Evaporator Heating with Optimum Fluid Temperature Changes

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Abstract

In order to speed up the boiler start-up, especially from the cold state, the boiler evaporator may be flooded with hot water at the beginning of the process. The temperature of water supplied to the boiler evaporator through lower chambers of the boiler furnace chamber waterwalls will be found from the condition of equality between allowable stresses and circumferential thermal stresses on the edge of the opening for the downcomer. Two mathematical models of the evaporator hot-water heating are presented in this paper – one with lumped and the other with distributed parameters. In both cases, the water level is identical. The hot water needed to fill the boiler evaporator may be supplied from the feed water tank of another operating boiler or from a hot water vessel.

Keywords: steam boiler, boiler evaporator, boiler start-up, thermal stresses, optimum heating, mathematical model of the evaporator

1. Introduction

Hot water tanks are often installed in combined heat and power plants to equalize the power unit load required to prepare hot water for the municipal heating network. Pressure heat accumulators have also been constructed lately to increase the plant power capacity under the peak load and decrease it in periods of lower demand for electricity during the night. Hot water from the pressure accumulators can be used for filling the evaporator in order to shorten the start-up of the boiler [1]. In previous works, optimum time temperature changes were determined for fluid in the boiler drum [5-8]. In this paper, optimum initial temperature of the hot water and time changes of water temperature after the drum is filled with hot water to the minimum level, will be determined from the condition of equality between allowable stresses and the total of circumferential stresses resulting from pressure and circumferential...
thermal stresses on the edge of the drum-downcomer junction. Since the water is supplied to the evaporator through lower headers, a mathematical model of the evaporator must be included while determining optimum fluid temperature changes. During the filling of the evaporator the hot water flows in parallel through the furnace waterwalls and the downcomer tubes. Boiler burners are turned off. Water gets cooler flowing from the lower headers to the drum. The air flows in the combustion chamber from the bottom up due to free convection. The waterwall tubes are cooled by air passing through the furnace chamber. A part of the heat is absorbed by the waterwall tube walls. In the case of the downcomer tubes, whose external surfaces are thermally insulated, the heat taken from water raises the tube metal temperature. For this reason, it is necessary to develop a mathematical model describing the cooling of water in both the waterwalls (riser tubes) and the downcomer tubes that will make it possible to determine the time-dependent history of the water temperature at the drum inlet for a prescribed time changes of the water temperature at the lower header inlet.

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbols</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a, b, c$</td>
<td>constants</td>
</tr>
<tr>
<td>$A$</td>
<td>area, m²</td>
</tr>
<tr>
<td>$b$</td>
<td>thickness of the longitudinal fin, m</td>
</tr>
<tr>
<td>$c$</td>
<td>specific heat, J/kgK</td>
</tr>
<tr>
<td>$d$</td>
<td>diameter, m</td>
</tr>
<tr>
<td>$D$</td>
<td>inner diameter of the drum, m</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient, W/(m²K)</td>
</tr>
<tr>
<td>$H$</td>
<td>wall thickness of the drum, m</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity, W/(mK)</td>
</tr>
<tr>
<td>$m$</td>
<td>mass, kg</td>
</tr>
<tr>
<td>$N$</td>
<td>number of control volumes</td>
</tr>
<tr>
<td>$N_c$, $N_p$</td>
<td>number of transfer units for water and air, respectively</td>
</tr>
<tr>
<td>$p$</td>
<td>pressure, MPa</td>
</tr>
<tr>
<td>$r$</td>
<td>radius, m</td>
</tr>
<tr>
<td>$s$</td>
<td>curvilinear coordinate, m</td>
</tr>
<tr>
<td>$t$</td>
<td>time, s</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature, °C</td>
</tr>
<tr>
<td>$V$</td>
<td>volume, m³</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Greek symbols</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_m$</td>
<td>stress concentration factor for circumferential stress due to pressure at the edge of the hole</td>
</tr>
<tr>
<td>$\beta$</td>
<td>linear thermal expansion coefficient, 1/K</td>
</tr>
<tr>
<td>$\eta$</td>
<td>efficiency</td>
</tr>
<tr>
<td>$\theta$</td>
<td>dummy variable</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Poisson's constant</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density, kg/m³</td>
</tr>
<tr>
<td>$\tau$</td>
<td>time constant, s</td>
</tr>
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</table>

2. Mathematical model of heating the evaporator with hot water

A simplified diagram of the evaporator heating with hot water is shown in Figure 1. The evaporator contains hot water with temperature $T_c$ and the metal temperature is $T_w$. The boiler combustion chamber walls and the convective evaporator give up the heat to air flowing through the furnace chamber. Diagram of water cooling and metal heating in the evaporator after it is suddenly filled with hot water. The heat flow rate transferred from the furnace chamber wall to air will be found taking account of the heat transfer through the tubes and flat bars (longitudinal fins) welded to the tubes. The weighted heat transfer coefficient $h_{cr}$ on the waterwall tube external surface, taking the heat...
exchange through the tube and fin surface into consideration (Fig. 2), can be determined from the following expression

\[
h_{cr}A_{zg}(T_c - T_k) = h_kA_{zr}(T_c - T_k) + h_kA_p(T_c - T_k)\eta_p
\]

(1)

where: \( A_{zg} \) – the surface area of the outer surface of a half of a smooth tube, \( A_{zr} \) – the surface area of the outer surface of a half of a smooth tube between fins, \( A_p \) – the fin surface area.

The expression for calculating the weighted heat transfer coefficient \( h_{cr} \) is

\[
h_{cr} = h_k\left(\frac{A_{zr}}{A_{zg}} + \frac{A_p}{A_{zg}}\eta_p\right)
\]

(2)

3. Numerical model of the evaporator heating using hot-water

Time changes in water and evaporator tubes temperature during the boiler filling with the hot water may be determined using finite volume method [2-4]. Using the finite difference method, a numerical procedure will be derived to calculate the temperature of: water in the waterwall tubes, the downcomer tube walls, air along the downcomer tube height, water in the downcomer tubes, the downcomer tube wall.

3.1. Determination of the unsteady-state distribution of temperature of water, tube walls and air in the combustion chamber

The diagram illustrating the division of the waterwall tube into finite volumes is presented in Fig. 3. The nodes for the wall temperature calculation are located in the middle of the cell, in between the nodes where the water temperature is calculated. The applied staggered difference mesh ensures better accuracy of the calculations.
Determination of the water temperature distribution in the waterwall tubes: The control volume for the waterwall tube is shown in Fig. 4.

Transforming the energy balance equation for a control volume with length $\Delta s$, the following is obtained

$$\tau_c \frac{\partial T_c}{\partial t} + \frac{1}{N_c} \frac{\partial T_c}{\partial s} = - \left( T_c - T_w \right) , \quad N_c = \frac{h_w \pi d_w L_s}{m_c c_c} \quad \text{and} \quad \tau_c = \frac{\left( \pi d_w^2 / 4 \right) \Delta s \rho_c}{\pi d_w \Delta s h_w} = \frac{d_w}{4h_w} \rho_c c_c \quad (4)$$

Equation (4) with conditions will be solved using the finite difference method. The space derivative will be approximated using the backward difference quotient, and the time derivative – using the forward difference quotient

$$\tau_c \frac{T^n_{c,i+1} - T^n_{c,i}}{\Delta t} + \frac{1}{N_c} \frac{T^n_{c,i+1} - T^n_{c,i}}{\Delta s} = - \left( \frac{T^n_{c,i} + T^n_{c,i+1}}{2} - T^n_{w,i} \right) , \quad i = 1, \ldots, N, \ n = 0, 1, \ldots \quad (5)$$

Determination of the waterwall tube wall temperature distribution: The energy balance equation for water flowing in the downcomer tubes is of the same form as Equation (4) for water flowing through the riser tube except that different values of the downcomer tube wall diameter and thickness need to be taken into account. The differential equation will be approximated for the wall in a similar manner

$$\tau_{w,i} \frac{T^n_{w,i+1} - T^n_{w,i}}{\Delta t} + T^n_{w,i} = \frac{1}{2r_w h_w + r_i h_{cr}} \times \left[ 2r_w h_w \left( \frac{T^n_{c,i} + T^n_{c,i+1}}{2} \right) + r_i h_{cr} \left( \frac{T^n_{p,j} + T^n_{p,j+1}}{2} \right) \right] , \quad \tau_{w,i} \quad (6)$$

Water temperature $T^n_{w,i+1}$ at the control volume outlet (Fig. 4) is
\[ T_{c,i+1}^{n+1} = T_{c,i}^{n} - \frac{\Delta t}{\tau_{we}} \left[ \frac{1}{N_c} \left( T_{c,i}^{n} - T_{c,i+1}^{n} \right) + \left( \frac{T_{c,i}^{n} + T_{c,i+1}^{n}}{2} - T_{w,i}^{n} \right) \right], \quad i = 1, \ldots, N, \quad n = 0, 1, \ldots \] (7)

Transforming Equation (6), the formula for the wall temperature in node \( i \) \( T_{w,i}^{n+1} \) is obtained

\[ T_{w,i}^{n+1} = T_{w,i}^{n} + \frac{\Delta t}{\tau_{we}} \left( \frac{1}{2r_w h_w + r_i h_r} \right) \left[ 2r_w h_w \left( \frac{T_{c,i}^{n+1} + T_{c,i+1}^{n+1}}{2} \right) + r_i h_r \left( \frac{T_{p,i}^{n+1} + T_{p,i+1}^{n+1}}{2} \right) \right] - T_{w,i}^{n}, \quad i = 1, \ldots, N, \quad n = 0, 1, \ldots \] (8)

**Determination of the air temperature distribution in the combustion chamber:** Transforming the energy balance equation for the \( i \)-th control volume on the air side presented in Fig. 3, the following is obtained

\[ \tau_p \frac{\partial T_p}{\partial t} + \frac{1}{N_p} \frac{\partial T_p}{\partial s^i} = (T_w - T_p), \quad N_p = \frac{n \pi r h L_a}{m_p c_p} \quad \text{and} \quad \tau_p = \frac{A_k \rho_p c_p}{n \pi r c_r h_r} \] (9)

Equation (9) with conditions will be solved using the finite difference method. The differential equation (9) can be approximated using the following expression

\[ \tau_p \frac{T_{p,i+1}^{n+1} - T_{p,i}^{n+1}}{\Delta t} + \frac{1}{N_p} \frac{T_{p,i+1}^{n} - T_{p,i+1}^{n}}{\Delta s^i} = \left( T_{w,i}^{n} - T_{p,i}^{n} + T_{p,i+1}^{n} \right), \quad i = 1, \ldots, N, \quad n = 0, 1, \ldots \] (10)

Equation (10) is used to determine temperature \( T_{p,i+1}^{n+1} \)

\[ T_{p,i+1}^{n+1} = T_{p,i}^{n} - \frac{\Delta t}{\tau_p} \left[ \frac{1}{N_p} \left( T_{p,i}^{n} - T_{p,i+1}^{n} \right) - \left( T_{w,i}^{n} - T_{p,i}^{n} + T_{p,i+1}^{n} \right) \right], \quad i = 1, \ldots, N, \quad n = 0, 1, \ldots \] (11)

The obtained expressions (7), (8) and (11) make it possible to find the water, wall and air temperature distributions at the time \((t+\Delta t)\) based on known temperature distributions at the time \( t \). Knowing the water temperature \( T_{c,i}^{n} \), the air temperature \( T_{w,i}^{n} \) for \( i = 1, \ldots, N+1 \), and the tube wall temperature \( T_{w,i}^{n} \) for \( i = 1, \ldots, N \), expressions (7), (8) and (11) are used respectively to determine the water temperature \( T_{c,i}^{n+1} \), the air temperature \( T_{p,i}^{n+1} \) for \( i = 1, \ldots, N+1 \), and the tube wall temperature \( T_{w,i}^{n+1} \) for \( i = 1, \ldots, N \).

**3.2. Determination of the unsteady-state distribution of temperature of water and the downcomer tube wall**

If the evaporator is fed from below through the lower downcomers, the water level in the evaporator tubes and in the riser tubes is identical.

**Determination of the water temperature distribution in the downcomer tubes:** The downcomer tube water temperature \( T_{co,i+1}^{n+1} \) will be calculated using Formula (7), which was also used to find the water temperature in the riser tubes. The temperature of water in the downcomer tube is determined from the following equation

\[ T_{co,i+1}^{n+1} = T_{co,i}^{n} - \frac{\Delta t}{\tau_{co}} \left[ \frac{1}{N_{co}} \left( T_{co,i}^{n} - T_{co,i+1}^{n} \right) + \left( T_{co,i}^{n} + T_{co,i+1}^{n} \right) - T_{w,i}^{n} \right], \quad i = 1, \ldots, N+1 \] (12)
Determination of the downcomer tube wall temperature distribution: The energy balance equation for the wall can be expressed as

\[ \tau_{w_0} \frac{dT_{w_0}}{dt} + T_{w_0} = T_{co} \]

Equation (13) will be solved using the explicit Euler method

\[ \tau_{w_0} \frac{T_{w_0,i+1} - T_{w_0,i}}{\Delta t} + T_{w_0,i} = \frac{T_{co,i} + T_{co,i+1}}{2} \]

Solving Equation (14) with respect to \( T_{w_0,i+1} \) gives

\[ T_{w_0,i+1} = T_{w_0,i} + \frac{\Delta t}{\tau_{w_0}} \left( \frac{T_{co,i} + T_{co,i+1}}{2} - T_{w_0,i} \right) \]

4. Optimum heating of the evaporator

The total mass of metal in the entire evaporator including the furnace waterwalls and the convection evaporator (festoon) is \( m_m = 171.9 \times 10^3 \) kg. Assuming that the drum is entirely filled with water, the volume of the water space in the evaporator is \( V_{par} = 59.5 \) m\(^3\). If the drum is only half-filled, the water space in the evaporator is smaller and amounts to \( V_{par} = 46.5 \) m\(^3\); water density: \( \rho = 971.78 \) kg/m\(^3\), steel specific heat \( c = 500 \) J/kgK. Assuming that the drum is half-filled, the surface area of the heat exchange between water and metal is \( A_w = 1923.87 \) m\(^2\).

The optimum temperature of water feeding the evaporator through its lower chambers will be approximated using the following function

\[ T_{w_0} = T_0 + a_w + b_w t \]

The constants \( a_w \) and \( b_w \) will be determined from the condition: \( \sigma_\phi(r, i, t) \leq \sigma_a \), \( i = 1, ..., n_t \) using the least-squares method. The symbol \( n_t \) designates the number of time points at which the calculated stress should be equal permissible stress. The sum of squares of calculated and allowable circumferential stress differences at point P2 situated on the opening edge is minimized

\[ \sum_{i=1}^{n_t} \left[ \int_0^{t_i} T_{w_0,N+1}^{i+1}(\theta) \frac{\partial u(r_i, t, \theta)}{\partial t} d\theta + \alpha_m(p_i - p_o) \frac{D + H}{2H} - \sigma_a \right]^2 = \text{min} \]

where \( T_{w_0,N+1}^{i+1} \) is the water temperature at evaporator outlet.

The analysed time interval is divided into \( (n_t - 1) \) intervals. The circumferential stresses are calculated at each time point \( t_i = (i - 1) \Delta t \) using the rectangle method for evaluating the Duhamel integral. The symbol \( \Delta t \) denotes the time step. Unknown coefficients were determined using Levenberg-Marquardt so that the sum (17) achieves a minimum [4-8]. The evaporator will be heated in two phases. First, the waterwall tubes, the downcomer tubes and the drum are filled with water supplied to the evaporator through its lower headers. Next, after the water in the boiler drum is heated to temperature \( x_1 + T_0 \), oil burners are activated and further heating of the evaporator proceeds so that the water and steam temperatures should change in time according to the optimum temperature history determined for the drum. The following data are assumed for the OP-210M boiler drum to find the optimum temperature of water supplied to the evaporator and the optimum changes in temperature and pressure in the drum:
the drum inner diameter: 1700 mm, the drum wall thickness: 90 mm, the downcomer tube diameter: 90 mm, the downcomer tube wall thickness: 6 mm, heat conductivity coefficient \( k = 42 \text{ W/}(\text{m}\cdot\text{K}) \), specific heat \( c_m = 538.5 \text{ J/(kg}\cdot\text{K}) \), density \( \rho = 7800 \text{ kg/m}^3 \), Young's modulus \( E = 1.96\cdot10^{11} \text{ N/m}^2 \), linear thermal expansion coefficient \( \beta = 1.32\cdot10^{-5} \text{ 1/K} \), Poisson's constant \( \nu = 0.3 \), heat transfer coefficient on the surface: 1000 W/(m\(^2\)\cdot\text{K})

The optimum time variations of the fluid temperature and pressure in the drum were found using the method presented in this paper. The optimum changes of the drum temperature are defined by the following functions (Fig. 5).

\[
T_f - T_0 = a + bt = 48.356 + 0.05775t, \quad t \leq 1800 \text{s} \tag{18}
\]

\[
T_f - T_0 = a + bt + c(t - 1800)^2 = 48.356 + 0.05775t + \left(\frac{7.542815}{10^6}\right)(t - 1800)^2, \quad t \geq 1800 \text{s} \tag{19}
\]

The comparison presented in Fig. 5 proves that Functions (18) and (19) approximate the optimum history of temperature found from the integral equation solution very well. Pressure affects the optimum history of the fluid temperature only in the final phase of the drum heating, for \( t > 2500 \text{ s} \). The optimum history of the evaporator feed water temperature surplus compared to the initial temperature value was determined according to the method presented in this chapter.

\[
T_{wz} = a_{wz} + b_{wz}t, \quad a_{wz} = 74.71 \text{ °C} \text{ and } b_{wz} = 0.0311 \text{ °C/s} \tag{20}
\]
Taking the entire evaporator initial temperature $T_0 = 20\,^\circ C$, it is assumed that the temperature of the water pumped into the evaporator lower header will be constant $a_{wz} + T_0 = 74.71\,^\circ C + 20\,^\circ C = 94.71\,^\circ C$. After the evaporator is heated together with the drum to the temperature of: $a + T_0 = 48.35\,^\circ C + 20\,^\circ C = 68.35\,^\circ C$, the feed pump will be switched off. Denoting the time after which water is no longer supplied to the evaporator as $t_p$, the boiler evaporator will be heated with the optimum temperature determined for the drum.

$$T_f = T_0 + a + \frac{b}{100}(t - t_p), \quad t \leq t_i + t_p \quad (21)$$

$$T_f = T_0 + a + \frac{b}{100}(t - t_p) + \frac{c}{10^6}(t - t_p - t_i)^2, \quad t \geq t_i + t_p \quad (22)$$

$$T_f = 20 + 48.35 + 0.05775(t - 335), \quad t \leq 2135 \quad (23)$$

$$T_f = 20 + 48.35 + 0.05775(t - 335) + \frac{7.542815}{10^6}(t - 2135)^2, \quad t \geq 2135 \quad (24)$$

At the beginning of the start-up process, the lower headers of the boiler waterwalls are supplied with water at the temperature of $74.7^\circ C$ for 335 seconds. After this time, when water from the riser and downcomer tubes is mixed, the water temperature in the drum is $48.3^\circ C$. Next, the water in the evaporator is heated using oil burners so that the temperature should change according to the functions expressed by Eqs. (23) and (24). The changes in the water temperature in the drum are presented in Fig. 6. The history of the water temperature in the drum in the time interval from 0 to 335 seconds, when the boiler evaporator is supplied with hot water through lower headers of the waterwalls and when oil burners are switched off, was determined using the mathematical model of the evaporator being filled with hot water.

5. Conclusions

A new technology of the start-up of the drum boiler was proposed. The drum boiler is filled with hot water at the beginning of the start-up. A transient mathematical model of evaporator filling with hot water has been developed. The temperature of water supplied to the boiler evaporator through lower chambers of the boiler furnace chamber waterwalls will be found from the condition of equality between allowable and circumferential thermal stress on the edge of the opening for the downcomer. The proposed technology of the start-up of the boiler has the following advantages: a) start-up time of the boiler shall be reduced by several tens of minutes, b) reducing the consumption of fuel oil during the boiler start-up, c) after filling the boiler evaporator with hot water and switching on the oil burners the water steam is quickly produced. This provides cooling superheater tubes and prevents tube material from overheating.

References
