

Article

Heat Loss Analysis and Energy-Saving Optimization of a High-Power Electric Air Heater

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Abstract

High-power electric air heaters are key charging components in air thermal energy storage systems, but the dominant heat-loss regions and retrofit basis of existing devices remain unclear. In this study, a three-dimensional conjugate heat-transfer model was developed for an existing 1200 kW vertical electric air heater and validated using three steady-state experimental cases, with a maximum outlet-temperature deviation of 2.17%. Based on the validated model, temperature-field characteristics and segmental heat-loss distributions were analyzed under different mass flow rates. The results show that heat loss was highly non-uniform: Segments 2 and 3 accounted for 37.26% and 54.51% of the total heat loss, respectively, contributing 91.77% in total. A targeted local retrofit scheme was, therefore, proposed by filling the non-flowing inner-cylinder region in Segments 2 and 3 with glass wool and enhancing insulation near local cooling boundaries. After optimization, the average total heat loss decreased from 31.94 kW to 17.69 kW, corresponding to a 44.6% reduction. Under the rated condition, the outlet temperature increased from 1421.6 K to 1482.0 K, providing 584.8 kWh of additional effective thermal storage per cycle and an estimated payback period of 399 d. This study provides a diagnosis-guided retrofit approach for existing high-power electric air heaters.

Keywords: high-power electric air heater; air thermal energy storage; heat loss; conjugate heat transfer; numerical simulation; energy-saving optimization



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1. Introduction

With the increasing penetration of renewable power generation, electric power systems are facing greater fluctuations between off-peak electricity surplus and peak-demand electricity shortage. Large-scale energy storage technologies, therefore, play an important role in improving energy flexibility, reducing peak-regulation pressure, and enhancing the utilization of renewable electricity [1]. Among different energy storage routes, air thermal energy storage is attractive because air is safe, inexpensive, readily available, and suitable for high-temperature heat transfer [2]. In such a system, the high-power electric air heater is a key component in the charging stage, because it converts electrical energy into the thermal energy of high-temperature air and directly affects the charging efficiency and operating economy of the storage system [3]. However, under high-power and high-temperature operating conditions, conductive, convective, and radiative heat dissipation

may occur through heating tubes, solid supports, stagnant-air regions, cooling boundaries, and the external shell, resulting in considerable heat loss [4,5]. Therefore, improving the heat retention performance and reducing the heat loss of high-power electric air heaters are important for enhancing the energy utilization efficiency of air thermal energy storage systems [6,7].

In practical engineering applications, high-power electric air heaters usually have complex three-dimensional structures, including pressure-bearing shells, electric heating tube bundles, support plates, electrode regions, cooling-water structures, thermal insulation layers, and inlet/outlet ducts. The internal heat-transfer process is, therefore, not governed only by the main forced airflow, but also by radial heat conduction, local natural convection in stagnant regions, heat transfer through support structures, and heat dissipation near local cooling boundaries. For existing high-power heaters, energy-saving retrofit is further constrained by the original shell size, heating tube arrangement, flow channel layout, support plate structure, and internal accessibility. As a result, simply increasing the overall insulation thickness may not be the most effective retrofit strategy. Before a targeted retrofit scheme can be designed, it is necessary to quantitatively determine where the dominant heat loss occurs, how heat is transferred from the high-temperature region to the external boundaries, and which local regions should be preferentially optimized.

Extensive studies have been conducted to improve the energy efficiency of industrial heating equipment. Relevant reviews have shown that the energy-saving performance of industrial heating systems depends not only on the heating device itself, but also on operating strategies, waste heat recovery, process optimization, and equipment retrofit [8]. In terms of operational control, previous studies have investigated temperature regulation, flow coordination, and control strategies for high-temperature airflow systems, hot-air wind tunnels, and heat calibration devices [9–12]. These studies are useful for improving outlet-temperature stability and operation reliability, but they mainly act on the operating process and have limited ability to modify the internal thermal resistance distribution and structural heat leakage paths of existing equipment.

Structural heat-transfer optimization has also been studied for industrial ovens, paint-curing ovens, air heaters, and other thermal equipment [13–16]. These studies have demonstrated that improving insulation, optimizing flow organization, and modifying local heat-transfer structures can enhance heating efficiency. For existing industrial thermal equipment, measures such as equipment replacement, duct sealing, insulation enhancement, and local retrofit have also been reported to provide energy-saving benefits [17–19]. Nevertheless, many retrofit schemes are still based on overall heat-loss estimation or empirical insulation improvement, rather than on quantitative identification of dominant internal heat-loss regions. In addition, CFD-based thermal management and numerical simulation have been widely used to analyze complex temperature and flow fields in thermal equipment [20,21]. Studies on physical-field reconstruction under limited sensor conditions have further indicated that experimental monitoring alone is often insufficient to capture complex three-dimensional thermal-field distributions [22,23]. However, most existing studies do not establish a complete diagnostic retrofit framework that links validated three-dimensional conjugate heat-transfer modeling, segmented heat-loss calculation, dominant heat-transfer path identification, targeted local retrofit, and engineering-economic evaluation for existing high-power electric air heaters.

To clarify the position of the present study relative to previous research, Table 1 compares the main research streams related to air thermal energy storage, high-temperature air heating, industrial heating equipment, retrofit strategies, and CFD-based thermal diagnosis.

Table 1. Comparison of previous studies and the present work.

Research Stream	Representative References	Main Focus of Previous Studies	Remaining Gaps Related to the Present Study	Contribution of the Present Study
Energy storage systems and air thermal energy storage	[1–3]	Renewable energy utilization, energy storage flexibility, air thermal energy storage, and the role of electric air heaters in the charging process	The internal heat-transfer paths and local heat-loss characteristics of the electric air heater are usually not resolved at the component level	Focuses on an existing 1200 kW electric air heater as the key charging component in an air thermal energy storage scenario
Heat loss and energy-saving improvement of heating equipment	[4–8]	Heat dissipation mechanisms, heat retention performance, energy-saving requirements, and general efficiency improvement of industrial heating equipment	Existing studies usually discuss overall heat loss or general energy-saving measures, but do not identify dominant heat-loss regions inside a complex high-power heater	Quantifies the non-uniform segmental heat-loss distribution and identifies the dominant heat-loss segments
Operational control of high-temperature airflow systems	[9–12]	Temperature regulation, flow coordination, and operational stability of high-temperature airflow systems, hot-air wind tunnels, and calibration devices	Control strategies can improve outlet-temperature stability but cannot directly modify internal structural heat leakage paths	Provides a structural heat-loss diagnosis rather than only an operation control improvement
Structural optimization of ovens, air heaters, and thermal equipment	[13–16]	Flow organization, insulation enhancement, CFD analysis, and local structural optimization of industrial heating devices	The studied equipment types and heat-loss mechanisms differ from existing high-power electric air heaters used in air thermal energy storage systems	Establishes a three-dimensional conjugate heat-transfer model for the specific geometry and boundary conditions of the studied heater
Retrofit of existing industrial thermal equipment	[17–19]	Equipment replacement, duct sealing, insulation enhancement, and local retrofit of existing thermal systems	Retrofit measures are often based on empirical or overall insulation improvement rather than quantified internal heat-loss paths	Proposes a targeted local retrofit scheme based on the identified dominant heat-loss regions
CFD-based thermal diagnosis and field reconstruction	[20–23]	Numerical thermal management, temperature-field prediction, and reconstruction of complex thermal fields under limited measurements	Numerical diagnosis is rarely connected with segmented heat-loss calculation, targeted retrofit design, and economic evaluation in one framework	Links validated 3D thermal diagnosis, segmented heat-loss analysis, local retrofit, and economic evaluation
Present study	This work	Existing high-power electric air heater in an air thermal energy storage application	—	Develops a diagnostic retrofit framework combining validated 3D conjugate heat transfer modeling, dominant heat-loss identification, local thermal resistance enhancement, and economic performance evaluation

As shown in Table 1, previous studies have provided useful methods for improving the energy efficiency, operational stability, and retrofit performance of thermal storage systems and industrial heating equipment. However, for existing high-power electric air heaters used in air thermal energy storage systems, three issues remain insufficiently addressed. First, retrofit-oriented three-dimensional conjugate heat-transfer diagnosis under practical structural constraints is still limited. Second, the segmental contribution of different structural regions to total heat loss has not been sufficiently quantified. Third, the connection between identified heat-loss paths, targeted local retrofit design, and engineering–economic evaluation remains weak. These gaps limit the development of targeted and low-intrusion retrofit schemes for existing high-power electric air heaters.

To address these gaps, this study takes an existing 1200 kW vertical electric air heater used in an air thermal energy storage scenario as the research object. A three-dimensional conjugate heat-transfer model is established, verified through mesh independence and time-step independence tests, and validated using three steady-state experimental cases. Based on the validated model, the temperature-field distribution and segmental heat-loss characteristics under different mass flow rates are analyzed to identify the dominant heat-loss regions. A targeted local retrofit scheme is then proposed by filling the non-flowing air region of the inner cylinder with glass wool and enhancing insulation near local cooling boundaries. Finally, the retrofit performance is evaluated in terms of heat-loss reduction, thermal efficiency improvement, outlet-temperature increase, effective thermal storage gain, and payback period. Through this procedure, this study forms a diagnostic retrofit framework that connects model validation, segmental heat-loss identification, targeted local retrofit design, and engineering–economic assessment.

2. Materials and Methods

2.1. Description of the Study Object

A high-power electric air heater with a rated power of 1200 kW is taken as the research object. The heater is mainly used for heating air in a high-temperature air supply system. The device has a vertical configuration and mainly consists of the top electrode and cooling water region, heating tube bundle region, support plate region, bottom cooling water region, and air inlet and outlet. Since the internal airflow and heat-transfer processes are relatively complex, a three-dimensional geometric model is established, as shown in Figure 1.

Figure 1 shows the geometric model and segment division of the electric air heater. When the device is considered in its vertical installation state, Segment 1 is located at the top of the heater, whereas Segment 4 is located at the bottom. After entering the heater through the air inlet, the air first enters the top chamber, namely, Segment 1. It then gathers in this region and enters the heating tube bundle, flowing downward along the axial direction of the heating tubes. During this process, the air successively passes through Segments 2 and 3 and exchanges heat with the heating tubes and surrounding structures. Finally, the air reaches the bottom region, namely, Segment 4, where it continues to exchange heat with the surrounding structures before leaving through the air outlet. According to the airflow path, heating tube distribution, and differences in the local heat-transfer environments, the heater is divided into four segments, namely, Segments 1–4, to provide a basis for subsequent temperature field analysis, heat-loss path identification, and segmented energy calculation. Specifically, Segment 1 mainly corresponds to the region affected by the top electrode and cooling water boundary, as well as the inlet transition region. Segment 2 is the initial heating region of the heating tube bundle and is adjacent to the support plate and cooling water inlet. Segment 3 is the central main heating-tube bundle region, where the air is continuously heated, and the high-temperature region

is formed. Segment 4 mainly corresponds to the region affected by the bottom cooling water boundary and the outlet transition region. This segment division reflects both the streamwise heating process of the air and the influence of local cooling boundaries on the heat-transfer environment.

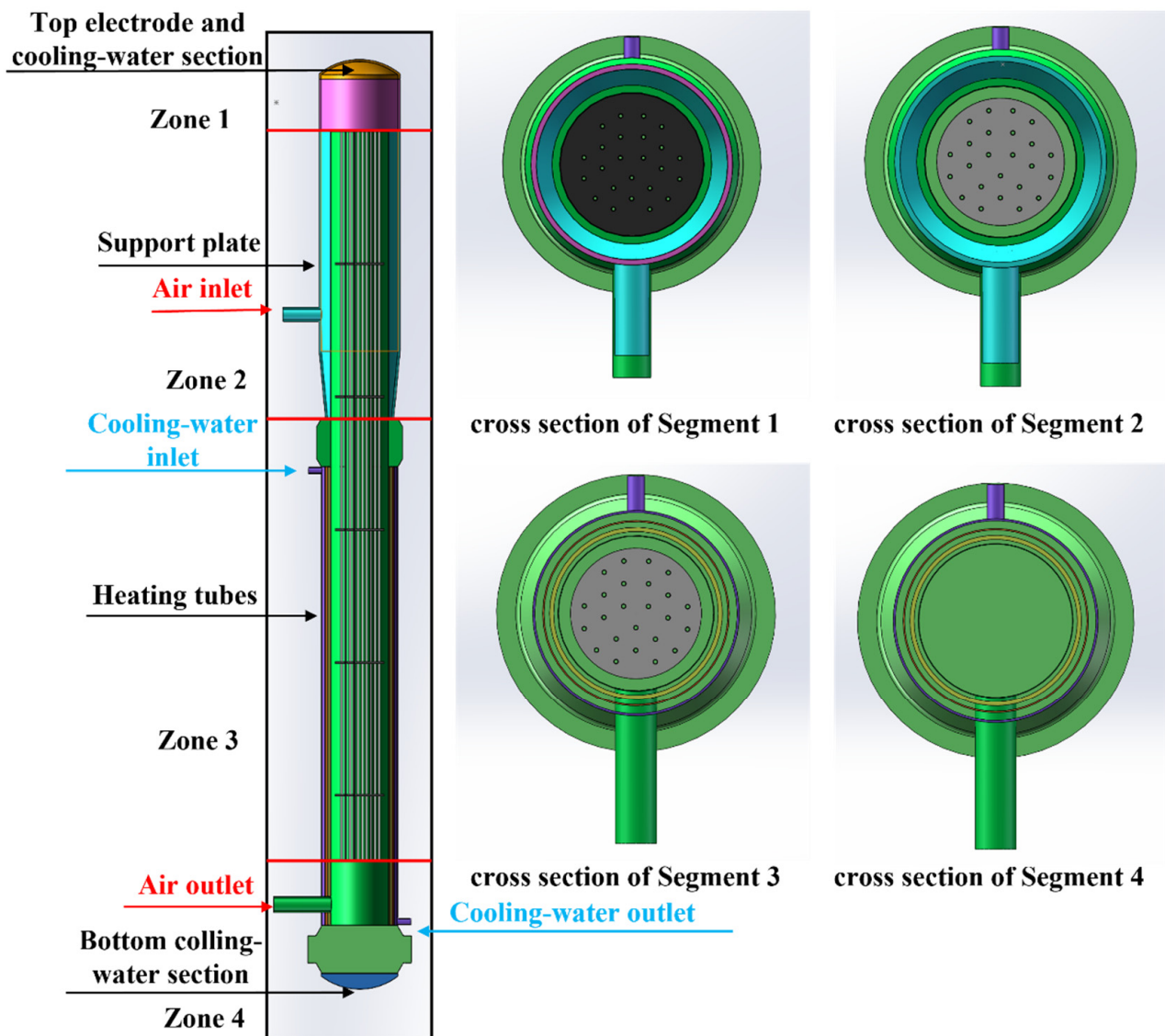


Figure 1. Geometric model, segment division, and cross-sectional views of the electric air heater.

During model construction, local geometric and physical details are appropriately simplified. The electrodes and electrode cooling water structures at the head and bottom of the heater are equivalently treated as constant-temperature solids, and thermal flow boundary conditions are applied to their contact surfaces with the fluid to maintain the consistency of local thermal boundary conditions. The hole-tube connection structure at the support plate is simplified by setting the hole diameter equal to the outer diameter of the heating tube, so that the tube-hole interfaces are matched. On the basis of retaining the main heat-transfer boundary conditions and structural features, these simplifications reduce geometric complexity and facilitate subsequent mesh generation and numerical solution.

2.2. Model Formulation and Numerical Methods

Based on the above model, a three-dimensional conjugate heat-transfer model is established for the electric air heater. The computational domain includes both the air flow domain and the solid domain, and the following assumptions are adopted: air is treated as a continuum; turbulence is used to describe the flow; the thermophysical properties of both the fluid and solid domains vary with temperature; temperature continuity and heat-flux conservation are satisfied at the fluid–solid interface; and heat conduction and thermal storage are considered in the solid domain.

The air flow in the fluid domain is described using the standard conservative form of the governing equations. The mass, momentum, and energy conservation equations are given by Equations (1)–(3), respectively [24].

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

$$\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\nabla p + \nabla \cdot (\mu_{\text{eff}} \nabla \mathbf{u}) \quad (2)$$

$$\frac{\partial (\rho c_p T)}{\partial t} + \nabla \cdot (\rho c_p \mathbf{u} T) = \nabla \cdot (\lambda_{\text{eff}} \nabla T) \quad (3)$$

In these equations, t is time, with the unit of s ; ρ is the fluid density, with the unit of kg/m^3 ; $\mathbf{u}$ is the velocity vector, with the unit of m/s ; p is the pressure, with the unit of Pa ; T is the temperature, with the unit of K ; c_p is the specific heat capacity at constant pressure, with the unit of $\text{J}/(\text{kg} \cdot \text{K})$; μ_{eff} is the effective dynamic viscosity, with the unit of $\text{Pa} \cdot \text{s}$; and λ_{eff} is the effective thermal conductivity, with the unit of $\text{W}/(\text{m} \cdot \text{K})$. The terms in Equation (1) have the unit of $\text{kg}/(\text{m}^3 \cdot \text{s})$, those in Equation (2) have the unit of N/m^3 , and those in Equation (3) have the unit of W/m^3 .

In the solid domain, only heat conduction and thermal storage are considered. The corresponding energy conservation equation is expressed by the transient heat conduction equation, as given in Equation (4) [25].

$$\rho_s c_s \frac{\partial T}{\partial t} = \nabla \cdot (\lambda_s \nabla T) \quad (4)$$

In Equation (4), ρ_s is the solid density, with the unit of kg/m^3 ; c_s is the specific heat capacity of the solid, with the unit of $\text{J}/(\text{kg} \cdot \text{K})$; λ_s is the thermal conductivity of the solid, with the unit of $\text{W}/(\text{m} \cdot \text{K})$; and T is the solid temperature, with the unit of K . All terms in Equation (4) have the unit of W/m^3 .

Because the Reynolds number of the air flow inside the device is relatively high, the standard k – ε turbulence model is employed to close the governing equations. The transport equations for turbulent kinetic energy and turbulent dissipation rate are given by Equations (5) and (6), respectively [26].

$$\frac{\partial (\rho k)}{\partial t} + \nabla \cdot (\rho \mathbf{u} k) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + G_k - \rho \varepsilon \quad (5)$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \varepsilon) = \nabla \cdot \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_1 \frac{\varepsilon}{k} G_k - C_2 \rho \frac{\varepsilon^2}{k} \quad (6)$$

In Equations (5) and (6), k is the turbulent kinetic energy, with the unit of m^2/s^2 ; ε is the turbulent dissipation rate, with the unit of m^2/s^3 ; μ is the dynamic viscosity, with the unit of $\text{Pa} \cdot \text{s}$; μ_t is the turbulent dynamic viscosity, with the unit of $\text{Pa} \cdot \text{s}$; G_k is the production term of turbulent kinetic energy due to the mean velocity gradients, with the unit of $\text{kg}/(\text{m} \cdot \text{s}^3)$; σ_k and σ_ε are the turbulent Prandtl numbers for k and ε , respectively, and

are dimensionless; and C_1 and C_2 are empirical model constants and are dimensionless. The terms in Equation (5) have the unit of $\text{kg}/(\text{m}\cdot\text{s}^3)$, while those in Equation (6) have the unit of $\text{kg}/(\text{m}\cdot\text{s}^4)$.

The main boundary conditions used in the pre- and post-optimization simulations are summarized as follows. At the air inlet, a mass flow inlet boundary condition was applied, with $\dot{m} = \dot{m}_{\text{in}}$, and $T = T_{\text{in}}$. At the air outlet, a pressure outlet boundary condition was used, with $p = p_{\text{out}}$. The heating tubes were treated as the electrical heat input region, and the total input power was prescribed as $\dot{Q}_{\text{total}} = P_{\text{in}}$. At the cooling water boundaries, an equivalent constant-temperature boundary condition was imposed, namely, $T = T_{\text{c}}$. At the external wall of the heater, an equivalent ambient heat-loss boundary was applied, expressed as $-\lambda_s \nabla T \cdot \mathbf{n} = h_{\text{eq}}(T_w - T_{\text{amb}})$. At the fluid–solid interfaces, temperature continuity and heat-flux conservation were imposed, namely, $T_f = T_s$ and $-\lambda_{\text{eff}} \nabla T_f \cdot \mathbf{n} = -\lambda_s \nabla T_s \cdot \mathbf{n}$. No-slip velocity boundary conditions were applied at all solid walls. The initial condition was set as a uniform temperature field, $T_{(x,0)} = T_0$, with the initial temperature specified according to the corresponding operating condition. For the comparative simulations in Section 3, $T_0 = 400$ K was used. The initial velocity was set to zero, and the calculation was advanced until a steady state was reached. The same inlet, outlet, heating, cooling, and external wall boundary conditions were used in the pre- and post-optimization simulations. The difference between the two cases lies in the material assignment and local thermal resistance enhancement in the optimized regions, where the stagnant-air region in Segments 2 and 3 was replaced with glass wool, and local insulation was enhanced near the cooling boundaries in Segments 1 and 4.

The governing equations are discretized and solved using the finite-volume method. The SIMPLE algorithm is used for pressure–velocity coupling [26]. The three-dimensional geometric model is established using SolidWorks (2025) and then imported into ANSYS (2024) Fluent for numerical simulation. After convergence, the calculated flow-field and temperature-field data are exported for post-processing. Tecplot is used to visualize the temperature contours, extract representative sectional data, and generate the contour plots shown in the manuscript. The average temperatures of characteristic cross-sections and the heat-loss-related data are obtained by area-weighted averaging and segmented energy calculations based on the numerical results. A second-order upwind scheme is adopted for the convective terms in the momentum and energy equations, and the standard scheme is used for the pressure term. Convergence is determined jointly by equation residuals and key monitored parameters. The residual criteria are set to 10^{-4} for the continuity, momentum, and turbulence equations and 10^{-6} for the energy equation, while variations in outlet temperature and representative monitoring-point temperatures are also checked to ensure computational stability.

2.3. Mesh Generation and Independence Verification

Before mesh generation, the geometric model is preprocessed, and volume extraction is performed to generate the air domain, insulation layer, cooling water region, and other computational domains so as to satisfy the requirements of unified meshing and conjugate heat-transfer calculation. An unstructured polyhedral mesh is adopted for the entire model, with local refinement applied at the air inlet, near the heating tubes, at the hole–tube connection of the support plate, and in regions with abrupt cross-sectional change. More regular regions are meshed more uniformly to balance computational accuracy and efficiency. Three layers of boundary-layer mesh are generated near the wall, with a growth rate of 1.3. The overall mesh is shown in Figure 2a.

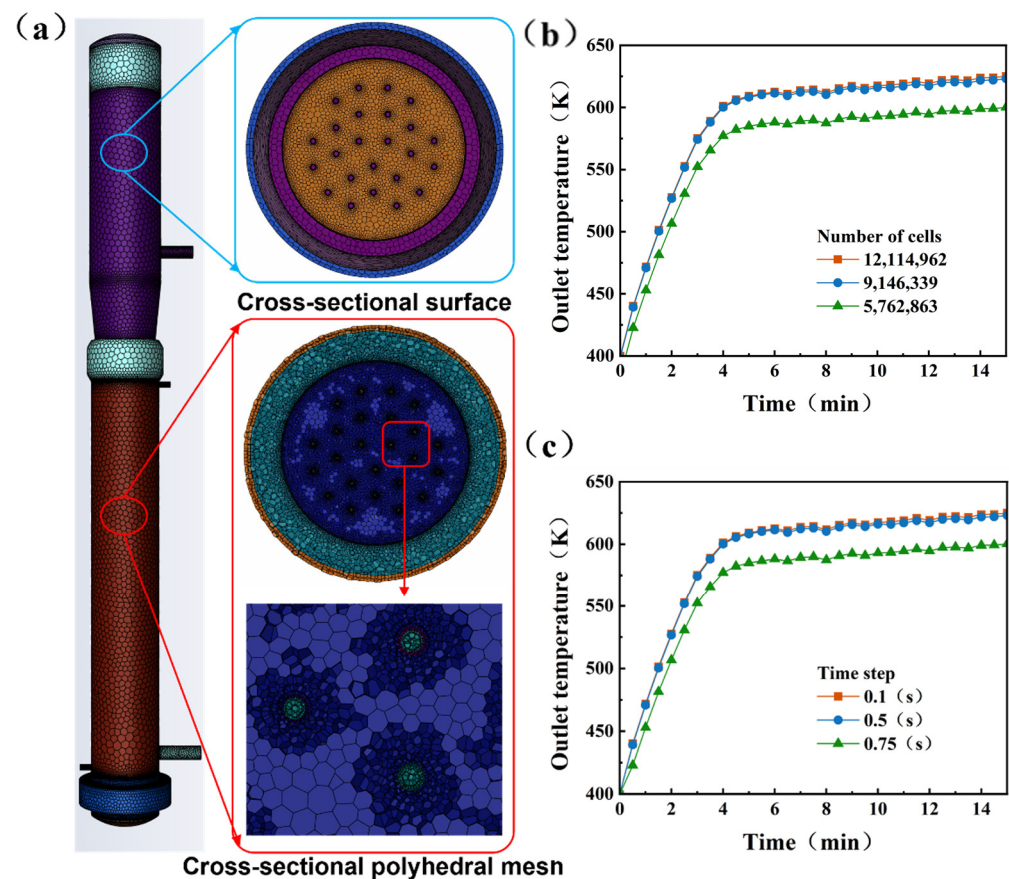


Figure 2. Mesh generation and independence verification: (a) computational mesh of the electric air heater; (b) grid independence and (c) time-step independence tests.

To ensure the appropriateness of the discretization parameters, mesh independence and time-step independence tests are conducted using the outlet air temperature as the evaluation index, and the results are shown in Figure 2b. For the mesh independence test, three meshes with 5,762,863, 9,146,339, and 12,114,962 cells are compared. When the number of cells is further increased beyond 9,146,339, the change in outlet temperature becomes very small, whereas the computational cost increases significantly. When the number of cells is reduced to 5,762,863, the deviation becomes large, approaching 5% at maximum. Therefore, 9,146,339 cells are selected as the baseline mesh. For the time-step independence test, three time-steps of 0.1 s, 0.5 s, and 0.75 s are compared. The results show that reducing the time-step further below 0.5 s causes no obvious change in outlet temperature, whereas increasing it to 0.75 s leads to an obvious increase in deviation, with a maximum approaching 4%. Accordingly, a mesh with 9,146,339 cells and a time step of 0.5 s is used in the subsequent simulations.

2.4. Model Validation

To verify the accuracy of the model, measured data from three steady operating conditions of the existing experimental platform are selected for comparison. The validation parameters include the initial temperature, heating power, flow rate, and inlet and outlet pressures, and the outlet air temperature is used as the validation index. As shown in Table 2, the simulated outlet temperatures under the three operating conditions are 612.8 K, 641.5 K, and 637.4 K, respectively, which agree well with the measured values. The maximum relative deviation is 2.17%. These results indicate that the established model can accurately reflect the heat-transfer and flow characteristics of the electric air heater.

under different operating conditions and can, therefore, be used for subsequent analyses of temperature-field characteristics, heat-loss characteristics, and retrofit performance.

Table 2. Measured data of stable operating conditions from the existing test platform for model validation.

	Power (kW)	Initial Temperature (K)	Mass Flow Rate (kg/s)	Inlet Pressure (kPa)	Actual Outlet Temperature (K)	Simulated Outlet Temperature (K)
Case 1	350	352.7	0.87	1270.8	615.5	612.8
Case 2	120	449.2	0.38	503	627.9	641.5
Case 3	505	327.1	0.98	1250.7	628.7	637.4

The experimental data used for model validation were provided by the project partner based on tests performed on the existing high-power electric air heater. No companion paper has been published for these experimental results. Owing to project confidentiality and restrictions associated with the industrial equipment, only limited test data, including inlet conditions, operating power, pressure, mass flow rate, and outlet air temperature, can be disclosed in this manuscript. Therefore, the validation is performed using the available steady-state outlet temperature data. It should be noted that this validation strategy mainly verifies the overall heat-transfer performance of the device, rather than the detailed spatial distribution of the internal temperature field and local heat loss. The lack of internal temperature and direct heat-loss measurements may introduce uncertainty into the quantitative prediction of segmental heat-loss distribution and local heat-transfer paths. Therefore, the spatial heat-loss results in this study are interpreted as model-based diagnostic results under the available validation conditions, and further experimental measurements of internal temperature profiles and local heat loss should be conducted in future work when test conditions and confidentiality constraints permit.

In addition to the boundary conditions described above, the uncertainties associated with model assumptions are considered qualitatively. The heat-loss prediction may be affected by the equivalent treatment of cooling water boundaries, the external ambient heat-loss boundary, temperature-dependent material properties, and radiative heat transfer at high temperatures. In this study, temperature-dependent thermophysical properties were used for both the fluid and solid domains to reduce the uncertainty caused by property variation. The cooling water regions were treated as equivalent constant-temperature boundaries to preserve their local cooling effect while avoiding excessive geometric complexity. The external heat loss was represented by an equivalent heat-loss boundary, which accounts for the combined influence of convection and radiation from the outer surface to the surroundings. Nevertheless, because detailed surface temperature measurements and radiative heat-flux data are not available due to experimental and confidentiality limitations, these treatments may introduce uncertainty into the quantitative prediction of local heat loss. Therefore, the segmental heat-loss results should be interpreted mainly as comparative and diagnostic indicators for identifying dominant heat-loss regions, rather than as direct measurements of absolute local heat loss. Future work should include sensitivity analyses of boundary conditions, material properties, and radiative parameters when more detailed experimental data become available.

2.5. Performance Evaluation Indices

The effective heating power of Segment *i*, used to characterize the heat actually gained by the air in Segment *i*, is defined by Equation (7):

$$\dot{Q}_{\text{eff},i} = \dot{m}(h_{\text{out},i} - h_{\text{in},i}) \quad (7)$$

where $\dot{Q}_{\text{eff},i}$ is the effective heating power of Segment i , with the unit of kW; $\dot{m}$ is the air mass flow rate, with the unit of kg/s; and $h_{\text{in},i}$ and $h_{\text{out},i}$ are the air specific enthalpies at the inlet and outlet of Segment i respectively, with the unit of kJ/kg. Therefore, the product of $\dot{m}$ and the specific enthalpy difference gives the effective heating power in kJ/s, equivalent to kW.

The enthalpy increment of air is calculated by Equation (8):

$$h_{\text{out},i} - h_{\text{in},i} \approx \int_{T_{\text{in},i}}^{T_{\text{out},i}} c_p(T) dT \quad (8)$$

The specific heat capacity at constant pressure of air is obtained from public thermo-physical property data and piecewise linearly fitted in the range of 220–1000 K to facilitate the integration of the enthalpy increment [27].

The input power of each segment is allocated in proportion to length and is given by Equation (9):

$$\dot{Q}_{\text{in},i} = \frac{\dot{Q}_{\text{total}} \cdot L_i}{L} \quad (9)$$

where $\dot{Q}_{\text{total}}$ is the total input power, with the unit of kW. L is the total length of the heating tubes, and L_i is the heating tube length of Segment i , with the unit of m.

The thermal efficiency of a segment, used to evaluate the heat utilization level of key segments such as Segments 2 and 3, is defined by Equation (10):

$$\eta_i = \frac{\dot{Q}_{\text{eff},i}}{\dot{Q}_{\text{in},i}} \times 100\% \quad (10)$$

The proportion of heat loss in Segment i , used to identify the dominant heat-loss segments, is defined by Equation (11):

$$\phi_i = \frac{\dot{Q}_{\text{loss},i}}{\sum \dot{Q}_{\text{loss},i}} \times 100\% \quad (11)$$

where $\dot{Q}_{\text{loss},i}$ is the heat loss of Segment i .

The overall thermal efficiency, used to evaluate the overall heat utilization performance of the device, is defined by Equation (12):

$$\eta = \frac{\dot{m}(h_{\text{out}} - h_{\text{in}})}{\dot{Q}_{\text{total}}} \times 100\% \quad (12)$$

where h_{in} and h_{out} are the specific enthalpies of the air at the inlet and outlet of the device, respectively.

The energy-saving ratio, used to characterize the relative reduction in total heat loss before and after retrofitting, is defined by Equation (13):

$$\psi_i = \frac{\dot{Q}_{\text{loss,before}} - \dot{Q}_{\text{loss,after}}}{\dot{Q}_{\text{loss,before}}} \times 100\% \quad (13)$$

where $\dot{Q}_{\text{loss,before}}$ and $\dot{Q}_{\text{loss,after}}$ are the total heat losses before and after optimization, respectively.

3. Results

3.1. Temperature-Field Characteristics

To analyze the heat-transfer and heat-loss characteristics under different operating conditions, three simulation cases are designed by varying only the air mass flow rate while keeping the inlet temperature, input power, and inlet and outlet pressures unchanged. The specific parameters are as follows: an initial temperature of 400 K, a power of 200 kW, an inlet pressure of 101.3 kPa, an outlet pressure at atmospheric pressure, and mass flow rates of 0.5 kg/s, 0.75 kg/s, and 1.0 kg/s, respectively.

Figure 3a–c show the temperature-field distributions and representative sectional temperatures of the electric air heater under different mass flow rates, while Figure 3d presents the variations in the average temperatures of characteristic cross-sections under different operating conditions. It can be seen that with an increasing air mass flow rate, the outlet air temperature decreases, whereas the overall temperature-field pattern inside the heater remains essentially unchanged, indicating good similarity in the heating process under different operating conditions. From the temperature distributions in different segments, Segment 1 remains close to the inlet temperature and exhibits only a small temperature rise. Owing to the influence of the top cooling boundary, a slight local temperature decrease is also observed. Upon entering Segment 2, the air temperature begins to rise significantly, indicating that this region is the initial heating segment after the air enters the heater. At this stage, an obvious temperature gradient exists within the air layer inside the shell, suggesting weak natural convection and heat transfer dominated mainly by conduction; consequently, the heat loss is relatively small. In Segment 3, the temperatures of both the main air stream and the heating tubes rise further, forming a larger high-temperature region. Meanwhile, the temperature distribution in the air layer inside the shell becomes more uniform, and more pronounced natural convection features appear in the temperature field. This indicates stronger mixing in the air layer and more significant heat exchange between the heating tubes and the outer shell, resulting in relatively larger heat loss in this segment. In Segment 4, the air temperature decreases slightly before reaching the outlet, indicating that the bottom cooling boundary has a certain weakening effect on the high-temperature air.

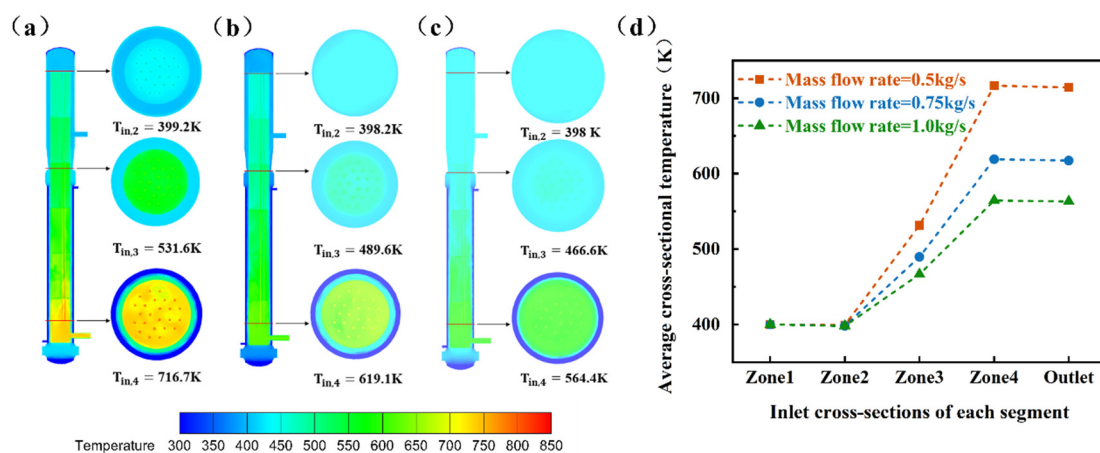


Figure 3. Simulation results: (a) mass flow rate = 0.5 kg/s, (b) mass flow rate = 0.75 kg/s, (c) mass flow rate = 1.0 kg/s, and (d) average temperatures at different sectional planes under different operating conditions.

3.2. Heat-Loss Distribution Characteristics

After identifying the main heating regions, the heat loss in each segment was further quantified. As shown in Figure 4a, the heat-loss ranking is generally Segment 3 > Segment

2 > Segment 1 $\approx$ Segment 4 under the three mass-flow-rate conditions, indicating that the heat loss is spatially non-uniform and mainly concentrated in the central heating segments. Figure 4b shows that the heat-loss proportions of different segments change only slightly with mass flow rate, suggesting that the segmental heat-loss pattern is relatively stable. Statistical results show that the average heat losses of Segments 2 and 3 are 11.9 kW and 17.41 kW, accounting for 37.26% and 54.51% of the total heat loss, respectively. Together, these two segments contribute 91.77% of the total heat loss. Therefore, Segments 2 and 3 are identified as the dominant heat-loss regions and should be preferentially considered in the subsequent structural optimization.

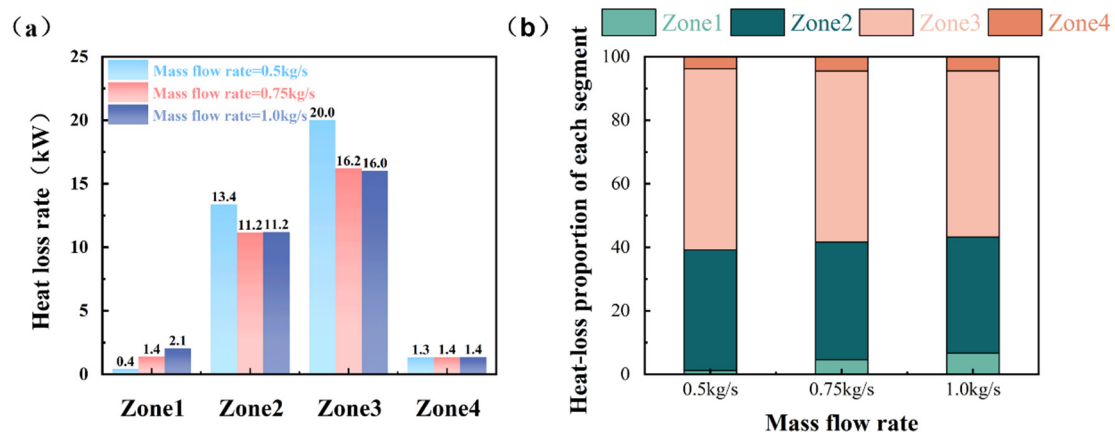


Figure 4. (a) Unit heat loss under different operating conditions, (b) heat loss proportion of each section under different operating conditions.

3.3. Optimization Scheme

Based on the identified heat-loss distribution, a local structural optimization scheme was proposed without changing the main flow channel structure or the overall geometric dimensions of the heater, as shown in Figure 5. In Segments 2 and 3, the original stagnant-air region inside the inner cylinder was replaced with glass wool to suppress natural convection, reduce the equivalent heat-transfer capacity, and weaken radial heat leakage. In Segments 1 and 4, local thermal insulation was enhanced near the cooling boundaries to increase the local thermal resistance and reduce additional heat dissipation. The detailed engineering applicability and implementation constraints of this scheme are further discussed in Sections 4.3 and 4.5.

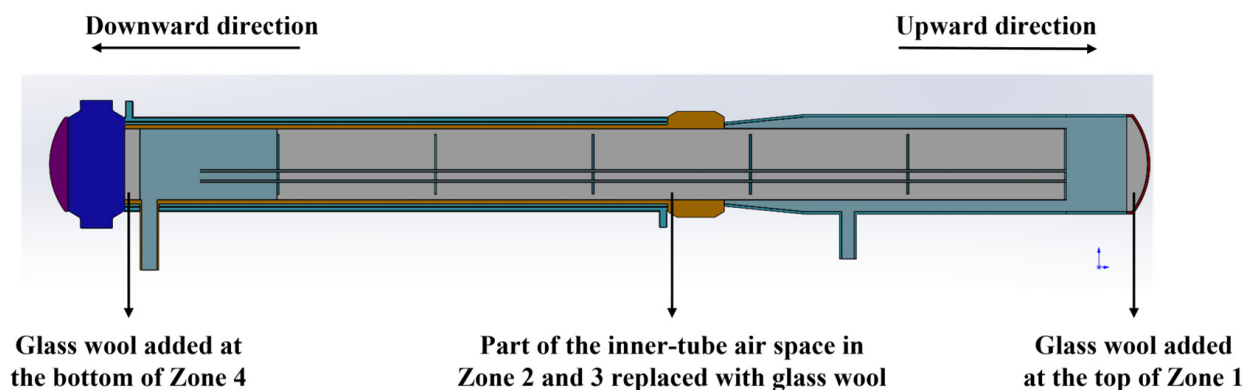


Figure 5. Schematic of the structural optimization scheme for key segments of the electric air heater.

3.4. Optimization Performance

Figure 6 compares the temperature-field distributions and average sectional temperatures before and after optimization. The overall heating pattern remains essentially

unchanged after optimization, indicating that the proposed scheme does not alter the fundamental heating process of the device. Compared with the pre-optimization case, the radial temperature gradients in the inner-cylinder region of Segments 2 and 3 are weakened, and the temperature mixing in the original stagnant-air layer is reduced. The average temperatures at the characteristic cross-sections are higher after optimization, especially in Segment 3, Segment 4, and the outlet region, indicating improved heat retention in the main airflow.

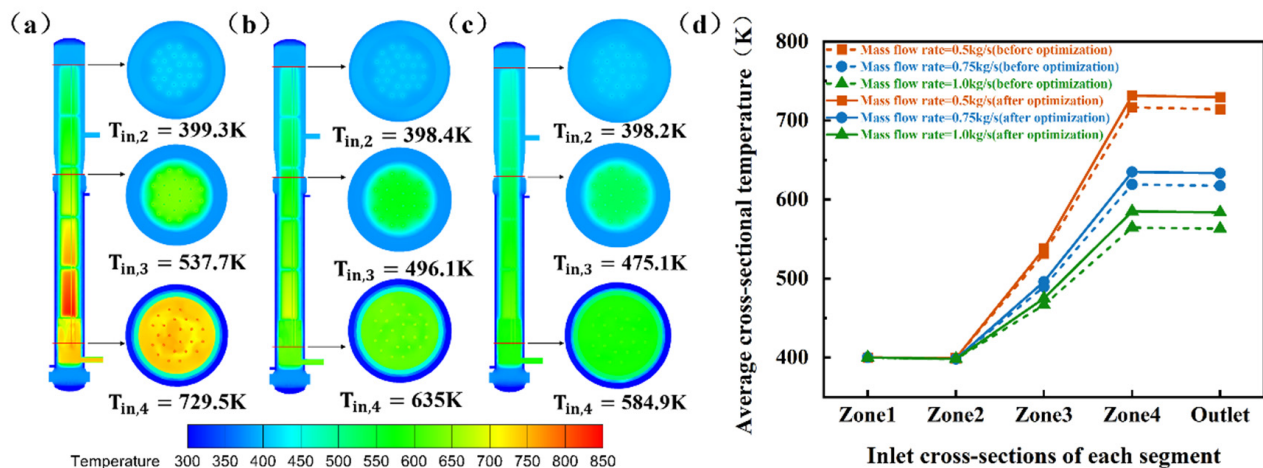


Figure 6. After-optimization simulation results: (a) mass flow rate = 0.5 kg/s, (b) mass flow rate = 0.75 kg/s, (c) mass flow rate = 1.0 kg/s, and (d) comparison of average temperatures at characteristic cross-sections before and after optimization under different operating conditions.

Figure 7 shows the reduction in heat loss after optimization. Under the three operating conditions, the total heat loss per unit time is reduced by 7.99, 12.81, and 21.77 kW, corresponding to reduction ratios of 22.74%, 42.59%, and 71.07%, respectively. On average, the total heat loss decreases from 31.94 kW to 17.69 kW, corresponding to a reduction of 44.6%. The segmented results show that the heat-loss reduction is mainly concentrated in Segments 2 and 3, with average reduction ratios of approximately 46.7% and 48.2%, respectively. These results verify that the proposed local optimization effectively reduces heat loss in the dominant heat-loss segments.

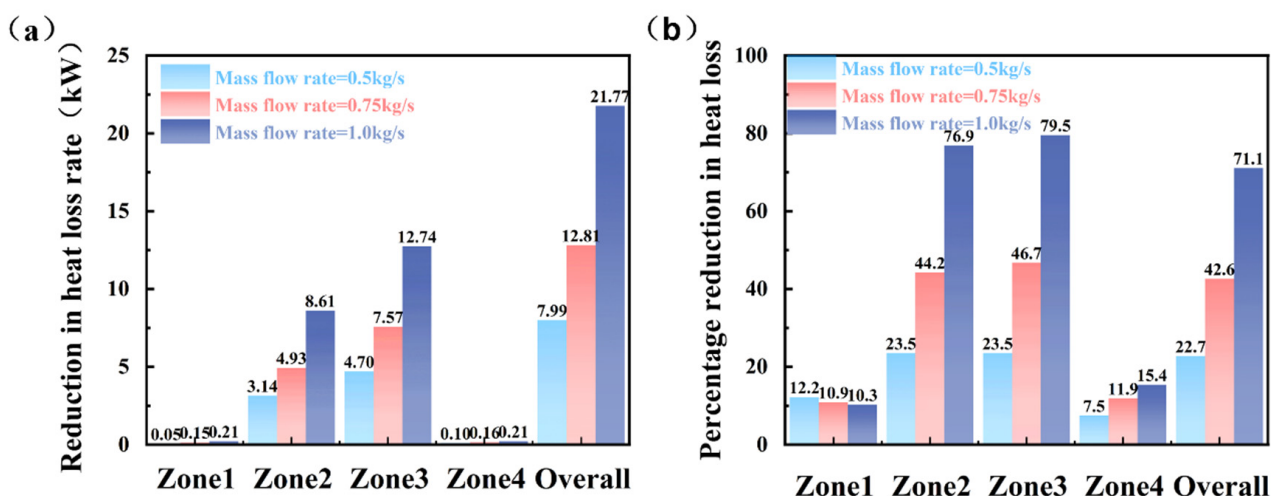


Figure 7. Reduced heat loss per unit time and reduction ratio of heat loss in each section after optimization.

In addition, the thermal efficiencies of Segments 2 and 3 increase by approximately 6.7% and 7.2%, respectively, indicating that more input electrical energy is converted into effective heat gained by the main airflow after optimization.

3.5. Economic Evaluation

In addition to the three comparative operating conditions discussed above, a supplementary simulation is performed under the rated condition, i.e., an input power of 1200 kW and an air mass flow rate of 1.0 kg/s, to better reflect the practical operating requirements of air thermal energy storage systems. The results are then used for economic evaluation.

The retrofit cost mainly includes the costs of insulation materials and auxiliary materials, lifting equipment and temporary tooling, labor and installation recovery, transportation, on-site management, and contingencies. Considering the large overall size of the electric air heater, with a height of approximately 7 m, and the fact that the retrofit process involves disassembly, lifting, and on-site recovery, the associated construction and organizational costs are substantially higher than the cost of the insulation material itself. Taking into account the equipment size, construction complexity, and project duration, the total retrofit cost is conservatively estimated to be approximately CNY 70,000 for subsequent economic evaluation.

The rated condition is adopted for the economic analysis, with an input power of 1200 kW, an air mass flow rate of 1.0 kg/s, and a single off-peak charging duration of 8 h to determine the cycle storage capacity. Figure 8a shows the temperature contours before and after optimization under the rated condition. The simulation results indicate that the outlet air temperature increases from 1421.6 K to 1482.0 K after optimization, corresponding to a reduction of 73.1 kW in total heat loss per unit time. Accordingly, the additional effective thermal storage per cycle is 584.8 kWh. For quantitative evaluation, an equivalent off-peak electricity price of CNY 0.3/(kW·h) is adopted to convert the additional stored thermal energy into an equivalent revenue per cycle. Under the rated condition, the equivalent revenue per cycle is CNY 175.44. Assuming one charge–discharge cycle per day, the corresponding investment payback period is approximately 399 d, as shown in Figure 8b. These results indicate that the proposed optimization scheme can effectively improve the heat retention capability during the charging stage without changing the main structure of the electric air heater and exhibits a certain degree of economic feasibility in air thermal energy storage applications.

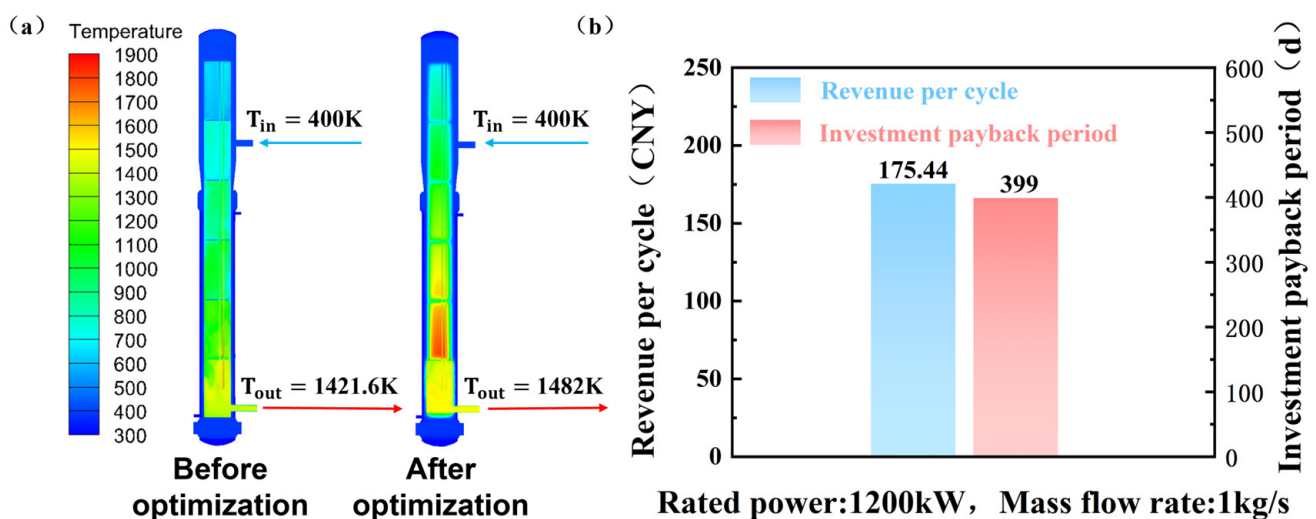


Figure 8. Economic evaluation results: (a) comparison of temperature contours before and after optimization; (b) equivalent revenue per cycle and investment payback period.

To further evaluate the uncertainty of the economic performance, a sensitivity analysis was conducted by considering variations in equivalent electricity price, daily charge–discharge cycle number, retrofit cost, and maintenance cost, as Table 3. The additional effective thermal storage per cycle is calculated as $E_{\text{add}} = \Delta \dot{Q}_{\text{loss}} \cdot \tau$, where $\Delta \dot{Q}_{\text{loss}}$ is the reduction in heat loss per unit time, and τ is the charging duration. The equivalent revenue per cycle is calculated as $R_{\text{cycle}} = E_{\text{add}} \cdot p_{\text{eq}}$ where p_{eq} is the equivalent electricity price. The payback period is estimated as $T_{\text{payback}} = \frac{C_{\text{retrofit}}}{R_{\text{cycle}} \cdot N_{\text{cycle}} - C_{\text{m}}/365}$, where C_{retrofit} is the retrofit cost, N_{cycle} is the number of charge–discharge cycles per day, and C_{m} is the annual maintenance cost. For international comparison, the equivalent values in USD are also provided using an assumed exchange rate of 1 USD = 7.2 CNY.

Table 3. Sensitivity analysis of the economic performance under different assumptions.

Case	Parameter Setting	Equivalent Revenue	Payback Period
Base case	$p_{\text{eq}} = 0.3 \text{ CNY/kWh}$, $N_{\text{cycle}} = 1$, $C_{\text{retrofit}} = 70,000 \text{ CNY}$,	175.44 CNY/cycle, 24.37 USD/cycle	399 d
Low electricity price	$p_{\text{eq}} = 0.2 \text{ CNY/kWh}$	116.96 CNY/cycle, 16.24 USD/cycle	598 d
High electricity price	$p_{\text{eq}} = 0.4 \text{ CNY/kWh}$	233.92 CNY/cycle, 32.49 USD/cycle	299 d
Half cycle per day	$N_{\text{cycle}} = 0.5$	175.44 CNY/cycle, 24.37 USD/cycle	798 d
Two cycles per day	$N_{\text{cycle}} = 2$	175.44 CNY/cycle, 24.37 USD/cycle	200 d
Lower retrofit cost	$C_{\text{retrofit}} = 50,000 \text{ CNY}$	175.44 CNY/cycle, 24.37 USD/cycle	285 d
Higher retrofit cost	$C_{\text{retrofit}} = 90,000 \text{ CNY}$	175.44 CNY/cycle, 24.37 USD/cycle	513 d
5% annual maintenance	$C_{\text{m}} = 3500 \text{ CNY/year}$	175.44 CNY/cycle, 24.37 USD/cycle	422 d
10% annual maintenance	$C_{\text{m}} = 7000 \text{ CNY/year}$	175.44 CNY/cycle, 24.37 USD/cycle	448 d

The sensitivity analysis indicates that the payback period is sensitive to electricity price, daily operating frequency, retrofit cost, and maintenance cost, which are further discussed as economic uncertainties in Section 4.5.

4. Discussion

4.1. Claim–Proof Relationship and Evidence Chain

The preceding sections present the model validation, baseline heat-loss diagnosis, retrofit design, and performance evaluation of the studied electric air heater. To explicitly clarify the relationship between the main claims of this study and their supporting evidence, Table 4 summarizes how each claim is substantiated by the corresponding model verification, segmental heat-loss calculation, retrofit comparison, and economic evaluation in the core sections of this manuscript.

Table 4. Main claims of this study and corresponding evidence in this manuscript.

Main Claim	Supporting Evidence	Key Proof	Location
A validated 3D conjugate heat-transfer model is established	Geometry, governing equations, boundary conditions, mesh/time-step tests, and experimental validation	Baseline mesh: 9,146,339 cells; time step: 0.5 s; maximum outlet-temperature deviation: 2.17%	Sections 2.1–2.4; Figures 1 and 2; Table 2
Segment-based energy analysis reveals non-uniform heat loss	Four-segment division and segmental heat-loss calculation	Heat-loss ranking remains Segment 3 > Segment 2 > Segment 1 $\approx$ Segment 4 under different mass flow rates	Section 2.1, Section 2.5, and Section 3.2; Figure 4
Segments 2 and 3 are dominant heat-loss regions	Segmental heat-loss proportions and heat-loss mechanism analysis	Segments 2 and 3 together account for 91.77% of total heat loss	Section 3.2; Figure 4
A targeted local retrofit scheme is proposed based on identified heat-loss paths	Glass-wool filling in Segments 2 and 3 and insulation enhancement near cooling boundaries in Segments 1 and 4	The scheme suppresses natural convection in the stagnant-air region and weakens radial heat leakage without changing the main flow channel structure	Section 3.3; Figure 5
Retrofit performance is verified thermally and economically	Pre- and post-optimization temperature fields, heat losses, segmental reductions, rated condition simulation, and sensitivity analysis	Average total heat loss decreases from 31.94 kW to 17.69 kW; rated condition additional effective thermal storage reaches 584.8 kWh per cycle; base payback period is about 399 d	Sections 3.4 and 3.5; Figures 6–8; Table 3

4.2. Interpretation of the Dominant Heat-Loss Mechanism

The dominant heat-loss regions are not determined only by the local input power, but also by the coupled effects of streamwise heating, radial temperature gradients, stagnant-air heat transfer, and local cooling boundaries. Segment 2 is the region where the air begins to be significantly heated after entering the heating tube bundle, and the temperature difference between the internal heating region and external structures increases rapidly. Segment 3 contains the developed high-temperature region, where both the air temperature and heating tube temperature are higher. In this region, thermal mixing in the stagnant-air layer increases the equivalent heat-transfer capability and promotes radial heat diffusion toward the outer shell.

Segments 1 and 4 show smaller heat-loss contributions because their thermal environments are different. Segment 1 is close to the inlet and top cooling boundary, which limits its temperature level and radial heat-transfer driving force. Segment 4 is close to the outlet but is affected by the bottom cooling boundary and limited effective heat-transfer length. The stable heat-loss ranking under different mass flow rates indicates that the dominant heat-loss pattern is mainly governed by structural segmentation, airflow path, and local thermal boundaries.

The post-optimization results further support this mechanism. After glass-wool filling, the radial temperature gradients and thermal mixing in the original stagnant-air region of Segments 2 and 3 are weakened, and the heat-loss reduction is mainly concentrated in these two segments. This consistency between the diagnosed dominant loss regions and the post-optimization reduction pattern confirms that the stagnant-air region and associated radial heat-transfer path are key targets for heat-loss control.

4.3. Engineering Significance of Targeted Local Retrofit

The engineering significance of the proposed retrofit lies in its diagnosis-guided and low-intrusion nature. For existing high-power electric air heaters, large-scale reconstruction of the shell, heating tube bundle, support plates, or main flow channel is usually constrained by the original structural layout, equipment size, maintenance accessibility, and on-site construction conditions. Therefore, simply increasing the overall insulation thickness may increase construction difficulty and cost without necessarily addressing the dominant heat leakage path.

In the present study, the retrofit region is selected according to the quantified segmental heat-loss distribution. Segments 2 and 3 are treated as the priority regions because they account for the dominant part of the original heat loss, whereas Segments 1 and 4 are only locally enhanced near the cooling boundaries. This strategy increases the thermal resistance of the main radial heat leakage path while retaining the original flow channel structure and overall geometric dimensions of the heater.

It should be emphasized that the significance of the retrofit scheme does not lie in glass wool as an insulation material itself, but in the diagnostic basis used to determine where the local thermal resistance should be increased. Compared with indiscriminate overall insulation thickening, identifying dominant heat-loss regions first and then strengthening local thermal resistance is more suitable for existing equipment whose main structure cannot be significantly changed.

4.4. Comparison with Previous Studies

As summarized in Table 1, previous studies have addressed thermal energy storage, operational control, industrial heating equipment optimization, retrofit measures, and CFD-based thermal diagnosis from different perspectives. The present work differs from system-level studies on air thermal energy storage because it focuses on the component-level heat-loss behavior of the electric air heater. It also differs from operational control studies because the objective is not only to regulate outlet temperature or flow rate, but to identify and weaken the internal structural heat-leakage paths of an existing heater.

Compared with studies on industrial ovens, air heaters, and general retrofit measures, the present study emphasizes that retrofit design should be guided by quantified internal heat-loss distribution rather than by empirical or uniform insulation enhancement. The results indicate that, for high-power electric air heaters or similar industrial thermal equipment with internal heating tube bundles and local cooling boundaries, heat loss may be strongly concentrated in limited structural regions rather than uniformly distributed throughout the whole device.

Therefore, the broader value of this work lies in the diagnostic retrofit framework. The combination of validated three-dimensional conjugate heat-transfer modeling, segmented heat-loss evaluation, dominant heat-loss region identification, and local thermal-resistance enhancement provides a practical methodology for heat-loss diagnosis and targeted retrofit design of similar existing thermal equipment. In this sense, the proposed approach extends beyond the specific glass-wool filling scheme and provides a basis for selecting retrofit regions in other high-power heating devices with comparable structural constraints.

4.5. Limitations and Future Work

Several limitations should be noted. First, although the numerical model was validated using three steady-state experimental cases, the validation was mainly based on outlet air temperature because detailed internal temperature profiles and local heat-flux measurements were unavailable due to industrial equipment and confidentiality constraints.

Therefore, the segmental heat-loss results should be interpreted mainly as model-based diagnostic indicators rather than direct measurements of absolute local heat loss.

Second, the numerical prediction may be affected by the equivalent treatment of cooling water boundaries, the external ambient heat-loss boundary, temperature-dependent material properties, and high-temperature radiative heat transfer. Although these treatments preserve the main thermal boundary effects and reduce geometric complexity, future work should include sensitivity analyses of boundary conditions, material properties, and radiation parameters when more detailed experimental data become available.

Third, the proposed optimization scheme is developed for the energy-saving retrofit of an existing high-power electric air heater under the premise that the original main flow-channel structure and overall geometric dimensions remain unchanged. Therefore, it is more suitable for local thermal resistance enhancement and heat-loss control of existing equipment, while its direct applicability to the overall structural design of newly built electric air heaters is still limited.

Fourth, the glass-wool filling and local insulation enhancement scheme has not yet been experimentally implemented on the actual device. In practical engineering applications, equipment disassembly, limited internal construction space, insulation material fixation, prevention of material intrusion into the main airflow passage, long-term high-temperature stability, start-up and shutdown thermal cycling, and operational vibration should be further considered. High-temperature-resistant metal mesh, retaining rings, or segmented limiting structures may be required to ensure the stability and safety of the filled insulation material.

Finally, the economic evaluation should be regarded as a preliminary engineering estimate. The payback period depends on the equivalent electricity price, daily charge-discharge frequency, retrofit cost, and maintenance cost. Future work should combine internal temperature measurements, local heat-flux monitoring, engineering operation data, long-term reliability tests, and detailed cost accounting to further verify the energy-saving performance and economic feasibility of the proposed retrofit scheme. For newly built high-power electric air heaters, future optimization may also consider the integrated design of heating tube arrangements, flow channel configuration, cooling boundary locations, inner- and outer-cylinder insulation structures, and thermal bridge control of support components.

5. Conclusions

A three-dimensional conjugate heat-transfer model was established for an existing 1200 kW high-power electric air heater and validated using three steady-state experimental cases. The maximum outlet-temperature deviation was 2.17%, indicating that the model can be used for subsequent heat-transfer and heat-loss diagnosis. The segmental analysis shows that the heat loss is strongly non-uniform. Segments 2 and 3 are the dominant heat-loss regions, with average heat losses of 11.9 kW and 17.41 kW, accounting for 37.26% and 54.51% of the total heat loss, respectively. Together, they contribute 91.77% of the total heat loss.

Based on the identified heat-loss distribution, a targeted local retrofit scheme was proposed by filling the non-flowing air region of the inner cylinder with glass wool in Segments 2 and 3 and enhancing insulation near local cooling boundaries in Segments 1 and 4. After optimization, the fundamental heating pattern of the device remains essentially unchanged, while the heat retention capability is improved. The average total heat loss decreases from 31.94 kW to 17.69 kW, corresponding to a reduction of 44.6%, and the thermal efficiencies of Segments 2 and 3 increase by approximately 6.7% and 7.2%, respectively.

Under the rated condition of 1200 kW and 1.0 kg/s with an 8 h off-peak charging period, the outlet temperature increases from 1421.6 K to 1482.0 K after optimization, and

the total heat loss is reduced by 73.1 kW. This corresponds to an additional effective thermal storage of 584.8 kWh per cycle. Based on an equivalent off-peak electricity price of CNY 0.3/(kW·h), the equivalent revenue is CNY 175.44 per cycle, and the estimated payback period is approximately 399 d under one charge–discharge cycle per day. These results indicate that the proposed diagnostic retrofit approach can provide a useful reference for heat-loss reduction and energy-saving optimization of existing high-power electric air heaters in air thermal energy storage systems.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

c_p	Specific heat capacity at constant pressure (kJ/(kg·K))	Greek letters	
c_s	Specific heat capacity of the solid (kJ/(kg·K))	ε	Turbulent dissipation rate (m ² /s ³)
C_1, C_2	Empirical constants in the standard $k - \varepsilon$ model	η	Overall thermal efficiency of the heater (%)
G_k	Production term of turbulent kinetic energy (kg/(m·s ³))	η_i	Thermal efficiency of Segment i (%)
h	Specific enthalpy of air (kJ/kg)	λ_{eff}	Effective thermal conductivity (W/(m·K))
h_{in}, h_{out}	Specific enthalpies of air at the inlet and outlet of the device (kJ/kg)	λ_s	Thermal conductivity of the solid (W/(m·K))
$h_{out,i}, h_{in,i}$	Specific enthalpies of air at the inlet and outlet of Segment i (kJ/kg)	μ	Dynamic viscosity (Pa·s)
k	Turbulent kinetic energy (m ² /s ²)	μ_{eff}	Effective dynamic viscosity (Pa·s)
L	Total length of the heating tubes (m)	μ_t	Turbulent dynamic viscosity (Pa·s)
L_i	Heating tube length of Segment i (m)	ρ	Fluid density (kg/m ³)
$\dot{m}$	Air mass flow rate (kg/s)	ρ_s	Solid density (kg/m ³)
p	Pressure (Pa)	σ_k	Turbulent Prandtl number for k
$\dot{Q}_{eff,i}$	Effective heating power of Segment i (kW)	σ_ε	Turbulent Prandtl number for ε
$\dot{Q}_{in,i}$	Input power allocated to Segment i (kW)	ϕ_i	Heat-loss proportion of Segment i (%)
$\dot{Q}_{loss,i}$	Heat loss of Segment i (kW)	ψ_i	Energy-saving ratio (%)
$\dot{Q}_{loss,before}$	Total heat loss before optimization (kW)	Subscript	
$\dot{Q}_{loss,after}$	Total heat loss after optimization (kW)	i	Segment index
$\dot{Q}_{total}$	Total input power of the heater (kW)	in	Inlet
t	Time (s)	out	Outlet
T	Temperature (K)	eff	Effective
$T_{in,i}, T_{out,i}$	Air temperatures at the inlet and outlet of Segment i (K)	$total$	Total value
Abbreviations		$total$	Loss
CFD	Computational fluid dynamics	$before$	Before optimization
FVM	Finite-volume method	$after$	After optimization
SIMPLE	Semi-implicit method for pressure-linked equations	s	Solid domain

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