Dynamic modeling and solution algorithm of the evaporation system for the ultra-supercritical power plant

Liang Xu^{*}, Jingqi Yuan, Jingcheng Wang

*Department of Automation, Shanghai Jiao Tong University, and the Key Laboratory of
System Control and Information Processing, Ministry of Education of China
800 Dongchuan Road, 200240 Shanghai, China*

^{*}Corresponding author. Tel. & Fax: +86 21 34204055.

E-mail addresses: xuliang34@163.com (L. Xu), jqyuan@sjtu.edu.cn (J. Yuan),
jcwang@sjtu.edu.cn (J. Wang)

Abstract: The evaporation system of the ultra-supercritical power plant is characterized with a once-through steam generator. Modeling of such evaporation system is of extreme importance for the purpose of efficiency assessment and operation optimization. There still exist some major problems in the modeling because of the complexity of the phase transitions in the evaporation system at different loads. This paper establishes a moving segment model (including subcooled water, saturated two-phase and superheated steam segments) based on the mass and energy conservation principles under the subcritical and supercritical working conditions, and proposes a solution algorithm to deal with working condition switching in an automatic mode. Pseudo-online simulations are performed to test the model and the algorithm by using the geometric parameters and the history operation data of a real ultra-supercritical unit in China. The calculated heat absorption rate of the working fluid from the entire evaporation system agrees well with the reference value. The maximum absolute relative error is found as small as 0.54% in the time resolution of 30 min, which indirectly demonstrates the feasibility and accuracy of the presented model and the solution algorithm.

Keywords: evaporation system, ultra-supercritical unit, once-through steam generator, moving segment model, solution algorithm

1. Introduction

Due to higher thermal efficiency, ultra-supercritical coal-fired power plants have attracted more and more attention and become a prior choice of newly built units [1]. According to the latest statistics in China only, 58 ultra-supercritical coal-fired power plants of 600 MW and 78 of 1000 MW are in service, and 70 of 600 MW and 74 of 1000 MW are under construction. In 2020, the capability share of the ultra-supercritical power plants in China is expected to be more than 30% of 1885 GW [2]. In recent years, ultra-supercritical units obtain intensive investigations [3-7] including facilities design, control strategy, performance assessment, etc.

Compared to the subcritical unit, the once-through evaporation system of the ultra-supercritical unit is distinctly different, which may operate under the supercritical or subcritical working conditions. Under supercritical working condition, the working fluid becomes superheated steam from subcooled water without the saturated two-phase state. In order to obtain the real dynamics of the evaporation system and support the efficiency assessment and operation optimization of the boiler-side whole process, it is necessary to model the evaporation system applying to any

working condition.

The data-driven and the first-principle modelings are two basic approaches for once-through evaporation system modeling. The data-driven modeling principally regards the evaporation system as a black box to be identified with the reliable input and output data and is often used for fast controller design [8, 9]. In contrast, the first-principle modeling is more accurate in the expression of the internal dynamic characteristics [10] and more useful for dynamic simulation and performance analysis.

The evaporation system of the ultra-supercritical power plant is a typical distributed parameter system [11-13]. In literature, the evaporation system may be spatially discretized into finite control volumes. For each control volume, the mass, energy, and momentum conservation equations of the working fluid and the heat transfer equation between the flue gas and the working fluid are constructed, respectively. Based on the given boundary conditions, the dynamic characteristics of the evaporation system can be obtained. However, the computation time of the distributed parameter model relies to a large extent on the number of the control volumes and the simulation software performance (for example, APROS, gPROMS, Fluent) [14-16], which is usually not suitable for online application.

In light of the extreme complexity of distributed parameter model of the evaporation system, lumped parameter models are investigated in literature, which have two elementary forms: the total and the local lumped models. The total lumped models treat the entire evaporation system as a two-phase mixture with an assumed load-associated vapor-liquid fraction [17-21]. The local lumped models divide the flow channel of the once-through evaporation system into two (i.e. subcooled water, superheated steam), three (i.e. subcooled water, saturated two-phase and superheated steam) or four (i.e. subcooled water, nucleate boiling, film boiling and superheated steam) segments according to the states of the working fluid, and build up lumped sub-models for each segment [22-25], which is the compromise of system complexity and accuracy. Nevertheless, some of these reported models do not cover both the supercritical and the subcritical working conditions and others need to prescribe the working conditions of the evaporation system in advance.

In this contribution, a moving segment model aiming at obtaining the internal dynamic characteristics of the evaporation system is established, which belongs to the local lumped models. Differently, the proposed moving segment model is oriented to the online operation optimization based on the distributed control system (DCS) data, rather than the pure process simulation. Additionally, the proposed moving segment model can automatically deal with the switching between the supercritical and the subcritical working conditions through the proposed solution algorithm, which is very necessary for the online application. The solution algorithm is characterized with transforming the model online solving into three minimization problems with different objective functions and feasible regions. By the use of the realtime data from the DCS onsite, the proposed model and solution algorithm can identify the real working condition and help to obtain the internal dynamics of the evaporation system.

2. Description of the evaporation system of the ultra-supercritical power plant

Different from the subcritical unit, the ultra-supercritical boiler has a once-through steam generator without a drum. Fig. 1 depicts the front view of the evaporation system of a real 1000 MW ultra-supercritical power plant in China, which is used as the investigated unit in this study. It

usually consists of the helical water wall made of the rifled tubes and the vertical water wall made of the smooth tubes. L_0 ($L_0=12.07$ m) is the side length of the bottom of the helical water wall. H_1 ($H_1=33.50$ m) is the height of the upper of the helical water wall. θ_0 ($\theta_0=23.58^\circ$) is the lift angle of the helical water wall. The vertical water wall is in series with helical water wall via the headers. H_2 ($H_2=25.13$ m) is the height of the vertical water wall. At the rear of the vertical water wall, a few tubes are expanded to be the slag-cooling tubes, which are subsequently led to the steam-water separator together with the remaining vertical tubes.

When the unit works at 30% full load or higher, the subcooled water leaving the economizer (at about 30 MPa, 602 K) is led into the water wall and then heated into the superheated steam (at about 28.5 MPa, 683 K) in the once-through mode. The superheated steam leaving the evaporation system is further heated in the superheaters and fed into the steam turbine.

Fig. 1. Front view of the evaporation system of a 1000 MW ultra-supercritical power plant. A few irregular short tubes and headers are not displayed.

3. Model of the evaporation system for the ultra-supercritical power plant

3.1 Geometric equivalence and segmentation

The parallel tubes of the evaporation system for the investigated unit are classified into five groups according to their geometric parameters, see Table 1, which includes the number n_i ($i=1, \dots, 5$), the outer diameter d_i , and the wall thickness δ_i of the tubes in each group.

Table 1. Geometric parameters of the evaporation system tubes of the investigated unit.

I	Description of the tubes	Number n_i	Outer diameter d_i (mm)	Thickness δ_i (mm)
1	helical water wall	778	38.1	7.5
2	front of the vertical water wall	536	31.8	7.5
3	sides of the vertical water wall	2×244	31.8	7.5
4	rear of the vertical water wall	468	31.8	6.4
5	slag-cooling tubes	66	76.2	20

Along with the flow direction of the working fluid, all the parallel tubes in each group are

equivalent to a single tube for simplification based on the invariant total volume and cross-sectional area. Additionally, the thermodynamic properties of the working fluid at the same cross-section of the tube are considered as uniform. For the helical water wall, the equivalent length L_1 (m) and the cross-sectional area S_1 (m²) becomes:

$$L_1 = \frac{H_1 + L_0}{\sin \theta_0} = 113.94 \quad (1)$$

$$S_1 = n_1 \cdot \frac{\pi(d_1 - 2\delta_1)^2}{4} = 0.3259 \quad (2)$$

As to the vertical water wall, the equivalent length L_2 (m) and the cross-sectional area S_2 (m²) are:

$$L_2 = H_2 = 25.123 \quad (3)$$

$$S_2 = \sum_{i=2}^5 n_i \cdot \frac{\pi(d_i - 2\delta_i)^2}{4} = 0.4274 \quad (4)$$

In this way, the tubes of the evaporation system are equivalent to a single tube with two different cross-sectional areas, as shown in Fig. 2.

Fig. 2. Equivalent structure of the evaporation system. 0,1,2,3: section index; L_S , L_E , L_W : the length of the superheated steam, two-phase, subcooled water segments, respectively; Q_S , Q_E , Q_W : the heat absorption rate of the working fluid in the superheated steam, two-phase, subcooled water segments, respectively; S_1 , S_2 : the inlet and outlet section area of the equivalent tube; A_1 , A_2 : the area of the interface 1 and 2, respectively.

Fig.2 illustrates the equivalent structure of the evaporation system. The evaporation system generally behaves in two potential working conditions, i.e. the subcritical and the supercritical working conditions, during the normal operation. Under the subcritical working condition, the working fluid in the evaporation system has three states: subcooled water, saturated liquid-vapor two-phase, and superheated steam. However, the working fluid under the supercritical working condition directly becomes superheated steam from the subcooled water without the saturated two-phase state.

According to the state of the working fluid under the subcritical working condition, the entire evaporation system is divided into three segments: the subcooled water segment, the liquid-vapor two-phase segment, and the superheated steam segment, whose lengths are denoted as L_W , L_E , and L_S , respectively. The sections 0, 1, 2, and 3 are the inlet of the evaporation system, the interface between the subcooled water and the saturated liquid-vapor two-phase, the interface between the

two-phase and the superheated steam, and the outlet of the evaporation system, respectively.

Obviously, the sections 1 and 2 may move up and down at different unit loads, and their areas (A_1 and A_2) may switch between S_1 and S_2 . Also, the movements of the sections 1 and 2 lead to the changes of L_W , L_E , and L_S . Under the supercritical working condition, the sections 1 and 2 are merged into one (but still denoted as section 1), and L_E is equal to zero.

3.2 Dynamic modeling

Fig. 3 demonstrates the input and output variables of the moving segment model. The input variables are online acquired via the DCS, while the geometric parameters of the evaporation system are set as the constants. The outputs are some concerned state variables for operation optimization.

Fig. 3. The input and output variables of the moving segment model of the evaporation system.
 p : pressure; T : temperature; L : length; D : mass flow rate; Q : heat absorption rate of the working fluid; m : mass storage; H : enthalpy storage; subscripts 0, 1, 2, 3: section 0, 1, 2, 3; subscripts W , E , S : subcooled water, two-phase, superheated steam segments, respectively; subscript sum: summation.

3.2.1 Moving segment model

For different working conditions, the sections 1 and 2 are respectively defined as:

(1) Under the subcritical working condition, the working fluid on the section 1 and 2 is the saturated water and the saturated vapor, respectively. Their thermodynamic parameters are determined uniquely by the pressure or temperature according to the IAPWS-IF97 [26].

(2) Under the supercritical working condition, the working fluid on the section 1 (the sections 1 and 2 are merged into one) is defined as the quasi-critical state. The quasi-critical state refers to the state where the working fluid has the maximum specific heat at a specified pressure [27]. Nine quasi-critical points covering the pressure range of 22.064 ~ 30 MPa are obtained by the IAPWS-IF97. Furthermore, a quasi-critical curve is regressed in the form of quadric polynomial with the least square method (LSM):

$$T = -0.0411 \times p^2 + 5.6719 \times p + 541.96 \quad (5)$$

Fig. 4. Regression of the quasi-critical curve

Due to the complexity of the resistance characteristic and the momentum equation of the evaporation system tubes, the accurate damping coefficients of the tubes cannot be obtain [11, 14, 27]. Therefore, on account of the pressure drop of the working fluid in the evaporation system is small enough compared to its absolute pressure, the pressure distribution of the working fluid is treated as quadric distributed based on the fact that the pressure of the working fluid with the smaller density drops faster:

$$p_0 - p_x = (p_0 - p_3) \left(\frac{L_x}{L} \right)^2 \quad (6)$$

where p represents the pressure of the working fluid; the subscripts 0 and 3 indicate the sections 0 and 3; the subscript x is a section which has a distance L_x from the section 0; L is the total length of the equivalent tube of the evaporation system:

$$L = L_E + L_W + L_S \quad (7)$$

Thus, the pressures p_1 and p_2 of the working fluid on the sections 1 and 2 are calculated by Eq. (6). Furthermore, the temperatures T_1 and T_2 of the working fluid on the section 1 and 2 are uniquely determined by the pressure p_1 and p_2 according to the IAPWS-IF97 for the subcritical working condition or Eq. (5) for the supercritical condition.

(I) The subcritical case

Under the subcritical working condition, for the subcooled water, the mass and energy conservation equations give:

$$\frac{\partial(\rho_w V_w)}{\partial t} = D_0 - (D_1 - A_1 \rho_1 \frac{dL_w}{dt}) \quad (8)$$

$$\frac{\partial(\rho_w V_w h_w)}{\partial t} = D_0 h_0 - (D_1 - A_1 \rho_1 \frac{dL_w}{dt}) h_1 + Q_w \quad (9)$$

where, h , ρ , V , D , and Q are the specific enthalpy, density, volume, mass flow rate, and heat absorption rate of the working fluid, respectively; the subscript W refers to the subcooled water segment; the subscripts 0 and 1 represent the sections 0 and 1; A_1 and A_2 are:

$$A_1 = \begin{cases} S_1, & L_w < L_1 \\ S_2, & L_w \geq L_1 \end{cases}, \quad A_2 = \begin{cases} S_1, & L_s \geq L_2 \\ S_2, & L_s < L_2 \end{cases} \quad (10)$$

For the saturated liquid-vapor two-phase, the mass and energy conservation equations are:

$$\frac{\partial(\rho_E V_E)}{\partial t} = (D_1 - A_1 \rho_1 \frac{dL_W}{dt}) - (D_2 - A_2 \rho_2 \frac{d(L_W + L_E)}{dt}) \quad (11)$$

$$\frac{\partial(\rho_E V_E h_E)}{\partial t} = (D_1 - A_1 \rho_1 \frac{dL_W}{dt}) h_1 - (D_2 - A_2 \rho_2 \frac{d(L_W + L_E)}{dt}) h_2 + Q_E \quad (12)$$

where the subscript E and 2 refer to the two-phase segment and the section 2, respectively.

For the superheated steam, the mass and energy conservation equations are as follows:

$$\frac{\partial(V_S \rho_S)}{\partial t} = D_2 + A_2 \rho_2 \frac{dL_S}{dt} - D_3 \quad (13)$$

$$\frac{\partial(V_S \rho_S h_S)}{\partial t} = (D_2 + A_2 \rho_2 \frac{dL_S}{dt}) h_2 - D_3 h_3 + Q_S \quad (14)$$

where the subscript S and 3 refer to the superheated segment and the section 3.

The lumped parameters, i.e. the specific enthalpies h_W , h_E , h_S , and the densities ρ_W , ρ_E , ρ_S of the working fluid in the state of subcooled water, liquid-vapor two-phase, and superheated steam, are calculated with the IAPWS-IF97 based on the lumped arithmetic average pressure p_W , p_E , p_S and temperature T_W , T_E , T_S , respectively. The lumped arithmetic average p_W , p_E , p_S , T_W , T_E , and T_S are expressed as:

$$T_W = \frac{T_0 + T_1}{2}, p_W = \frac{p_0 + p_1}{2} \quad (15)$$

$$T_E = \frac{T_1 + T_2}{2}, p_E = \frac{p_1 + p_2}{2} \quad (16)$$

$$T_S = \frac{T_2 + T_3}{2}, p_S = \frac{p_2 + p_3}{2} \quad (17)$$

Specifically, it is assumed that the lumped two-phase working fluid has a liquid phase fraction of 0.5. Combining Eqs. (8), (11) and (13), the mass storage of the working fluid in the entire evaporation system is:

$$\frac{\partial(\rho_W V_W + \rho_E V_E + \rho_S V_S)}{\partial t} = D_0 - D_3 \quad (18)$$

Then, for the sake of simplification, the mass storage is spread into two equal parts with the parameter α ($\alpha \in [0, 1]$):

$$\frac{\partial(\rho_W V_W + \alpha \cdot \rho_E V_E)}{\partial t} = 0.5 \cdot (D_0 - D_3) \quad (19)$$

where α is adjusted by the accuracy requirement of the model.

(II) The supercritical case

As mentioned above, under the supercritical working condition, the sections 1 and 2 are merged into the section 1 and $L_E = 0$. For the subcooled water, the mass and energy conservation equations are the same as Eqs. (8) and (9). For the superheated steam, the mass and energy conservation equations become:

$$\frac{\partial(V_S \rho_S)}{\partial t} = D_1 + A_1 \rho_1 \frac{dL_S}{dt} - D_3 \quad (20)$$

$$\frac{\partial(V_S \rho_S h_S)}{\partial t} = (D_1 + A_1 \rho_1 \frac{dL_S}{dt}) h_1 - D_3 h_3 + Q_S \quad (21)$$

Moreover, the lumped arithmetic average p_W , p_S , T_W , and T_S are expressed as follows:

$$T_W = \frac{T_0 + T_1}{2}, p_W = \frac{p_0 + p_1}{2} \quad (24)$$

$$T_s = \frac{T_1 + T_3}{2}, p_s = \frac{p_1 + p_3}{2} \quad (25)$$

Combining Eq. (8) and (20), the mass storage of the working fluid in the entire evaporation system is:

$$\frac{\partial(\rho_w V_w + \rho_s V_s)}{\partial t} = D_0 - D_3 \quad (24)$$

3.3.2 Calculation of the mass flow rates D_0 and D_3 of the working fluid

Generally, the mass flow rate D_{fw} of the feed water is measured by the DCS in real time, and the mass flow rate D_{ms} of the main steam is calculated with the steam pressure p_{fs} at the first stage (governing stage) of the turbine and the temperature T_{ms} of the main steam according to the Flügel formula [28], which is expressed as:

$$D_{ms} = \frac{D_{ms_ref} \sqrt{T_{ms_ref}}}{p_{fs_ref}} \frac{p_{fs}}{\sqrt{T_{ms}}} \quad (27)$$

where D_{ms_ref} , p_{fs_ref} , and T_{ms_ref} are the mass flow rate of the main steam, the steam pressure at the first stage of the turbine and the temperature of the main steam under the reference working condition of the turbine, respectively.

The spray water of the first and second stage attemperators coming from the feed water leads to the superheaters. Hence, the mass flow rates D_0 and D_3 of the working fluid yield:

$$D_0 = D_{fw} - D_{at1} - D_{at2} \quad (28)$$

$$D_3 = D_{ms} - D_{at1} - D_{at2} \quad (29)$$

where D_{at1} and D_{at2} are the mass flow rates of the spray water of the first and second stage attemperators, respectively.

3.3 Solution algorithm

A generic algorithm to solve the evaporation system model, which automatically responds the switching between the subcritical and the supercritical working conditions, is developed. The algorithm is characterized with converting the model solving process into optimization problems.

The temperature, pressure and mass flow rate of the working fluid at the inlet and the outlet of the evaporation system are online provided by the DCS. Once the lengths L_W , L_E , and L_S of the three moving segments are determined, the other output variables shown in Fig. 3 can be calculated by substituting the lengths back into the model of the evaporation system. Hence, L_W , L_E , and L_S are the key intermediate variables of the model. Furthermore, L_W and L_S are used as the two independent variables.

In the subcritical case, L_W and L_S are determined by Eqs. (7), (18) and (19), while in the supercritical case, L_W and L_S are determined by Eqs. (7) and (24). The feasible region, denoted as Γ , covers two dimensions of L_W and L_S , and changes with the working conditions of the evaporation system.

The model solving is now converted into three minimization problems with L_W and L_S as the independent variables and Γ as the feasible region, depending on p_0 and p_3 :

- (1) When $p_0 \leq 22.064$, meaning the pressure of the working fluid in the evaporation system is

always less than the critical pressure (22.064 MPa), the two-phase segment exists definitely and the evaporation system works under the subcritical working condition.

The objective function is formulated as:

$$J = \min_{L_w, L_s} \left\{ \sqrt{(\Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \rho_e V_e + \rho_s V_s))^2 + (0.5 \cdot \Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \alpha \rho_e V_e))^2} \right\} \quad (30)$$

and the feasible region Γ is:

$$\Gamma = \begin{cases} L_w + L_s < L \\ L_w, L_s \geq \varepsilon \end{cases} \quad (31)$$

where Δt is the sampling period; ε the tolerant minimum length of the subcooled water and superheated steam segments. Approximately, Γ is a triangle region.

(2) When $p_3 \geq 22.064$, representing the pressure of the working fluid in the evaporation system is always greater than the critical pressure, the two-phase segment does not exist definitely and the evaporation system works under the supercritical working condition.

The objective function becomes:

$$J = \min_{L_w, L_s} |\Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \rho_s V_s)| \quad (32)$$

and the feasible region Γ is:

$$\Gamma = \begin{cases} L_w + L_s = L \\ L_w, L_s \geq \varepsilon \end{cases} \quad (33)$$

Herein, Γ is a line segment.

(3) When $p_3 < 22.064 < p_0$, whether the two-phase segment exists need to be automatically discriminated by the solution algorithm. L_c is defined as the critical length of the subcooled water segment and satisfies:

$$p_0 - 22.064 = (p_0 - p_3) \left(\frac{L_c}{L} \right)^2 \quad (34)$$

When $\varepsilon \leq L_w < L_c$, p_1 exceeds the critical pressure, which means the two-phase segment does not exist. Hence, the feasible region Γ is:

$$\Gamma_1 = \begin{cases} L_w < L_c \\ L_w + L_s = L \\ L_w, L_s \geq \varepsilon \end{cases} \quad (35)$$

In contrast, when $L_w \geq L_c \geq \varepsilon$, p_1 is lower than the critical pressure and the two-phase segment exists. The feasible region Γ is:

$$\Gamma_2 = \begin{cases} L_w \geq L_c \\ L_w + L_s < L \\ L_w, L_s \geq \varepsilon \end{cases} \quad (36)$$

In this context, the minimization problem is defined as:

$$J = \min \left\{ \begin{array}{l} \min_{L_w, L_s} |\Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \rho_s V_s)|, \quad L_w < L_c \\ \min_{L_w, L_s} \left\{ \sqrt{(\Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \rho_e V_e + \rho_s V_s))^2} \right. \\ \left. + (0.5 \cdot \Delta t \cdot (D_0 - D_3) - \Delta(\rho_w V_w + \alpha \rho_e V_e))^2 \right\}, \quad L_w \geq L_c \end{array} \right\} \quad (37)$$

$$s.t. \quad \Gamma = \Gamma_1 \cup \Gamma_2 \quad (38)$$

The optimization problems are then solved by an online optimization method, such as inter-point method [29].

4. Results and discussion

The pseudo-online simulation based on the established model and the solution algorithm is performed on a personal computer (Inter Q6600 CPU and 4 GB RAM) with 24-hour data of a real 1000 MW ultra-supercritical unit in Shandong, China. The data sampling period is 15 s.

4.1 Pseudo-online simulation results

Fig. 5 summarizes the pseudo-online simulation results with 24-hour data. It is found that the evaporation system switches between the subcritical and supercritical working conditions, when the unit load changes in the range of 60% ~ 90% boiler rated load (BRL).

The upper of Fig. 5(a) plots the inlet and outlet pressure of the evaporation system. The pressure drop (0.2 ~ 1.0 MPa) in the evaporation system is approximately 1% ~ 3% of the absolute inlet pressure (19.0 ~ 28.5 MPa), which confirms the explanation in Section 3.2.1. Along with the fluctuations of the pressure p_1 and p_3 , the working conditions switching of the evaporation system is automatically detected by the solution algorithm. As shown in the lower part of Fig. 5(a), the two-phase segment emerges when the evaporation system works under the subcritical working condition, for example at 2nd to 5th hour.

The increase of the unit load means more heat to be released by the coal, leading to more evaporation of the working fluid. On the other hand, more fresh subcooled water is fed into the heat cycle system to meet the increasing load demand. The vapor-liquid ratio or the positions of the sections 1 and 2 between segments, depend on the collective effect of the feeding and the evaporation of the working fluid.

When the evaporation system shifts from the supercritical to the subcritical working condition, less working fluid becomes the superheated steam and more wet steam is generated in the once-through process, and vice versa. Hence, the length of the superheated steam segment behaves the opposite tendency to that of the two-phase segment.

By substituting the calculated lengths of the three segments back into the model, the heat absorption rate of the working fluid in each segment is obtained. In the upper of Fig. 5(b), Q_{sum} is the summation of the heat absorption rates of the working fluid in all segments. It is found that the total heat absorption rate Q_{sum} of the working fluid in the entire evaporation system accounts for only ca. 80% of the unit power P . It is quite different from the 300 MW subcritical unit in Ref. [30], where Q_{sum} and P are comparable.

Inside the once-through evaporation system, the pressure and the temperature of the working fluid change simultaneously with the working condition. It is well known, for both the subcooled

water and superheated steam, the density is negatively related to the temperature but positively to the pressure, and the specific enthalpy is positively related to temperature. However, between the specific enthalpy and the pressure, there is a positive correlation for the subcooled water but a negative correlation for the superheated steam. Therefore, the pressure and the temperature together determine the variant of the mass and enthalpy storage of the working fluid. Fig. 5(c) shows the mass and enthalpy storages of the working fluid in each segment as well as the entire evaporation system. It may be found that the total mass and total enthalpy storages in the evaporation system are quasi-constant, which shows the once-through boiler has bad heat storage capacity compared with the drum boiler. Additionally, the mass and enthalpy storages in the superheated steam segment are more sensitive to the working condition than in the subcooled water segment.

Fig. 5. Pseudo-online simulation results based on the established model and solution algorithm with 24-hour data. (a) Inlet/outlet pressure and length of the water/two-phase/steam segment; (b) Unit load and heat absorption rates of the working fluid from the evaporation system; (c) Mass and enthalpy storage in the evaporation system. p_0 , p_3 : pressure at the inlet and outlet of the evaporation system; P : Unit load; Q_{sum} : the total heat absorption rate of the working fluid from the evaporation system; Water: subcooled water segment; Two-phase: two-phase segment; Steam: superheated steam segment.

Table 2 gives the average computation time of the solution algorithm of the proposed moving segment model in each sample period. It is found that the average time (total 5760 computations) is 0.46 s and the solving of the minimization problem costs 0.44 s, which implies the moving segment model and its solution algorithm are quite suitable for the typical online application.

Table 2. Average computation time of the solution algorithm of the proposed moving segment model.

Terms	Average computation time (s)
Minimization problem solving (inter-point method)	0.44
Model back-substitution calculation	0.02

The simulation results reveal the internal physical characteristics of the evaporation system. The online calculated lengths and heat absorption rates of each segment help to analyze the dynamics of the evaporation system qualitatively under the variant working conditions and supply the data support quantificationally for the control model establishment. Meanwhile, they are also the necessity of the boiler-side whole process simulation and operation optimization.

4.2 Model validation

The segment model is established with the assumption on the physically definite interfaces between segments. On the other hand, the transients of the output variables, such as L_i , Q_i , m_i , and H_i ($i=W, E, S$) cannot be online measured. Therefore, the average heat absorption rate of the working fluid from the entire evaporation system in the time resolution of 30 min, denoted as

$\bar{Q}_{sum}$, is used as validation. $\bar{Q}_{sum}$ is calculated by:

$$\bar{Q}_{sum} = \frac{1}{1800} \int_0^{1800} (D_3(t) \cdot h_3(t) - D_0(t) \cdot h_0(t)) dt \quad (39)$$

The relative error e of the calculated heat absorption rate of the working fluid versus the reference value is defined as:

$$e = \frac{\frac{1}{1800} \int_0^{1800} Q_{sum}(t) dt - \bar{Q}_{sum}}{\bar{Q}_{sum}} \times 100\% \quad (40)$$

Fig. 6 compares the calculated dynamic heat absorption rate of the working fluid from the evaporation system and the 30-min average reference value. The calculated heat absorption rate is well in coincidence with the reference value, see Fig. 6(a). Fig. 6(b) demonstrates the distribution of the relative error e . The maximum absolute relative error is as small as 0.54%, and the mean absolute relative error is only 0.10%.

Fig. 6. Calculated heat absorption rate of the working fluid in the evaporation system versus the 30-min average reference value. (a) Comparison in time domain; (b) Relative error distribution.

5. Conclusions

The moving segment model of the evaporation system of the ultra-supercritical power plant is established based on dynamic mass and the energy balance. To realize automatic switching

between the subcritical and the supercritical working conditions, a solution algorithm is proposed, which converts the task of model solution into different optimization problems. The average heat absorption rate of working fluid from the evaporation system in the time resolution of 30 min is used as the reference to validate the proposed model and the solution algorithm. The maximum absolute relative error is found less than 1.0%, which shows the feasibility and the accuracy of the proposed model. The model is online applicable for both the subcritical and the supercritical operation conditions. It may be applied to the operation optimization in online mode and to estimate the heat absorption rate of the working fluid dynamically, which is potentially beneficial for the real-time identification of the lower heating value of the feed coal.

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Nomenclature

d	outer diameter (mm)
e	relative error
h	specific enthalpy (kJ/kg)
m	mass storage (kg)
n	number of tubes
p	pressure (MPa)
t	time (s)
A	interface area (m ²)
D	mass flow rate (kJ/s)
H_i	height (m, $i=1, 2$)
H	enthalpy storage (GJ)
L	length (m)
P	unit load (MW)
Q	heat absorption rate (MW)
S	section area (m ²)
T	Temperature (K)
V	volume (m ³)

Greek letters

δ	thickness of tubes (mm)
ρ	density (kg/m ³)
Δt	sampling period (s)

Subscripts

$at1, at2$	attenuator 1, 2
c	critical
fs	first stage
ms	main steam
ref	reference working condition
sum	summation
x	section
E	saturated two-phase segment
S	superheated steam segment
W	subcooled water segment
0,1,2,3	section index