

Article

Research on Dynamic Leakage Fault Characteristics of High-Pressure Heater Under Wide-Load Conditions of Coal-Fired Units

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Abstract

Tube bundle leakage occurs in high-pressure heaters of coal-fired units after long periods of variable-load operation. As these units increasingly participate in deep peak regulation, understanding leakage behavior over a wide load range becomes practically important. In this work, static and dynamic thermodynamic models of a three-section high-pressure heater were built for a 600 MW unit and checked against available design data of the No. 1 heater. For feedwater outlet temperature, the relative error stays below 1%. The dynamic model then simulates transient responses of feedwater outlet temperature, metal tube wall temperature, and shell-side drain volume under micro-leakage scenarios (1%, 5%, and 10% of feedwater flow) from turbine heat acceptance (THA) to 30% THA. Under THA, the final decreases in feedwater outlet temperature are about 1 K, 4.6 K, and 8.4 K for the three leakage levels, and similar patterns appear at partial loads. Taken together, the changes in temperature drop and drain volume can provide trend-level reference signals for engineering judgment within the studied operating conditions.

Keywords: coal-fired units; high-pressure heater; dynamic characteristics; wide-load operation

1. Introduction

With the rapid development of renewable energy power generation, the large-scale renewable energy grid connection brings undeniable challenges to the safety and stability of the power grid. To enhance the capability of the grid for renewable energy integration, it is necessary for coal-fired power units to adjust their load to participate in deep peak regulation [1]. As critical auxiliary equipment in coal-fired power plants [2], high-pressure heaters have a significant impact on operational safety and economic performance. Tube-bundle leakage in high-pressure heaters is a common issue after long-term operation [3]. Due to prolonged exposure to high temperature and pressure differentials, these heaters are prone to several failure modes, among which inner tube-bundle leakage is one of the most common and serious [4].

At present, research on the modeling and simulation of high-pressure heater leakage has achieved considerable progress. Yang et al. [5] adopted a three-section modeling method to establish a mathematical model of a high-pressure heater under variable operation conditions based on water level effects, quantitatively analyzing the relationships among unit load, heater water level, and upper/lower terminal temperature differences.



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Ni [6] conducted a systematic review of the causes of high-pressure heater leakage and corresponding preventive measures in Tuotetuo power plant. Through a simulation analysis of the No. 1 high-pressure heater of a 300 MW unit, Quan [7] reported operating data and dynamic response data after leakage occurred. Ma et al. [8] proposed an approach integrating symptom zoom optimization and a neural network to diagnose high-pressure heater faults of varying severity in the 70–100% rated-load range.

In summary, the existing studies mainly focused on the leakage response characteristics under medium and high load conditions, while there are relatively fewer studies on the dynamic characteristics of micro-leakage under the condition of wide-load deep peak shaving. This research takes the No. 1 high-pressure heater of a power plant as an example, establishes a three-stage static and dynamic thermodynamic model, and conducts micro-leakage dynamic analysis within the range of 30–100% THA. Based on this, the study quantifies the transient dynamic responses of feedwater outlet temperature, metal tube wall temperature, and shell-side drainage volume under different levels of micro-leakage severity, reveals the dependency of leakage signal manifestations on load, and clarifies the potential thermodynamic mechanism of parameter changes caused by leakage. This research provides a theoretical basis for fault diagnosis of high-pressure heaters in deep load-shaving operation of thermal power generating units and the formulation of targeted leakage detection strategies.

2. Three-Section High-Pressure Heater

This study is based on the feedwater high-pressure heater of a power plant. The feedwater high-pressure heater is a critical part of the feedwater regenerative system, which includes 3 high-pressure heaters, a deaerator, and 3 low-pressure heaters. In this paper, the No. 1 high-pressure heater is selected as the case study. Based on the physical state of the working fluid on the shell side, the model is simplified into a three-section model [9,10]. The schematic diagram of the mechanism model for the high-pressure heater is shown in Figure 1.

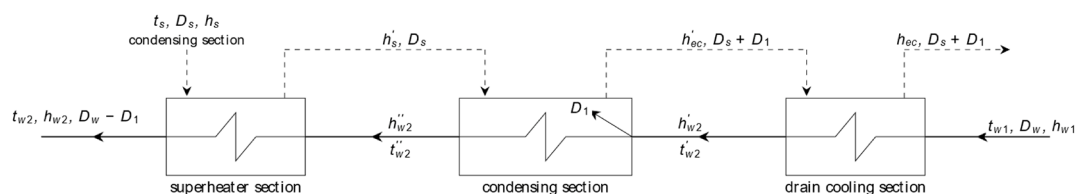


Figure 1. Schematic diagram of the mechanism model of a high-pressure heater (Dashed line: Extraction steam and its resulting condensate water; Solid line: Feedwater).

3. Mathematical Model of High-Pressure Heater

3.1. Fundamental Assumptions

Before establishing the mathematical model of high-pressure heater, several fundamental assumptions are proposed to simplify the model, thus ensuring the simulation problems are easy to analyze:

1. The pressure on the shell side of the high-pressure heater is assumed to be constant and uniform.
2. The density of the feedwater on the tube side is assumed to be constant, determined by the value at the inlet of the tube.
3. The temperature of the tube wall is assumed to be uniform and the bundle of parallel heating tubes is simplified and treated as a single equivalent tube. The heat transfer is considered only through the tube wall, while heat transfer along the tube wall is neglected.

4. The impact of non-condensable gas on condensation heat transfer is neglected.
5. The metal of the tube wall is assumed to participate in the heat transfer process as well as the thermal storage effect.
6. The heat transfer areas of the superheater section, condensing section and drain cooling section will change depending on the load of the unit and other conditions.

It is further assumed that leakage occurs only in the condensing section and that the drain-valve opening remains constant during the simulated initial leakage stage. These assumptions are based on engineering practice: the condensing section has the largest heat-transfer area and typically experiences severe thermal-hydraulic conditions, so it is a representative leakage-prone location. A constant drain-valve opening represents the early period before automatic water-level control or operator intervention changes the valve position. Alternative leakage locations or valve responses may change the response amplitude, but they are not expected to change the main trends discussed in this study.

3.2. The Static Model of High-Pressure Heater

The simulation model is established based on a horizontal, three-section, U-tube high-pressure heater, as depicted in Figure 1. Given that the heat transfer area of condensing section is largest among all of sections, it is assumed that leakage occurs in this section, and the opening of drain regulating valve remains constant. The heat balance equation of the steam cooling section is shown in Equation (1):

$$(D_w - D_1)(h_{w2} - h''_{w2}) = D_s(h_s - h'_s) \quad (1)$$

where D_w is the feedwater flow rate of the high-pressure heater (kg/s); D_1 is the leakage flow rate from high-pressure heater (kg/s); h_{w2} is the enthalpy of the feedwater at the outlet of the high-pressure heater (J/kg); h''_{w2} is the enthalpy of the feedwater at the outlet of the steam condensing section (J/kg); D_s is the extraction steam flow rate (kg/s); h_s is the enthalpy of the extraction steam at the inlet of high-pressure heater (J/kg); and h'_s is the enthalpy of the extraction steam at the outlet of the steam cooling section (J/kg).

Heat balance equation of steam condensing section is shown in Equation (2):

$$(D_w - D_1)(h''_{w2} - h'_{w2}) + D_1(h_{ec} - h'_{w2}) = D_s(h'_s - h'_{ec}) \quad (2)$$

where h''_{w2} is the enthalpy of the feedwater at the outlet of drain cooling section (J/kg); h_{ec} is the enthalpy of the high-pressure heater drains; h'_{ec} is the enthalpy of the condensate at the outlet of the condensing section (J/kg).

Heat balance equation of drain cooling section is given by Equation (3):

$$D_w(h'_{w2} - h_{w1}) = (D_s + D_1)(h'_{ec} - h_{ec}) \quad (3)$$

where h_{w1} is the enthalpy of the feedwater at the inlet of the high-pressure heater (J/kg).

Following derivation, the overall heat balance equation of the high-pressure heater is given by Equation (4):

$$(D_w - D_1)(h_{w2} - h_{w1}) + D_1(h'_{ec} - h_{w2}) = D_s(h_s - h_{ec}) \quad (4)$$

Feedwater kinematic viscosity is defined by Equation (5):

$$v_w = \frac{\eta}{\rho} \quad (5)$$

where η is dynamic viscosity of water ($\text{N}\cdot\text{s}/\text{m}^2$); ρ is the density of the feedwater (kg/m^3); and v_w is feedwater kinematic viscosity (m^2/s).

Flow velocity in tube is defined by Equation (6):

$$V_{\text{in}} = \frac{4D_w}{n\rho\pi d_i^2} \quad (6)$$

where V_{in} is the flow velocity inside the feedwater tube (m/s); D_w is feedwater flow rate (kg/s); n is the number of feedwater tubes; and d_i is the inner diameter of the tubes (m).

Reynolds number is defined by Equation (7):

$$Re_w = \frac{d_i V_{\text{in}}}{v_w} \quad (7)$$

where Re_w is the Reynolds number of feedwater.

Thermal diffusivity of the feedwater is defined by Equation (8):

$$a = \frac{\lambda_w}{\rho c_p} \quad (8)$$

where a is thermal diffusivity of the feedwater (m^2/s); c_p is specific heat capacity at constant pressure ($\text{J}/(\text{kg}\cdot\text{K})$); and λ_w is thermal conductivity of the feedwater ($\text{W}/(\text{m}\cdot\text{K})$).

Prandtl number is defined by Equation (9):

$$Pr_w = \frac{v_w}{a} \quad (9)$$

where Pr_w is the Prandtl number of feedwater.

Nusselt number is defined by Equation (10):

$$Nu_w = 0.023 \times Re_w^{0.8} \times Pr_w^{0.4} \quad (10)$$

where Nu_w is the Nusselt number of feedwater.

Tube-Side Heat Transfer Coefficient is defined by Equation (11):

$$h_i = \frac{Nu_w \lambda}{d_i} \quad (11)$$

The calculation process of shell-side heat transfer coefficient (h_o) is similar to that of tube-side heat transfer coefficient (h_i), and thus the relevant equations are not repeated here.

Overall heat transfer coefficient is expressed by Equation (12):

$$U = \frac{1}{\frac{1}{h_o} + R_s + \frac{d_o}{2\lambda_w} \ln\left(\frac{d_o}{d_i}\right) + R_w \frac{d_o}{d_i} + \frac{1}{h_i} \frac{d_o}{d_i}} \quad (12)$$

where h_o is shell-side heat transfer coefficient ($\text{W}/(\text{m}^2\cdot\text{K})$); R_s is the fouling thermal resistance of shell-side and R_w is the fouling thermal resistance of tube-side. According to the recommended values from American heat transfer standards [11], R_s is set to $5.2833 \times 10^{-5} (\text{m}^2\cdot\text{K})/\text{W}$ and R_w is set to $3.5222 \times 10^{-5} (\text{m}^2\cdot\text{K})/\text{W}$.

3.3. The Dynamic Model of High-Pressure Heater

Shell-side heat transfer equation for high-pressure heater is expressed by Equation (13):

$$Q_s = K_s A_s (t_s - t_m) \quad (13)$$

where Q_s is the heat transfer rate between shell-side steam and metal tube wall (W); K_s is shell-side convection heat transfer coefficient (W/(m²·K)); A_s is shell-side metal heat exchange tube area (m²); t_s is shell-side average steam temperature, K; and t_m is average temperature of heat exchange tube wall (K).

Tube-side heat transfer equation for high-pressure heater is expressed by Equation (14):

$$Q_w = K_w A_w \left(t_m - \frac{t_{w1} + t_{w2}}{2} \right) \quad (14)$$

where Q_w is heat transfer between tube-side feedwater and the tube wall on the tube side (W); K_w is tube-side convection heat transfer coefficient (W/(m²·K)); A_w is tube-side metal heat exchange tube area (m²); t_{w1} is the inlet water temperature of the high-pressure heater; t_{w2} is the outlet water temperature of the high-pressure heater; and $\frac{t_{w1} + t_{w2}}{2}$ denotes the arithmetic mean of the inlet and outlet feedwater temperatures.

Shell-side volume conservation equation for high-pressure heater is given by Equation (15):

$$V' + V'' = \text{const} \quad (15)$$

where V' is shell-side drainage volume for high-pressure heater (m³); V'' is shell-side steam volume for high-pressure heater (m³).

Shell-side mass conservation equation for high-pressure heater is expressed by Equation (16):

$$\frac{d(V''\rho'')}{d\tau} + \frac{d(V'\rho')}{d\tau} = \sum D \quad (16)$$

where ρ'' is the saturated steam density of high-pressure heater (kg/m³); ρ' is the saturated water density (kg/m³); τ is operating time (s); and $\sum D$ is the sum of shell-side inlet and outlet water mass (kg/s).

Heat storage equation for the metal tube wall of high-pressure heater is expressed by Equation (17):

$$M_m c_m \frac{dt_m}{d\tau} = Q_s - Q_w \quad (17)$$

where M_m is the mass of metal tube wall (kg); c_m is specific heat capacity of the metal tube wall (J/(kg·K)).

Shell-side energy conservation equation for high-pressure heater is expressed by Equation (18):

$$\frac{d(V''\rho''h'')}{d\tau} + \frac{d(V'\rho'h')}{d\tau} = \sum Dh - Q_s \quad (18)$$

where h'' is the saturated steam enthalpy of high-pressure heater (J/kg); h' is the saturated water enthalpy of high-pressure heater (J/kg).

Tube-side energy conservation equation for high-pressure heater is expressed by Equation (19):

$$m_w c_p \frac{dt_{w2}}{d\tau} = Q_w - c_p (D_w - D_1)(t_{w2} - t_{w1}) \quad (19)$$

where m_w is the feedwater mass of high-pressure heater (kg).

By simplifying the aforementioned equations, the following equations are obtained:
Feedwater outlet temperature variation rate equation for high-pressure heater:

$$\frac{dt_{w2}}{d\tau} = \frac{Q_w - c_p (D_w - D_1)(t_{w2} - t_{w1}) - D_1(h_1 - h_{w1})}{c_p m_w} \quad (20)$$

Tube wall temperature variation rate equation for high-pressure heater:

$$\frac{dt_m}{d\tau} = \frac{Q_s - Q_w}{m_m c_m} \quad (21)$$

Shell-side operation pressure variation rate equation for high-pressure heater:

$$\frac{dp_s}{d\tau} = \frac{\sum Dh - Q_s - A \sum D}{B + C} \quad (22)$$

where $A = \frac{h'\rho' - h''\rho''}{\rho' - \rho''}$; $B = V''[\rho'' \frac{dh''}{dp_s} + \frac{\rho'(h'' - h')}{\rho' - \rho''} \frac{d\rho''}{dp_s} - 1]$; $C = V'[\rho' \frac{dh'}{dp_s} + \frac{\rho''(h' - h'')}{\rho' - \rho''} \frac{d\rho'}{dp_s} - 1]$.
Shell-side drainage volume variation rate equation for high-pressure heater:

$$\frac{dV'}{d\tau} = \frac{\sum D - \left(V' \frac{d\rho'}{d\tau} - V'' \frac{d\rho''}{d\tau}\right) \frac{dp_s}{d\tau}}{\rho' - \rho''} \quad (23)$$

3.4. Simulation Conditions

The high-pressure heater analyzed in this paper is based on the No. 1 three-section unit from a 600 MW power plant. A simulation model of this heater was developed in Simulink by employing the static and dynamic simulation equations mentioned above. The leakage fault D1 is modeled as a sudden bypass flow from the tube side to the shell side, activated at a specified simulation time. The simulation equations are derived from mass and energy conservation. Model consistency is checked by steady-state comparison with design values, as presented in the validation section. Table 1 presents the technical parameters used in the simulation model.

Table 1. Technical parameters for a high-pressure heater.

Parameter	Tube Side	Shell Side	Unit
Heat Transfer Area		2063	m ²
Design Pressure	31	7.47	MPa
Design Temperature	563	693/563	K
Max. Working Pressure	31	7.47	MPa
Operating Pressure	19.66	6.153	MPa
Operating Temperature	550.6	656.6/550.2	K
Number of Tubes		2634	(count)
Tube Outer Diameter		16	mm
Tube Wall Thickness		2.11	mm
Shell-Side Water Volume in Drain Cooling Section (at NWL *)		2.0	m ³
Tube-Side Water Volume in Drain Cooling Section (at NWL *)		2.3	m ³
Shell-Side Steam Volume (during operation)		18.3	m ³
Tube-Side Water Volume (during operation)		4.9	m ³

* NWL: Normal Water Level.

Moreover, the upper and lower terminal temperature differences in the No. 1 high-pressure heater have design values of −1.6 K and 5.6 K, respectively. For the simulation, the normal operating water level is set as the initial condition under all load conditions.

The leakage flow rate D_1 was implemented in Simulink by a Step module and a Gain module, which was set as the percentage of feedwater flow rate D_w (1%, 5% and 10% for simulation, respectively). The opening of the drain regulating valve was kept constant at 100% during the whole dynamic simulation without any adjustment. The initial shell side volumes were consistent under all load conditions, with the steam volume being 18.3 m³

and the water volume in the drain cooling section at normal water level being 2.0 m³. The initial shell side operating pressure varied linearly with the unit load, which was consistent with the extraction steam pressure at different loads (6.153 MPa for THA, 4.402 MPa for 75%THA, 2.985 MPa for 50%THA, 2.446 MPa for 40%THA and 1.908 MPa for 30%THA). All dynamic simulations took the normal operating water level as the unified initial boundary condition. Key numerical solution settings are summarized in Table 2 for clarity.

Table 2. Numerical solution settings and verification.

Parameter	Symbol	Value/Setting	Description
Solver type	—	ode15s (variable-step stiff solver)	Suitable for slow thermal-hydraulic dynamic processes
Relative error tolerance	—	10 ^{−6}	Relative error tolerance
Minimum time step	Δt_{min}	0.01 s	Prevents numerical instability in dynamic simulation
Maximum time step	Δt_{max}	1 s	Balances calculation accuracy and computational efficiency
Total simulation duration	T_{sim}	500 s	Covers the full dynamic response process of the system
Stability check criterion	—	No oscillation in shell side pressure/metal wall temperature	Confirms stable numerical computation
Convergence verification method	—	Time step independence test	Compares results with different time steps (0.001/0.01/0.1/1 s)
Convergence tolerance	—	<0.1 K	Allowable deviation of feedwater outlet temperature steady-state value

3.5. Model Validation

A comparison between the simulated feedwater outlet temperatures under non-leakage conditions from the static and dynamic models and the theoretical design value was performed across various loads.

As shown in Table 3, under different unit loads and thermal conditions at the inlet and outlet of high-pressure heaters, the absolute difference between simulated and design values for feedwater outlet temperature remains within 4 K, and the relative error is below 1%. For the scope of this study, this level of agreement is sufficient for trend analysis. The model is therefore used to describe wide-load behavioral trends of the high-pressure heater. Because plant transient data are limited, dynamic validation here is confined to trend consistency rather than one-to-one transient reproduction.

Table 3. Simulation model validation results.

Load Condition	Feedwater Inlet Temperature	Feedwater Outlet Temperature (Design Value)	Feedwater Outlet Temperature (Simulated Value)	Relative Error	Upper Terminal Difference
Unit	K	K	K	%	K
THA	522.0	551.4	550.6	0.25	−0.8
75%THA	502.8	530.7	528.9	0.69	0.2
50%THA	482.5	508.2	509.1	0.35	−2.5
40%THA	472.6	497.4	495.7	0.76	0.1
30%THA	460.7	484.6	483.2	0.66	−0.2

To further strengthen the credibility of the dynamic leakage simulation in the absence of field transient leakage data, a sensitivity and uncertainty analysis was conducted using the same dynamic model. The perturbed parameters and their ranges are listed in Table 4. The perturbation levels of $\pm 5\%$ and $\pm 10\%$ were selected to cover typical uncertainty in heat-transfer correlations, metal thermal inertia, inventory estimates, and initial shell-side boundary conditions. The leakage percentage and load condition were kept unchanged in each sensitivity run so that the influence of model-parameter uncertainty could be separated from the effect of leakage severity.

Table 4. Parameters included in the sensitivity and uncertainty analysis.

Purpose of Perturbation	Model Location	Perturbation	Parameter
Condensation-side heat-transfer uncertainty	Equation (13)	$\pm 5\%, \pm 10\%$	Shell-side heat-transfer coefficient,
Tube-side convection uncertainty	Equation (14)	$\pm 5\%, \pm 10\%$	Tube-side heat-transfer coefficient
Tube-side thermal storage uncertainty	Equation (19)	$\pm 5\%, \pm 10\%$	Feedwater inventory
Wall thermal inertia uncertainty	Equation (17)	$\pm 5\%, \pm 10\%$	Metal thermal capacity,
Extraction-pressure boundary uncertainty	Equation (22)	$\pm 5\%, \pm 10\%$	Initial shell pressure
Initial water-level/inventory uncertainty	Equation (23)	$\pm 5\%, \pm 10\%$	Initial drain volume

The sensitivity results show that the qualitative dynamic behavior remains unchanged under the tested perturbations: feedwater outlet temperature first rises slightly and then decreases to a lower steady level, while shell-side drainage volume increases after leakage. For the representative cases summarized in Table 5, the steady-state outlet-temperature and drain-volume metrics remain within about 2% of the baseline values, and the main variation appears in response time rather than in the direction of the leakage signal. This supports using the model for trend-level leakage-characteristic analysis under the investigated operating range.

Table 5. Summary metrics of representative sensitivity-response envelopes.

Maximum Change	$\pm 10\%$ Envelope	$\pm 5\%$ Envelope	Baseline	Output Metric	Representative Case
1.7%	−4.52 to −4.68 K	−4.55 to −4.65 K	−4.60 K	Final Δt_{w2}^*	THA, 5% leakage
10.0%	36–44 s	38–42 s	40 s	Time to first peak	THA, 5% leakage
9.6%	118–143 s	123–137 s	130 s	Time to 95% steady value	THA, 5% leakage
1.9%	−8.24 to −8.55 K	−8.31 to −8.49 K	−8.40 K	Final Δt_{w2}^*	THA, 10% leakage
1.9%	0.082–0.086 m ³ /s	0.083–0.085 m ³ /s	0.084 m ³ /s	Drain-volume growth rate	50% THA, 10% leakage
4.1%	211–229 s	214–226 s	220 s	Time to volume limit	50% THA, 10% leakage

* Final Δt_{w2} : It is defined as the difference between the final steady-state feedwater outlet temperature after leakage and the corresponding pre-leakage baseline value at the same load, i.e., $\Delta t_{w2} = t_{w2, \text{final}} - t_{w2,0}$. Therefore, a negative value indicates a decrease in feedwater outlet temperature after leakage.

Figure 2 shows representative sensitivity-response envelopes for feedwater outlet temperature and shell-side drainage volume. Although the perturbed parameters slightly change the response amplitudes, the main dynamic trends remain unchanged: the feedwater outlet temperature decreases after leakage, while the shell-side drainage volume increases. This confirms that the main leakage-response characteristics discussed in this study are robust within the tested parameter-perturbation range. Because no directly comparable plant transient data for induced leakage or non-leak load-change tests were available. . . the model is used to support trend-level diagnosis rather than exact reconstruction of a specific plant transient.

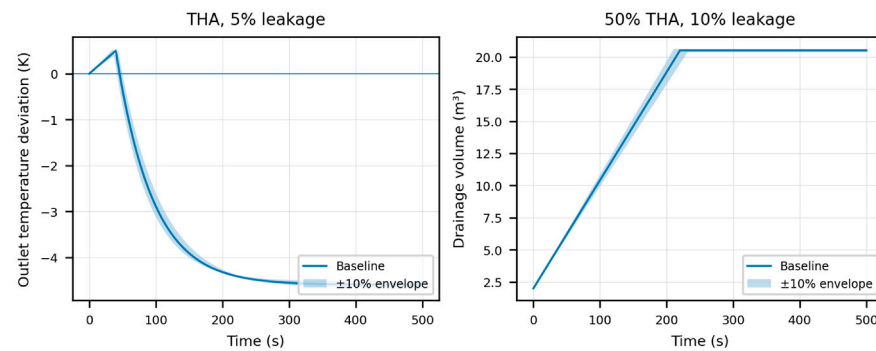


Figure 2. Representative sensitivity response envelopes for feedwater outlet temperature and shell-side drainage volume.

4. Results and Discussion

4.1. Effect of High-Pressure Heater Leakage on Feedwater Outlet Temperature Under Wide-Range Load Conditions

Based on the design parameters of the high-pressure heater, the initial THA parameters are: feedwater inlet temperature 522 K, inlet pressure 21.7 MPa, feedwater flow rate 547.7 kg/s, extraction steam temperature 660.4 K, and extraction steam pressure 6.11 MPa. The leakage flow rate is defined as a percentage of feedwater flow. Figure 3 shows the transient response of feedwater outlet temperature for different leakage rates.

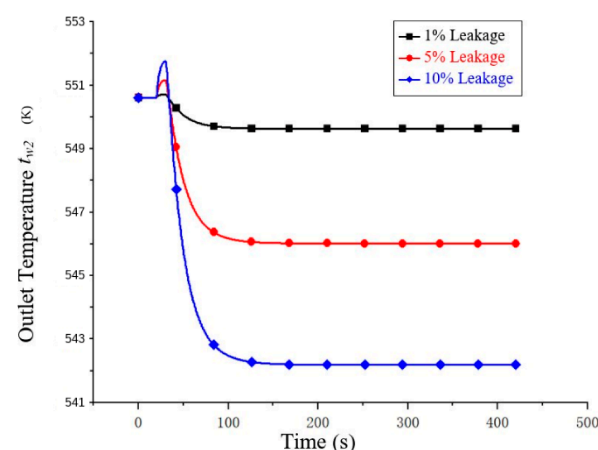


Figure 3. Feedwater outlet temperature at THA load.

Similarly, Figure 4 presents the transient response of feedwater outlet temperature under several typical partial-load conditions (from 75% to 30% THA).

Several key operating parameters of the HP heater's shell- and tube-sides for the 75% to 30% THA load range are listed in Table 6.

Table 6. Operating parameters of a high-pressure heater under off-design conditions.

Load Condition	Feedwater Inlet Temperature	Feedwater Inlet Pressure	Feedwater Flow Rate	Extraction Steam Temperature	Extraction Steam Pressure
Unit	K	MPa	kg/s	K	MPa
75% THA	502.8	18.73	376.68	643.6	4.402
50% THA	482.5	14.11	248.38	650.1	2.985
40% THA	472.6	12.2	202.15	552.27	2.446
30% THA	460.7	10.34	156.9	654.29	1.908

Figures 3 and 4 show the same overall behavior across leakage severities and loads: after leakage starts, feedwater outlet temperature rises slightly, then falls gradually, and finally stabilizes at a lower level. Under THA condition, the final drop of feedwater outlet temperature reaches about 1 K, 4.6 K and 8.4 K for 1%, 5% and 10% leakage respectively, and the final temperature drop shows a positive linear correlation with leakage severity at all partial loads.

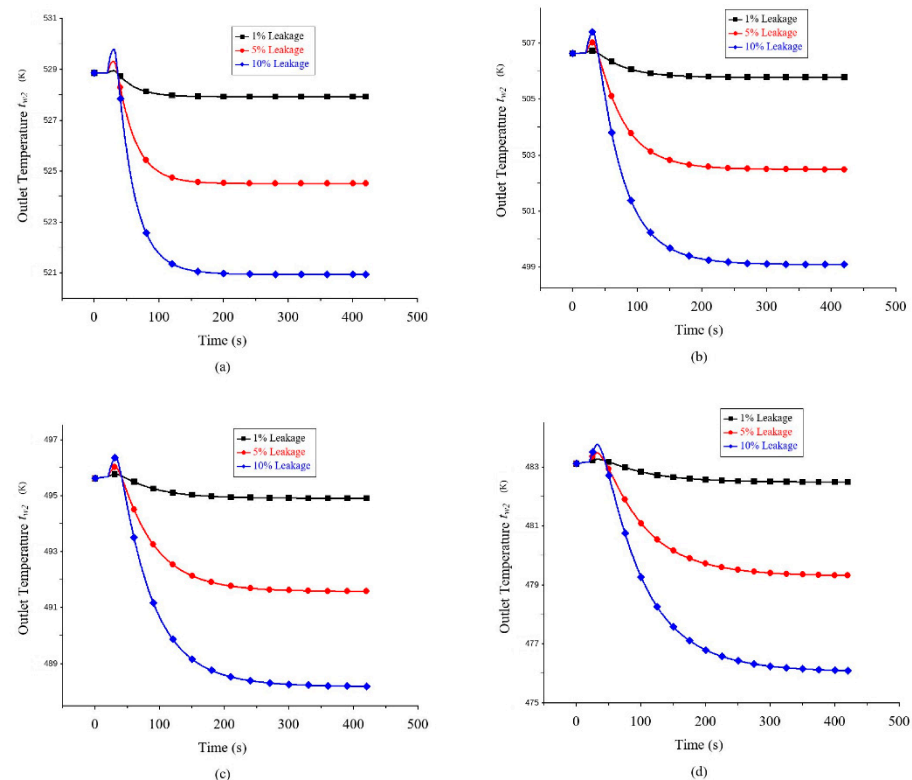


Figure 4. Feedwater outlet temperature under partial-load conditions. (a) 75% THA; (b) 50% THA; (c) 40% THA; (d) 30% THA.

This initial rise is interpreted as a short transient imbalance between heat input and tube-side mass flow based on the first law of thermodynamics and mass conservation: once leakage begins, part of the feedwater bypasses to the shell side, so tube-side flow drops abruptly while shell-side heat transfer from extraction steam remains unchanged in the short term. In this case, the heat absorbed per unit mass of feedwater in the tube side increases significantly, leading to a transient rise in feedwater outlet temperature; in Figure 2 (THA), the rise is negligible at 1% leakage but clearer at 5% and 10% leakage (about 0.5 K and 1.2 K). As leakage continues, shell-side drain accumulation raises the water level due to mass imbalance. Once part of the tube bundle is submerged, the effective condensation heat transfer area of the shell side is reduced, and the heat transfer driving force between high-temperature extraction steam and feedwater is weakened, which leads to a decline in the overall heat transfer efficiency of the high-pressure heater. Therefore, outlet temperature decreases with lower heat transfer efficiency, and larger leakage severity corresponds to a larger final temperature decrease due to a greater reduction in effective heat transfer area and more severe mass/energy imbalance.

As leakage continues, shell-side drain accumulation raises the water level. While the level is still below the bottom row of the U-tube bundle, the outlet temperature remains above the normal value. Once part of the tube bundle is submerged, the effective condensation heat-transfer area decreases. In addition, the submerged section tends to exchange heat with saturated water/drain instead of condensing steam, which weakens

heat-transfer effectiveness. Therefore, outlet temperature decreases and eventually reaches a new steady state lower than the initial state. Larger leakage severity corresponds to a larger final temperature decrease.

4.2. Effect of Varying Load Conditions on Outlet Temperature for the Same Leakage Severity

This section compares feedwater outlet temperature under 5% and 10% leakage severities, as shown in Figures 5 and 6. In each figure, curves 1 to 5 correspond to THA, 75% THA, 50% THA, 40% THA, and 30% THA, respectively.

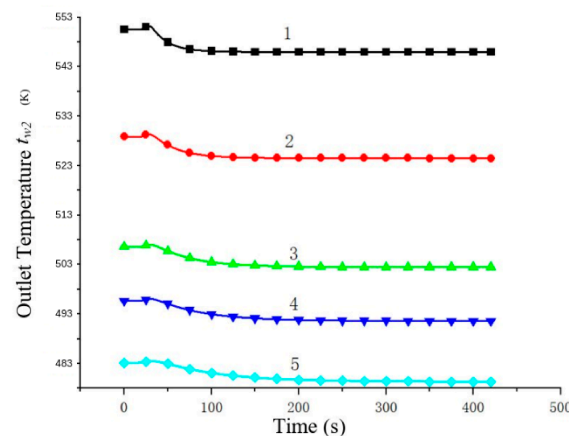


Figure 5. Feedwater outlet temperature comparison at 5% leakage under different loads.

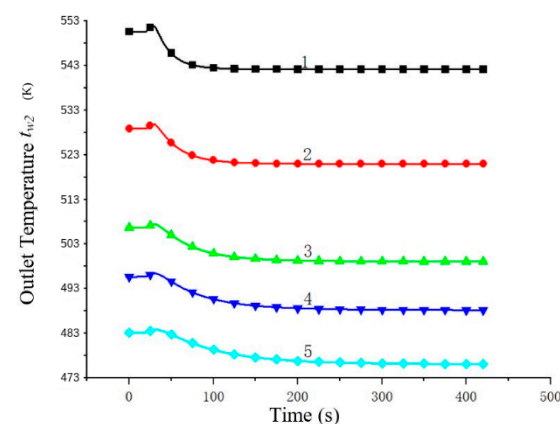


Figure 6. Feedwater outlet temperature comparison at 10% leakage under different loads.

Figures 5 and 6 indicate that, at the same leakage severity, higher load generally corresponds to a larger simulated decrease in feedwater outlet temperature from the normal value. This means small leakage signals are harder to distinguish at lower loads. Two factors are likely involved: the smaller temperature difference between extraction steam and feedwater weakens the heat-transfer driving force, and the relative influence of thermal inertia becomes stronger, slowing the response. In Figure 5, for 10% leakage, the final temperature drop is about 8.4 K at THA, compared with about 7.3 K at 40% THA and 7.2 K at 30% THA. In plant operation, these smaller low-load deviations can be close to normal fluctuation and sensor noise levels.

For the same leakage severity, the final feedwater outlet temperature drop decreases with the reduction in unit load, e.g., the 10% leakage leads to an 8.4 K drop at THA, while only about 7.3 K and 7.2 K drops at 40% THA and 30% THA respectively.

4.3. Effect of Different Leakage Severity on Metal Tube Wall Temperature

The following analysis compares the tube wall temperature profiles when leakage occurs under different operating load conditions: the full THA and the 50% THA, as illustrated in Figures 7 and 8.

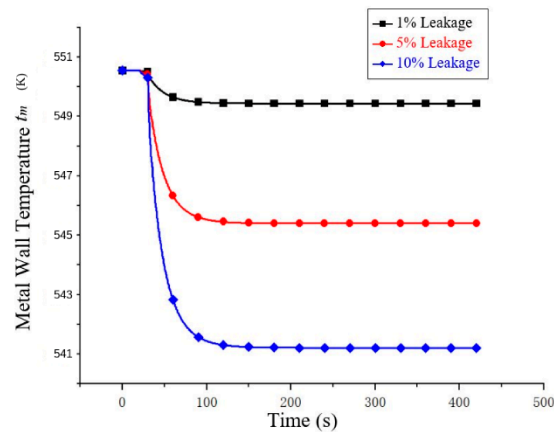


Figure 7. Metal tube wall temperature at THA load.

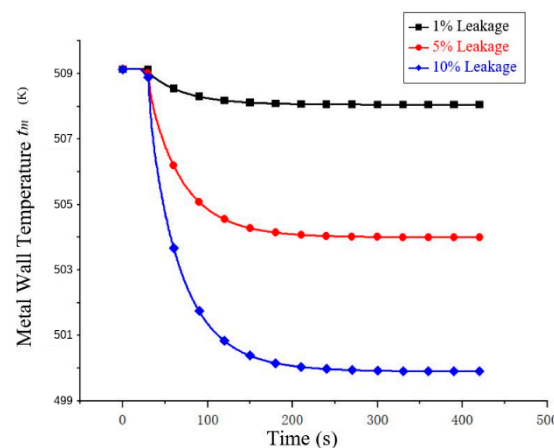


Figure 8. Metal tube wall temperature at 50% THA load.

Figures 7 and 8 show that the temporal profile of metal tube wall temperature follows the same direction as feedwater outlet temperature. At a fixed load, larger leakage severity leads to a larger decrease in wall temperature relative to normal operation. This trend suggests a potential increase in temperature-gradient-driven stress risk, which requires dedicated thermo-mechanical analysis for confirmation.

Coal-fired power plants often operate under frequent load variation. Over long-term operation, this condition may introduce additional alternating thermal stress and can aggravate leakage risk [12].

4.4. Effect of Different Leakage Severity on the Drainage Volume

This section analyzes the effect of leakage severity on the drainage volume on the shell-side of the high-pressure heater. Figures 9 and 10 present the change in drainage volume under THA and 50% THA conditions, respectively.

As shown in the figures, the drainage volume on the shell side is directly proportional to the leakage severity when the leakage occurs in the high-pressure heater. The more severe the leak, the more rapidly the drainage volume increases. For example, Figure 10 presents that when a 10% feedwater leakage occurs, it takes approximately 220 s for the

accumulating drains to reach the preset physical limit of shell-side volume. Because the opening of the drain regulating valve is kept constant, the drainage volume on the shell side increases at a steady linear rate over time. For the same leakage severity, the higher the unit load, the more rapid the increase in the drainage volume.

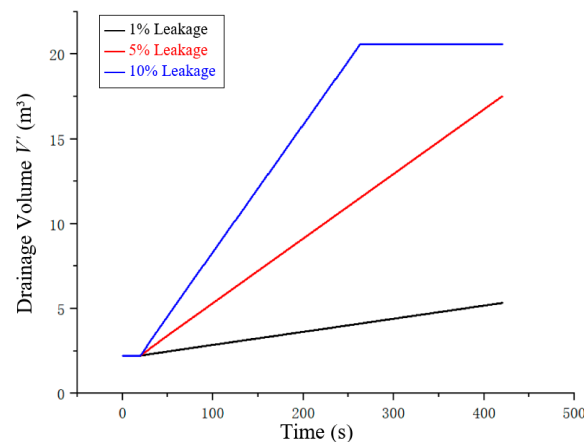


Figure 9. Drainage volume at THA load.

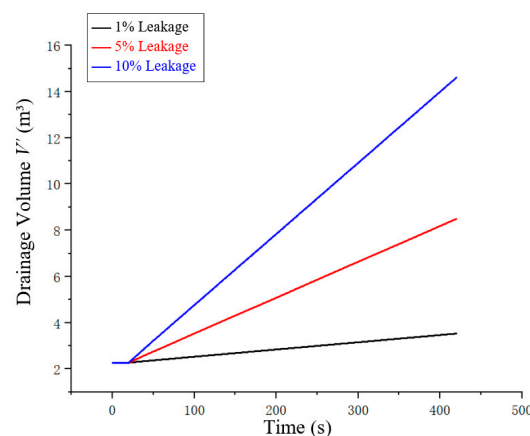


Figure 10. Drainage volume at 50% THA load.

5. Conclusions and Limitations

This paper developed static and dynamic simulation models for the No. 1 high-pressure heater. Based on leakage-fault simulations under wide-load conditions, the response characteristics for different leakage severities were analyzed. The main conclusions are as follows:

- (1) A three-section simulation model of the high-pressure heater was established. Under typical load conditions, the relative error of feedwater outlet temperature remains below 1%. Within this evidence range, the model is suitable for trend analysis of thermal-parameter responses under different leakage severities.
- (2) Under wide-load conditions, feedwater outlet temperature, metal tube wall temperature, and shell-side drainage volume all show sensitivity to leakage severity.
- (3) When leakage occurs in the condensing section, feedwater outlet temperature tends to decrease while drainage volume increases; larger leakage severity generally leads to larger deviation from normal operating values.

Limitations: This study relies on a mechanism-based simulation model. Due to the safety risk of inducing leakage faults in operating units, dynamic field data for direct leakage validation were unavailable in this work. Although the model is based on conservation

laws, checked against static design data, and further evaluated through sensitivity and uncertainty propagation analysis, the numerical results should be interpreted as trend-level reference values rather than universal alarm thresholds. In engineering application, the proposed response characteristics should be combined with plant-specific sensor accuracy, normal operating variability, water-level control logic and historical operation data to set practical leakage alarms and diagnostic limits.

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