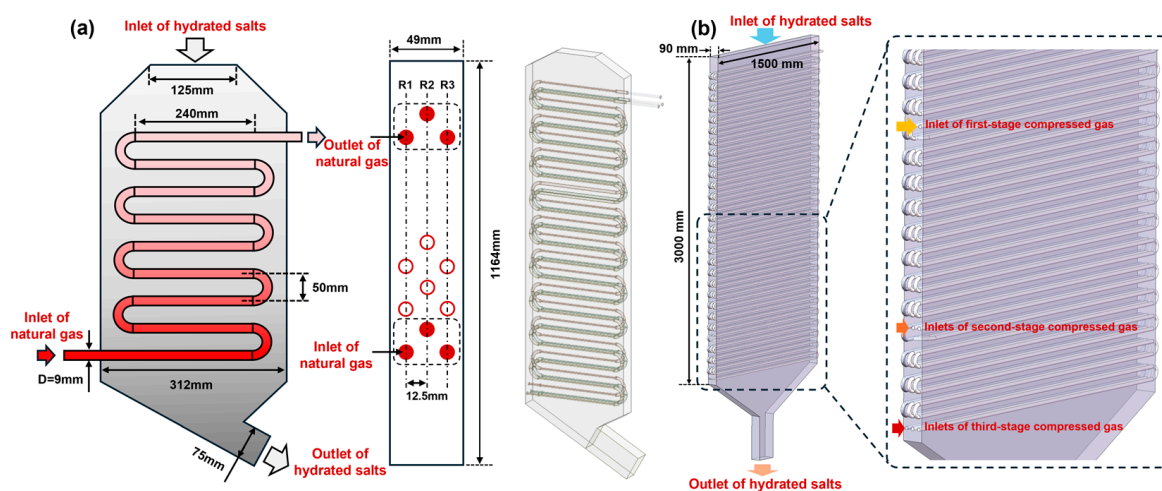


Figure 1. Schematic diagram of (a) the small-scale physical model [5] and (b) full-scale physical model.

To simulate the waste heat recovery process during multi-stage natural gas compression and injection, a full-scale model of the moving bed heat exchanger was constructed based on the small-scale prototype, as illustrated in Figure 1b. The full-size heat exchanger is 3000 mm in height, 1500 mm in width, and 90 mm in thickness. The serpentine heat exchange tubes have an outer diameter of 15 mm and are arranged in-line, consisting of

57 rows and 3 columns. The tubes have a transverse center-to-center spacing of 30 mm and a longitudinal spacing of 50 mm.

To replicate the staged temperature profile of compressed natural gas, three gas inlets are placed at different positions along the tube array. The first set of inlets—located at the first row of the tube bundle—introduces natural gas from the third stage of compression, with a temperature of approximately 100 °C. This high-temperature gas flows through three columns and merges into the innermost column by the 8th row. Two inlets for second-stage compressed gas (approximately 80 °C) are introduced at the 9th row, with their streams merging into one column by the 24th row. Finally, a single inlet for first-stage compressed gas (approximately 60 °C) is located at the 25th row. After heat exchange, the cooled natural gas exits the system through three outlets positioned at the upper-right end of the exchanger. It should be noted that the tube lengths for the third-stage, second-stage, and first-stage compressed gas are 90 m, 77.3 m, and 55.2 m, respectively. The corresponding pressures are 20 MPa, 14 MPa, and 8 MPa, while the resistances are 0.26 MPa, 0.27 MPa, and 0.53 MPa, respectively ($Q_{mg}/Q_{ms} = 7.35$, $u_s = 0.1$ mm/s, $T_{g-in-H} = 100$ °C).

2.2. Numerical Methodology

Numerical simulations in this study were conducted using the Fluent module within ANSYS 2023R1, employing the finite volume method (FVM). Although the discrete element method (DEM) would typically be more appropriate for modeling particulate systems, it was not adopted here due to the large volume and high particle count of the system, which would place an excessive burden on computational resources [23]. Similarly, the Euler–Euler two-fluid model has been shown to perform inadequately when predicting heat transfer and the low-velocity motion of densely packed particles in packed or moving beds [24]. To simplify the modeling process, the particle phase was treated as a monophasic fluid. The governing equations are as follows:

The Mass Conservation Equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \quad (1)$$

Momentum conservation equation:

$$\frac{\partial}{\partial t} (\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho \vec{g} + \vec{F} \quad (2)$$

where p is the static pressure, $\bar{\tau}$ is the stress tensor, $\rho \vec{g}$ and $\vec{F}$ are the gravitational volume force and other volume forces, respectively.

The Energy Equation:

$$\frac{\partial}{\partial t} (\rho H) + \nabla \cdot (\rho \vec{v} H) = \nabla \cdot \left(\frac{k_{eff}}{c_p} \nabla H \right) + Sh \quad (3)$$

k_{eff} is the effective conductivity, c_p is the thermal conductivity, S_h is the volumetric heat sources.

The SST κ - ω turbulence model was selected due to its superior stability and accuracy for systems exhibiting both laminar and turbulent flow regimes. In the current configuration, flow within the heat exchanger exhibits low velocities, with laminar flow dominating near-wall regions and turbulence developing further from the boundaries. Additionally, the SST κ - ω model places relatively low demands on mesh refinement near walls, enabling a reduction in the total number of grid cells required.

In this model, the particle inflow velocity is on the order of millimeters per second. Heat transfer between particles and tube bundles in the moving bed is dominated by

conduction, making the accurate estimation of the bed's effective thermal conductivity a critical factor. Assuming the particles are tightly packed, the apparent thermal conductivity is influenced by factors such as particle size, packing structure, inter-particle contact area, and the intrinsic thermal conductivity of the particle material. To estimate this property, the Meredith and Tobias modified model [25]—an empirical correction of the Rayleigh model—was applied to compute the apparent thermal conductivity of stacked particles in the moving bed:

$$\frac{k_{eff}}{k_f} = \frac{\frac{2+\lambda}{1-\lambda} - 2\Phi + 0.409\frac{6+3\lambda}{4+3\lambda}\Phi^{7/3} - 2.133\frac{3-3\Phi}{4+3\Phi}\Phi^{10/3}}{\frac{2+\lambda}{1-\lambda} + \Phi + 0.409\frac{6+3\lambda}{4+3\lambda}\Phi^{7/3} - 0.906\frac{3-3\Phi}{4+3\Phi}\Phi^{10/3}} \quad (4)$$

where k_{eff} is the apparent thermal conductivity of the stacked bed, k_f is the fluid thermal conductivity, $\lambda = k_s/k_{eff}$, k_s is the solid thermal conductivity, and Φ is the solid phase fraction (particle volume ratio) and ψ is the porosity (void volume ratio) and they satisfy $\Phi + \psi = 1$.

During natural gas injection, the flow rate remains consistent across different inlet positions, but the temperature varies. As a result, waste heat is distributed unevenly: approximately 50% lies in the 40–60 °C range, 33% in the 60–80 °C range, and 17% in the 80–100 °C range. This uneven distribution implies that using a single hydrated salt for heat storage would result in suboptimal recovery. $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ undergoes desorption to $\text{CaCl}_2 \cdot 2\text{H}_2\text{O}$ at 30–60 °C with an enthalpy change of 1147 J/g, while $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$ desorbs to $\text{MgSO}_4 \cdot 2\text{H}_2\text{O}$ at 60–100 °C with an enthalpy change of 1100 J/g [26]. Thus, a composite of two hydrated salts— $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$ —was selected as the thermal storage medium, with a mass ratio of 1:1. The proportion of mixed hydrated salt is selected based on the comprehensive consideration of particle fluidity and temperature range matching. The 1:1 proportion can make the hydrated salt particles fully carry out desorption and heat absorption reaction in both high temperature and medium temperature sections.

The physical properties of the mixed material were calculated using mass-weighted averages. A uniform porosity of 0.5 was assumed. Consequently, the density and effective thermal conductivity of the mixture were determined to be $\rho = 963.5 \text{ kg/m}^3$ and $K_{eff} = 0.094 \text{ W/(m} \cdot \text{°C)}$, respectively. Based on literature data [27,28] and experimental results [5], $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ is assumed to undergo complete desorption when exposed to natural gas temperatures of 60–100 °C, whereas the average desorption fraction for $\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$ under the same conditions is approximately 68%. Therefore, the overall desorption mass fraction of the mixed salts at the moving bed outlet can be expressed as α :

$$\alpha = \begin{cases} 0 & T < 30 \text{ °C} \\ \left(\frac{T}{30} - 1\right) \times 50\% & 30 \leq T < 60 \text{ °C} \\ 50\% + \left(\frac{T}{40} - 1.5\right) \times 34\% & 60 \leq T \leq 100 \text{ °C} \\ 84\% & T > 100 \text{ °C} \end{cases} \quad (5)$$

Similarly, the equivalent specific heat capacity can be calculated as [5]:

$$c_p = \begin{cases} 1215 & T < 30 \text{ °C} \\ 1228.4445T - 35638.3 & 30 \leq T < 60 \text{ °C} \\ -4.5T^2 + 925T - 33944.999 & 60 \leq T \leq 100 \text{ °C} \\ 1215 & T > 100 \text{ °C} \end{cases} \quad (6)$$

To evaluate the overall heat transfer performance between hydrated salt particles and natural gas within the moving bed, the logarithmic mean temperature difference (ΔT_m) was used to calculate the overall heat transfer coefficient h , as follows:

$$\Delta T_m = \frac{(T_{g-out} - T_{s-in}) - (T_{g-in} - T_{s-out})}{\ln\left(\frac{T_{g-out} - T_{s-in}}{T_{g-in} - T_{s-out}}\right)} \quad (7)$$

$$h = \frac{C_{pg} Q_{mg} (T_{g-in} - T_{g-out})}{A \cdot \Delta T_m} \quad (8)$$

Here, T represents temperature; subscripts g , s , in and out denote natural gas, hydrated salt, inlet and outlet parameters, respectively; c_{pg} is the specific heat capacity of the high-pressure natural gas; Q_{mg} is the mass flow rate of natural gas; and A is the effective heat exchange area within the moving bed. To assess waste heat recovery performance, the heat recovery efficiency is calculated with 40 °C as the benchmark gas outlet temperature. If the outlet temperature falls to or below 40 °C, the recovery efficiency is considered to be 100%. The corresponding expression is:

$$\eta = \frac{T_{g-in} - T_{g-out}}{T_{g-in} - 40} \times 100\% \quad (9)$$

2.3. Method Validation

In this study, heat transfer between the hydrated salt particles and the tube bundle is simplified as heat transfer between a single-phase fluid and the tube wall. This engineering-based approximation significantly reduces computational costs. However, its accuracy must be validated. It is particularly important to note that in actual moving beds, particles may form agglomerations, channelized flow patterns, or flow dead zones due to variations in particle size distribution, irregular shapes, wall effects, or localized flow resistance. These phenomena can create regions where partial desorption occurs, while the overall desorption fraction predicted by models might mask such non-uniformities. To this end, we referred to the experimental work by Liu et al. [29,30], who investigated heat transfer between high-temperature blast furnace slag particles and water-cooled tubes using a setup (approximately 8.91 million grids) nearly identical to our physical model.

A comparison was made between simulated and experimental results under the following conditions: internal pipe Reynolds number $Re = 5930$, particle inlet temperature = 700 °C, and particle inlet velocities of $u_s = 0.65 \text{ mm} \cdot \text{s}^{-1}$ and $1.46 \text{ mm} \cdot \text{s}^{-1}$. As listed in Table 1, simulation results closely matched experimental data. At high flow rate, the dominant heat transfer mode is dominated by forced convection, and the single-phase fluid model can predict temperature and power with high precision. At a low flow rate, the particle retention time is prolonged, forming dense accumulation, and the contact heat conduction mode accounts for a larger proportion, so it can only achieve medium precision at low speed. As particle velocity increased, the simulation error decreased. At $u_s = 1.46 \text{ mm} \cdot \text{s}^{-1}$, the errors in particle outlet temperature, outlet water temperature, and heat transfer power were 6.8%, 2.0%, and 3.0%, respectively, which confirms that the proposed model achieves a high degree of accuracy in predicting heat transfer between moving particles and tube bundles, and is therefore suitable for simulating the heat exchange behavior of hydrated salt in the bed.

Table 1. Comparison between experimental and simulation results.

u_s	Particle Outlet Temperature	Experiment Water outlet Temperature	Heat Transfer Power	Particle Outlet Temperature	Simulation Water Outlet Temperature	Heat Transfer Power
0.65 mm·s ^{−1}	159.5 °C	31.1 °C	8.02 kW	214.6 °C	34.2 °C	9.49 kW
1.46 mm·s ^{−1}	356.5 °C	44.4 °C	14.25 kW	382.6 °C	43.5 °C	13.84 kW

A mesh independence test was conducted using three small-scale models with 12.45, 14.60, and 17.47 million elements, respectively. When increasing the mesh size from 12.45 to 14.60 million, the outlet temperature of natural gas and total heat transfer power changed by only 0.8% and 3.2%, respectively. Further increasing the mesh to 17.47 million led to marginal changes of just 0.1% and 0.5%. Therefore, to balance accuracy and computational efficiency, the mesh with 14.60 million cells was adopted for the small-scale model in subsequent simulations. Using geometric scaling, the full-scale model was constructed by proportionally enlarging the validated small-scale model, resulting in a total mesh count of approximately 118 million elements.

3. Results and Discussion

3.1. Flow and Heat Transfer Characteristics of Mixed Hydrated Salts in the Small-Scale Moving Bed

To investigate the flow and heat transfer behavior of the mixed hydrated salts in the moving bed, an energy balance method was used to determine the mass flow rate ratio between natural gas and hydrated salts; that is, the heat released by natural gas and the heat absorbed by hydrated salt are conserved, yielding $Q_{mg}/Q_{ms} = 4.91$. Based on this ratio, five operating conditions were designed to examine the influence of inlet parameters on outlet temperatures, overall heat transfer coefficient, and waste heat recovery efficiency. For all cases, the inlet temperatures of natural gas and hydrated salts were fixed at 100 °C (ignoring multi-stage compression effects) and 25 °C, respectively. The inlet velocities of natural gas (u_g) and hydrated salts (u_s) are listed in Table 2.

Table 2. Inlet velocity parameters at $Q_{mg}/Q_{ms} = 4.91$.

Case	1	2	3	4	5
u_s (mm·s ^{−1})	0.05	0.1	0.4	1	2
u_g (m·s ^{−1})	0.263	0.525	2.1	5.25	10.5

Figure 2a shows the simulation results for outlet temperatures. As the inlet velocities of both gas and salt increase, the outlet temperature of the hydrated salts (T_{s-out}) decreases, while the outlet temperature of natural gas (T_{g-out}) increases. Specifically, in Case 1, the outlet temperatures of the salt and gas were 67.6 °C and 47.4 °C, respectively, while in Case 5, they reached 49.3 °C and 79.8 °C. Although the temperature difference for counter-current heat exchange narrows as velocities rise, the overall convective heat transfer coefficient increased significantly—from 14.2 to 111.3 W·m^{−2}·°C^{−1}, as shown in Figure 2b. Due to this enhanced heat transfer, even when the velocities of both fluids were increased by 40 times, the heat recovery efficiency of natural gas only declined from 88.7% to 33.6% (Figure 2c). However, to ensure safe operation and reduce energy consumption during gas injection, the outlet temperature of natural gas should ideally be reduced to around 40 °C. At the same time, a higher outlet temperature of the hydrated salts is desirable to promote desorption rate and improve the overall desorption fraction. In practical injection operations, the flow velocity of natural gas in the main pipeline can reach up to 15 m·s^{−1}. The results indicate

that the current small-scale moving bed is insufficient to accommodate both the technical demands of high-speed gas injection and efficient waste heat recovery. This underscores the need to increase the heat exchange area and extend residence time of the gas within the bed.

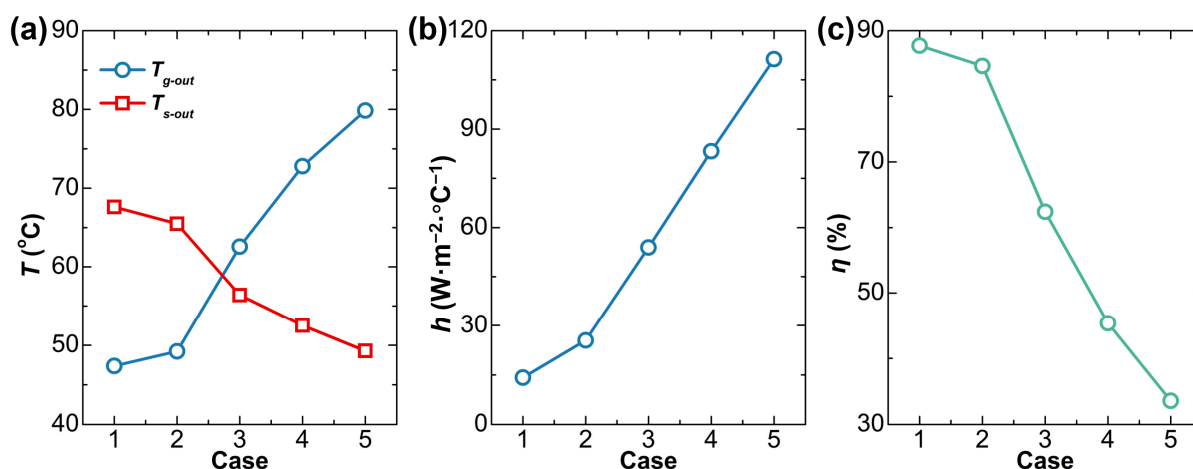


Figure 2. Effects of inlet parameters on heat transfer characteristics in the moving bed with mixed hydrated salt particles at $Q_{mg}/Q_{ms} = 4.91$. (a) Variation of natural gas and hydrated salt outlet temperatures, (b) Variation of overall heat transfer coefficient, (c) Variation of natural gas waste heat recovery efficiency.

3.2. Flow and Heat Transfer Characteristics of Mixed Hydrated Salts in the Full-Scale Moving Bed

A representative full-scale simulation was conducted using the following conditions: $u_s = 0.2 \text{ mm}\cdot\text{s}^{-1}$, $Q_{ms} = 0.026 \text{ kg}\cdot\text{s}^{-1}$, $Q_{mg} = 0.191 \text{ kg}\cdot\text{s}^{-1}$, yielding $Q_{mg}/Q_{ms} = 7.35$. The inlet temperatures for high-, medium-, and low-pressure natural gas streams were 100°C (T_{g-in-H}), 80°C (T_{g-in-M}), and 60°C (T_{g-in-L}), respectively, with the salt entering at 25°C . The physical properties of the three gas streams are summarized in Table 3.

Table 3. Physical properties of natural gas.

P_g (Mpa)	T_{g-in} ($^{\circ}\text{C}$)	K_g ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	μ_g ($\text{Pa}\cdot\text{s}$)	ρ_g ($\text{kg}\cdot\text{m}^{-3}$)	C_{pg} ($\text{J}\cdot\text{kg}^{-1}\cdot^{\circ}\text{C}^{-1}$)
20	100	0.0579	1.86E-05	134.006	3090.3
14	80	0.0501	1.61E-05	98.6383	2940.5
8	60	0.0425	1.38E-05	56.7136	2634.7

Figure 3 presents temperature contour maps of the hydrated salt particles within the full-scale moving bed. The maps show the temperature distribution of the full-size moving bed and central vertical cross-sectional plane. It is evident that both the salt and gas temperatures increase along the vertical direction as they flow through the bed. At the outlet, the hydrated salt reaches a temperature of 84°C , while the average outlet temperature of natural gas is 41°C . The temperature difference between the two phases remains relatively small throughout the bed. A comparison of the overall and mid-plane temperature fields reveals only minor differences, suggesting that the placement of the three gas inlets is effective in achieving a uniform temperature distribution across the bed's thickness. This confirms the rationality of the inlet configuration and supports the feasibility of using multiple gas streams with different temperature levels for efficient heat recovery.

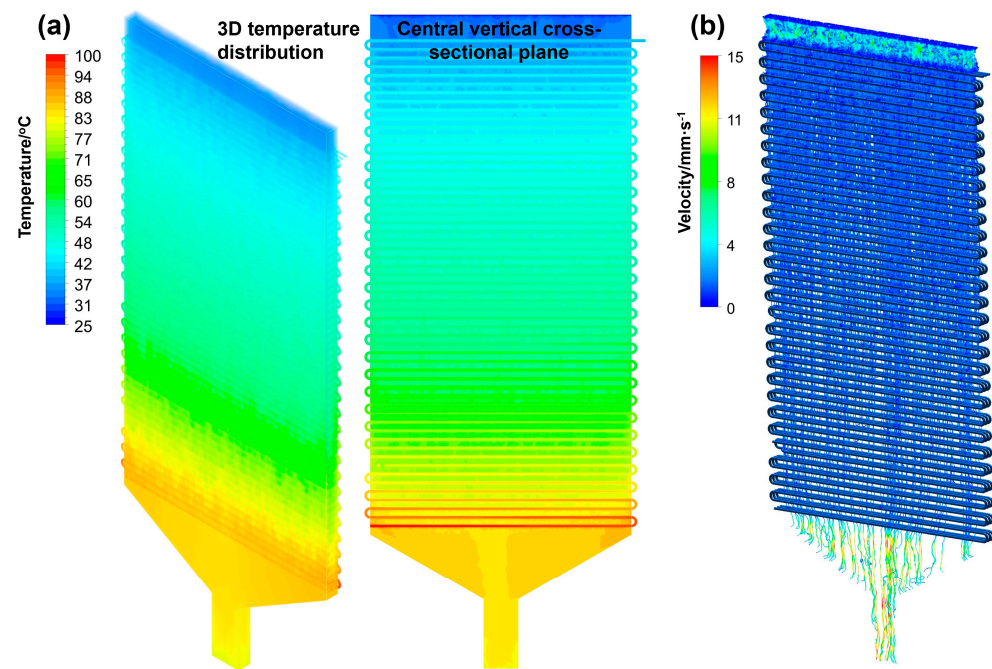


Figure 3. (a) Temperature distribution in the full-scale moving bed and (b) velocity distribution of mixed hydrated salts.

3.3. Effect of Natural Gas Inlet Temperature on Flow and Heat Transfer Characteristics in the Moving Bed

During the gas injection process in underground gas storage, natural gas parameters can fluctuate within a certain range. To evaluate the adaptability of the moving bed system to such fluctuations, three sets of natural gas inlet temperatures were considered: $T_{g-in-H} = 100/90/80$ °C; $T_{g-in-M} = 80/70/60$ °C; $T_{g-in-L} = 60/50/40$ °C. All three temperature groups were varied simultaneously. The other operating conditions were kept constant: $u_s = 0.3$ mm·s⁻¹, salt mass flow rate $Q_{ms} = 0.039$ kg·s⁻¹, gas mass flow rate $Q_{mg} = 0.234$ kg·s⁻¹, $Q_{mg}/Q_{ms} = 6$, $T_{s-in} = 25$ °C.

As shown in Figure 4a, both the outlet temperatures of the natural gas and the hydrated salt increased with rising inlet gas temperatures. Notably, when the outlet temperature of the salt exceeded 60 °C, the rate of temperature rise accelerated. This behavior is attributed to the reduced desorption enthalpy of the hydrated salts above 60 °C, leading to less energy absorption per unit temperature increase. Despite the variation in inlet temperatures, the outlet temperature of natural gas remained below the target cooling threshold of 40 °C in all cases. Consequently, as shown in Figure 4b, the waste heat recovery efficiency for natural gas reached 100% under all tested conditions, indicating complete recovery of the available thermal energy. Additionally, based on the outlet temperature of the hydrated salt, the average desorption mass fraction of the mixed salts was calculated, as illustrated in Figure 4c. The results indicate a clear upward trend in desorption fraction with increasing inlet gas temperature, rising from 43.0% to 62.6%. Nevertheless, this desorption efficiency remains suboptimal. To further enhance system performance, future work should focus on improving the desorption rate of $MgSO_4 \cdot 7H_2O$ within the 60–100 °C range, thereby increasing the overall desorption mass fraction and thermal storage effectiveness of the moving bed system.

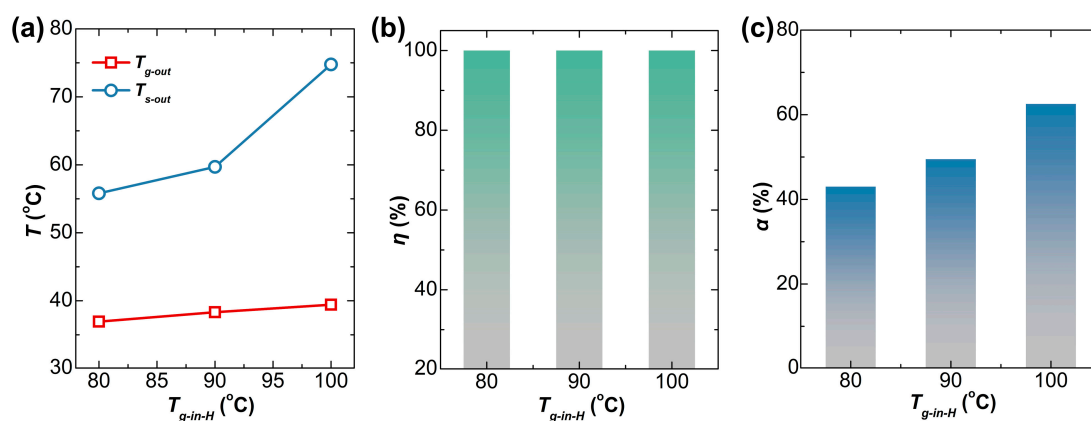


Figure 4. Effects of natural gas inlet temperature (three-stage compression) on heat transfer characteristics in the full-scale moving bed. (a) Variation of outlet temperature, (b) Variation of natural gas waste heat recovery efficiency, (c) Variation of mixed hydrated salt desorption mass fraction.

3.4. Effect of the Inlet Flow Rate of Natural Gas and Hydrated Salt Particles on Flow and Heat Transfer Characteristics in the Moving Bed

To further investigate system performance, the effect of inlet flow velocity for both natural gas and hydrated salt particles was analyzed. Since both gas and particle flow rates can vary independently, making comparative analysis more complex, the mass flow ratio was fixed at $Q_{mg}/Q_{ms} = 7.35$. Under this constraint, any change in one inlet flow automatically results in a proportional change in the other. The tested conditions were as follows: $u_s = 0.1/0.2/0.3 \text{ mm}\cdot\text{s}^{-1}$, $Q_{ms} = 0.013/0.026/0.039 \text{ kg}\cdot\text{s}^{-1}$, $Q_{mg} = 0.096/0.191/0.287 \text{ kg}\cdot\text{s}^{-1}$, $T_{g-in-H} = 100 \text{ }^{\circ}\text{C}$, $T_{g-in-M} = 80 \text{ }^{\circ}\text{C}$, $T_{g-in-L} = 60 \text{ }^{\circ}\text{C}$, $T_{s-in} = 25 \text{ }^{\circ}\text{C}$.

Figure 5a illustrates the outlet temperature trends. As inlet flow velocity increases, the outlet temperature of natural gas rises slightly, but remains consistently around $40 \text{ }^{\circ}\text{C}$. The outlet temperature of the mixed hydrated salts initially increases slightly and then shows a small decline, remaining above $80 \text{ }^{\circ}\text{C}$ in all cases. Overall, changes in flow velocity had a minimal effect on outlet temperatures. This is attributed to the large size of the moving bed, which provides ample heat exchange area to ensure sufficient thermal interaction between gas and salt particles. These results confirm that the designed moving bed has a high processing capacity and strong adaptability to fluctuations in natural gas flow rate.

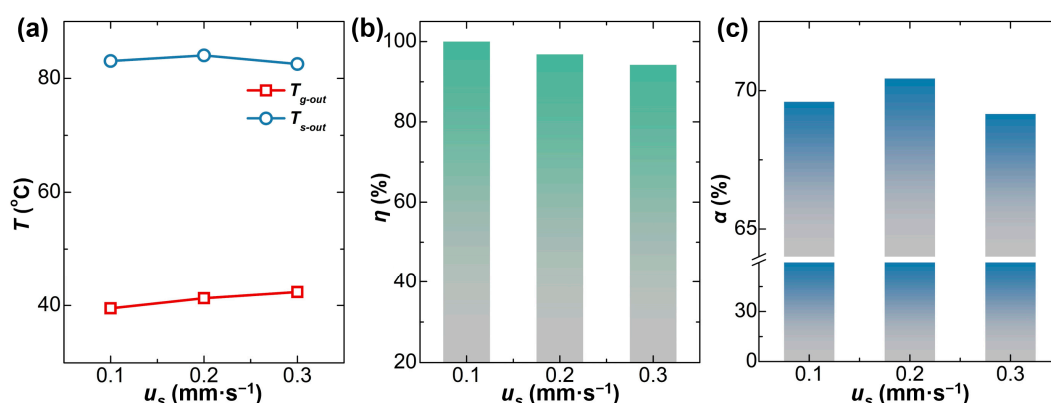


Figure 5. Effects of natural gas and hydrated salt particle inlet flow rates on heat transfer characteristics in the full-scale moving bed. (a) Variation of outlet temperature, (b) Variation of natural gas waste heat recovery efficiency, (c) Variation of mixed hydrated salt desorption mass fraction.

As shown in Figure 5b, the waste heat recovery efficiency of natural gas decreases slightly as inlet flow increases, but remains above 94% in all cases. This further indicates that the system consistently achieves effective heat recovery across varying flow conditions.

and pressure levels. Additionally, with increasing flow rate, the desorption mass fraction initially increases from 69.6% to 70.4%, then decreases slightly to 69.2%. This non-linear trend reflects a balance between two competing effects: higher flow enhances the effective heat transfer coefficient, which can raise the outlet temperature of the salts and promote desorption; however, it can also reduce residence time and potentially lower the final temperature. Therefore, when $u_s < 0.2 \text{ mm} \cdot \text{s}^{-1}$, the improvement in heat transfer dominates, resulting in gradually rising salt outlet temperatures and desorption fractions. When $u_s > 0.2 \text{ mm} \cdot \text{s}^{-1}$, the marginal gains in heat transfer weaken, leading to a slight decline in outlet temperature and desorption performance.

3.5. Effect of the Flow Rate Ratio Between Natural Gas and Hydrated Salt Particles on Flow and Heat Transfer Characteristics in the Moving Bed

The preceding analysis suggests that under well-designed heat exchange conditions, the mass flow rate ratio between natural gas and hydrated salt particles becomes the dominant factor influencing the outlet performance of the moving bed. To further clarify this relationship, this section focuses on analyzing the effect of this ratio on the thermal behavior of the system. The operating conditions were set: $u_s = 0.3 \text{ mm} \cdot \text{s}^{-1}$, $Q_{ms} = 0.039 \text{ kg} \cdot \text{s}^{-1}$, $Q_{mg} = 0.234 / 0.287 / 0.344 \text{ kg} \cdot \text{s}^{-1}$, $Q_{mg} / Q_{ms} = 6 / 7.35 / 8.82$, $T_{g-in-H} = 100 \text{ }^\circ\text{C}$, $T_{g-in-M} = 80 \text{ }^\circ\text{C}$, $T_{g-in-L} = 60 \text{ }^\circ\text{C}$, $T_{s-in} = 25 \text{ }^\circ\text{C}$.

As shown in Figure 6a, increasing the flow ratio led to higher outlet temperatures for both natural gas and hydrated salts. Specifically, as Q_{mg} / Q_{ms} increased from 6 to 8.82, the gas outlet temperature rose from $39.4 \text{ }^\circ\text{C}$ to $46.3 \text{ }^\circ\text{C}$, while the salt outlet temperature increased more significantly, from $74.8 \text{ }^\circ\text{C}$ to $86.5 \text{ }^\circ\text{C}$. Notably, once the ratio exceeded 7.35, the rate of temperature increase for the salts began to slow. Figure 6b presents the corresponding change in waste heat recovery efficiency. As the Q_{mg} / Q_{ms} increased, the efficiency declined markedly—from 100% at a ratio of 6 to 84.2% at a ratio of 8.82. At the same time, the average desorption mass fraction of the hydrated salts improved, increasing from 62.6% to 72.5% across the same range.

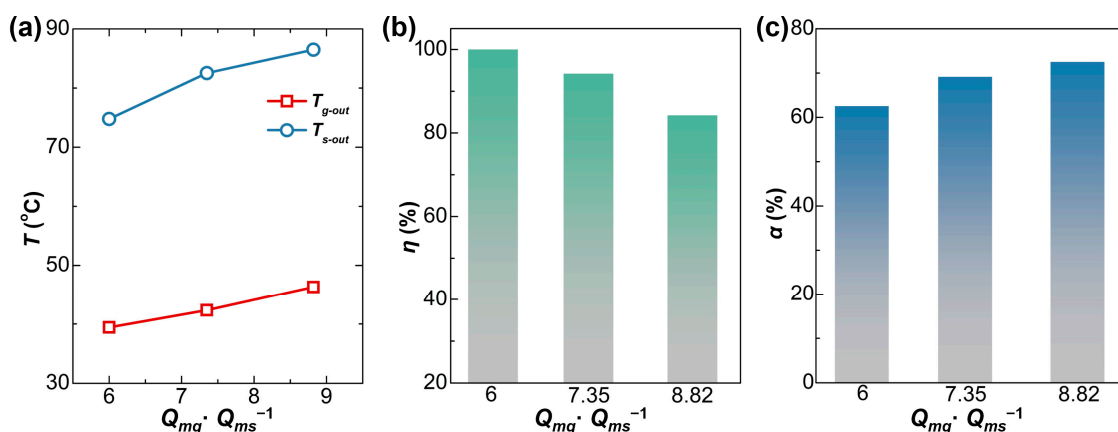


Figure 6. Effects of the flow rate ratio of natural gas to hydrated salt particles on heat transfer characteristics in the full-scale moving bed. (a) Variation of outlet temperature, (b) Variation of natural gas waste heat recovery efficiency, (c) Variation of mixed hydrated salt desorption mass fraction.

This highlights a trade-off: enhancing the desorption extent comes at the cost of reduced heat recovery efficiency. The underlying issue is that the desorption rate of the salts cannot fully keep pace with the increased heat transfer demand. Among the tested conditions, the optimal balance was achieved when $Q_{ms} = 0.039 \text{ kg} \cdot \text{s}^{-1}$, $Q_{mg} = 0.287 \text{ kg} \cdot \text{s}^{-1}$, $T_{g-in-H} = 100 \text{ }^\circ\text{C}$, $T_{g-in-M} = 80 \text{ }^\circ\text{C}$, $T_{g-in-L} = 60 \text{ }^\circ\text{C}$, $T_{s-in} = 25 \text{ }^\circ\text{C}$. Under this condition, the natural gas outlet temperature was $41.3 \text{ }^\circ\text{C}$, the salt outlet temperature was $82.5 \text{ }^\circ\text{C}$, the waste heat recovery efficiency reached 94.2%, and the salt desorption fraction was 69.2%—

representing the best trade-off between operational safety, heat exchange efficiency, and material utilization. Overall, these results emphasize the importance of real-time monitoring of the natural gas flow rate during operation. The flow rate of hydrated salts must be dynamically adjusted to match fluctuations in gas input, ensuring the long-term stability and energy efficiency of the moving bed heat exchanger system.

4. Conclusions

This study developed both a small-scale and full-scale three-dimensional moving bed model, incorporating the endothermic desorption behavior of hydrated salts, to simulate and evaluate the flow and heat transfer performance during natural gas waste heat recovery. The temperature and velocity fields within the moving bed were obtained to verify the feasibility of the proposed design. A parametric analysis was then conducted to assess the effects of hydrated salt inlet temperature, natural gas inlet temperature, flow velocity, and mass flow ratio on key thermal performance indicators, including outlet temperatures, overall heat transfer coefficient, heat recovery efficiency, and salt desorption fraction. The main findings are summarized as follows:

- (1) In the small-scale model (dimensions: $1164 \times 312 \times 49$ mm), the outlet temperatures of natural gas and mixed hydrated salts reached $79.8\text{ }^{\circ}\text{C}$ and $49.3\text{ }^{\circ}\text{C}$, respectively. However, the heat recovery efficiency was only 33.6%, primarily due to insufficient residence time for high-velocity natural gas ($10.5\text{ m}\cdot\text{s}^{-1}$) and salt particles ($2\text{ mm}\cdot\text{s}^{-1}$), indicating that the small-scale setup cannot simultaneously meet process and recovery requirements.
- (2) The full-scale moving bed model exhibited uniform temperature distribution across its thickness, confirming the rationality of the structural design. This validates its applicability for recovering low-grade heat from the three-stage compression process of natural gas injection.
- (3) When the natural gas inlet temperature increased from $80\text{ }^{\circ}\text{C}$ to $100\text{ }^{\circ}\text{C}$, both the heat recovery efficiency and hydrated salt desorption fraction improved—reaching 100% and 62.5%, respectively—while maintaining an outlet gas temperature below $40\text{ }^{\circ}\text{C}$. Further optimization of the mass flow rates allowed the desorption fraction to increase to 69.6%, still under full heat recovery conditions.
- (4) Among all tested conditions, the optimal performance was achieved at $Q_{ms} = 0.039\text{ kg}\cdot\text{s}^{-1}$, $Q_{mg} = 0.287\text{ kg}\cdot\text{s}^{-1}$, $T_{g-in-H} = 100\text{ }^{\circ}\text{C}$, $T_{g-in-M} = 80\text{ }^{\circ}\text{C}$, $T_{g-in-L} = 60\text{ }^{\circ}\text{C}$, $T_{s-in} = 25\text{ }^{\circ}\text{C}$. Under these conditions, the natural gas outlet temperature was $41.3\text{ }^{\circ}\text{C}$, the salt outlet temperature reached $82.5\text{ }^{\circ}\text{C}$, heat recovery efficiency was 94.2%, and the salt desorption mass fraction was 69.2%. This operating point provides an optimal balance between process safety, heat exchange capacity, energy recovery, and material utilization.

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