Optimum Heating of Thick-Walled Pressure Components Assuming a Quasi-steady State of Temperature Distribution (Postprint)

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Date: 2017-11-02T00:00:00+00:00

Abstract

As a result of the development of wind farms, the gas steam blocks, which shall quickly ensure energy supply in case the wind velocity is too low, are introduced to the energy system. To shorten the start-up time of the gas steam and conventional blocks, the structure of the basic components of the blocks are changed, e.g. by reducing the diameter of the boiler, the thickness of its wall is also reduced. The attempts were also made to revise the currently binding TRD 301 regulations, replacing them by the EN 12952-3 European Standard, to reduce the allowable heating and cooling rates of thick walled boiler components. The basic assumption, on which the boiler regulations allowing to calculate the allowable temperature change rates of pressure components were based, was the quasi - steady state of the temperature field in the simple shaped component, such as a slab, cylindrical or spherical wall.

Full Text

Optimum Heating of Thick-Walled Pressure Components Assuming a Quasi-steady State of Temperature Distribution

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Keywords: pressure component, thermal stress, quasi-steady state, optimum heating

Introduction

The development of wind farms has led to the introduction of gas-steam blocks into energy systems, which must quickly ensure energy supply when wind velocity is too low. To shorten the start-up time of both gas-steam and conventional blocks, the structure of basic components has been modified—for example, by reducing boiler diameter, which also reduces wall thickness. Attempts have also been made to revise the currently binding TRD 301 regulations by replacing them with the EN 12952-3 European Standard, aiming to reduce the allowable heating and cooling rates of thick-walled boiler components. The fundamental assumption underlying these boiler regulations for calculating allowable temperature change rates of pressure components was the quasi-steady state of the temperature field in simply shaped components such as slabs, cylindrical walls, or spherical walls.

The quasi-steady state occurs in structural components of boilers and turbines during long heating or cooling processes running at a constant rate of temperature change. This paper presents a more detailed analysis of the quasi-steady state for both non-weakened walls and walls weakened by openings. We also demonstrate the use of formulas for circumferential stress at the edges of openings to determine allowable heating and cooling rates for a given pressure. These formulas can be applied in practice because analysis of temperature and stress fields in the quasi-steady state can be quickly conducted using the Finite Elements Method (FEM), even for components with complex structures.

The quasi-steady state can be characterized by two factors: (1) constant temperature difference between selected points of the component (Fig. 1 [Figure 1: see original paper]), and (2) constant, time-independent thermal stress (Fig. 2 [Figure 2: see original paper]). The results presented in Figs. 1 and 2 were obtained using the following data: $r_{in} = 0.85$ m, $r_o = 0.94$ m, $c = 511$ J/(kg$\cdot$K), $\rho = 7775$ kg/m^3, $k = 47.3$ W/(m$\cdot$K), $E = 201 \times 10^5$ MPa, $\beta = 1.21 \times 10^{-5}$ 1/K, $\nu = 0.288$.

Let us consider the simplifying assumption that the thermo-physical properties of the wall material—k, c, ρ, E, and ν—are constant and determined for the average temperature over time and wall thickness. The transient heat transfer equation in the quasi-steady state can then be simplified to the form [1, 2]:

$$\nabla^2 T = \frac{v_T}{\kappa} = \text{const} \tag{1}$$

where v_T is the rate of wall temperature change and the symbol ∇^2 denotes the Laplace operator.

Nomenclature

Symbols

c - specific heat of the metal, J/(kg · K)
d - inner diameter of the downcomer, m
D - inner diameter of the drum, m
E - modulus of elasticity, MPa
h - heat transfer coefficient, W/(m² · K)
H - drum wall thickness, m
h_w - heat transfer coefficient in water zone, W/(m² · K)
h_s - heat transfer coefficient in steam zone, W/(m² · K)
k - thermal conductivity, W/(m · K)
p - absolute pressure in the drum, MPa
p_n - gauge pressure in the drum, MPa
p_o - atmospheric pressure, MPa
r - position vector
r_{in} - inner radius of the drum, m
r_o - outer radius of the drum, m
s - wall thickness, m
t - time, s
T - temperature, °C or K
T_m - average temperature over wall thickness
v_T - heating/cooling rate, K/min
u - ratio of outer to inner radius (r_o/r_{in})
α - thermal diffusivity, m²/s
α_T - thermal stress concentration coefficient
β - linear thermal expansion coefficient, 1/K
ν - Poisson' s ratio
ρ - density, kg/m³
σ - normal stress, MPa
σ_ϕ - circumferential stress, MPa
σ_a - allowable stress, MPa
ϕ_{ww}, ϕ_{wz} - shape factors for inner and outer surfaces

Subscripts

w - water zone
s - steam zone
$1, 2$ - points P1 and P2

Quasi-Steady State Analysis for Simple Geometries

The wall thickness is constant, and the external surface $r = r_o$ of the pressure component is thermally insulated:

$$\left.\frac{\partial T}{\partial r}\right|_{r=r_o} = 0 \tag{2}$$

The temperature on the internal surface $r = r_{in}$ rises linearly in time at a rate of v_T:

$$T(r_{in}, t) = T_0 + v_T t \tag{3}$$

Expressing the Laplace operator in a cylindrical or spherical coordinate system leads to the specific form of Eq. (1):

$$\frac{1}{r^m}\frac{d}{dr}\left(r^m \frac{dT}{dr}\right) = \frac{v_T}{\kappa} \tag{4}$$

where $m = 1$ for cylindrical components and $m = 2$ for spherical components. After double integration and determining the constants from boundary conditions (2)-(3), the following expression is obtained for the cylindrical wall:

$$T(r) = T_{in} + \frac{v_T}{\kappa}\left[\frac{r^2 - r_{in}^2}{4} - \frac{r_{in}^2}{2}\ln\left(\frac{r}{r_{in}}\right)\right] \tag{5}$$

The average temperature over the wall thickness is calculated from:

$$T_m = \frac{1}{s}\int_{r_{in}}^{r_o} T(r)dr \tag{6}$$

Substituting Eq. (5) into Eq. (6) with $m = 1$ and integrating gives the average temperature T_m across the cylindrical wall thickness:

$$T_m = T_{in} + \frac{v_T s^2}{\kappa}\phi_{ww} \tag{7}$$

where $u = r_o/r_{in}$ is the ratio of outer to inner radius and $s = r_o - r_{in}$ is the wall thickness.

The circumferential thermal stress σ_{ww} on the inner surface is:

$$\sigma_{ww} = \frac{E\beta v_T s^2}{\kappa(1-\nu)}\phi_{ww} \tag{8}$$

The circumferential thermal stress on the outer surface is calculated from a similar formula:

$$\sigma_{wz} = \frac{E\beta v_T s^2}{\kappa(1-\nu)}\phi_{wz} \tag{9}$$

The shape factors for the internal surface ϕ_{ww} and external surface ϕ_{wz} are given by:

$$\phi_{ww} = \frac{1}{4\ln u} - \frac{1}{2(u^2-1)} \tag{10}$$

$$\phi_{wz} = \frac{u^2}{4\ln u} - \frac{u^2}{2(u^2-1)} \tag{11}$$

Similarly, the average temperature over the sphere thickness can be derived from Eq. (6). The shape factors for a sphere–ϕ_{kw} for the inner surface and ϕ_{kz} for the external surface–are:

$$\phi_{kw} = \frac{1}{6}\left(\frac{u^3-1}{u^3}\right) - \frac{1}{2(u^3-1)} \tag{12}$$

$$\phi_{kz} = \frac{u^3}{6}\left(\frac{u^3-1}{u^3}\right) - \frac{u^3}{2(u^3-1)} \tag{13}$$

The shape factors at the inner and outer surfaces of cylindrical and spherical walls are presented in Fig. 3 [Figure 3: see original paper]. The absolute values of the shape factors for the spherical wall (ϕ_{kw} and ϕ_{kz}) are smaller than the corresponding values for the cylindrical wall (ϕ_{ww} or ϕ_{wz}). Consequently, for the same rate of temperature change, the thermal stresses in spherical walls will be greater than those in cylindrical walls.

Optimum Heating of Components Without Openings

Heating or cooling of components is conducted in a manner that ensures the circumferential thermal stress on the inner surface equals the allowable stress. Substituting Eq. (8) into Eq. (14) gives:

$$\sigma_{ww} = \sigma_a \tag{14}$$

After transformation of Eq. (15), expressions for the heating or cooling rate of thick-walled cylindrical or spherical boiler components are obtained:

Cylindrical wall:

$$v_T = \frac{\sigma_a \kappa(1-\nu)}{E\beta s^2 \phi_{ww}} \tag{16}$$

Spherical wall:

$$v_T = \frac{\sigma_a \kappa(1-\nu)}{E\beta s^2 \phi_{kw}} \tag{17}$$

The temperature change rate is positive during heating and negative during cooling. The wall temperature depends on the fluid temperature, but the temperature difference between selected points in the components remains constant and independent of time and fluid temperature value.

Quasi-Steady Distributions in Components with Openings

The quasi-steady state temperature distribution in complex-shaped components can be determined using Eq. (1) with appropriate boundary conditions. Equation (1) describes a steady-state temperature distribution in the body $T(r, t)$ with uniformly distributed heat sinks having specific power equal to v_T/κ.

We now discuss the application of the quasi-steady state approach for determining allowable heating rates in complex-shaped components. Since stresses at points P1 and P2 (Fig. 4 [Figure 4: see original paper]) limit the heating rate of boiler drums or other thick-walled vessels weakened by nozzles [3-6], thermal and pressure stresses are calculated for these points.

The following geometrical dimensions of the boiler drum were assumed for calculations: inner diameter $D = 1.7$ m, inner diameter of downcomer $d = 0.09$ m, drum wall thickness $H = 0.09$ m, downcomer wall thickness $h = 0.006$ m. Physical properties of the boiler material were: $k = 42$ W/(m · K), $c = 538.5$ J/(kg · K), $\rho = 7800$ kg/m³, $E = 1.96 \times 10^{11}$ N/m², $\beta = 1.32 \times 10^{-5}$ 1/K, and $\nu = 0.3$.

Time variations of circumferential stress at point P1 for different heat transfer coefficients were computed (Fig. 5 [Figure 5: see original paper]). Analysis shows that at low heat transfer coefficient values ($h = 500$ W/(m²·K)), the quasi-steady state is established only after prolonged heating at a constant rate. After 4000 seconds of heating, stresses did not reach the constant values characteristic of quasi-steady state.

Equivalent stress distributions in the boiler drum-downcomer junction, determined using ANSYS for different pressures and heating rates, are depicted in Figures 6, 7, and 8. The maximum stress due to pressure occurs at point P1, located at the hole edge (Fig. 6 [Figure 6: see original paper]). The equivalent stress at point P1 is about five times larger than at point P2 when the drum is loaded only by internal pressure. If the pressure inside the drum equals zero, the equivalent thermal stress at point P2 is higher compared to point P1 (Figs. 7a and 7b). When pressure increases to design pressure $p = 10.87$ MPa, stresses due to pressure for $v_T = 1$ K/min at point P1 exceed thermal stresses, and the largest equivalent stress occurs at point P1 (Fig. 8a). After increasing the heating rate to $v_T = 12$ K/min, negative thermal stresses exceed positive pressure stresses, and the largest equivalent stress appears at point P2 (Fig. 8b [Figure 8: see original paper]).

The stress concentration coefficients α_T at points P1 and P2 (Fig. 4) for the quasi-steady state as a function of heat transfer coefficient h are presented in Fig. 9 [Figure 9: see original paper].

Thermal stresses determined from Eqs. (8) and (9) refer to simple shapes such as hollow cylinders and spheres. In practice, components used in power units often contain openings that create material discontinuities causing stress concentration, which must be considered when determining allowable heating or cooling rates. Most often, the assumed value of thermal stress concentration coefficient at the edge of an opening is $\alpha_T = 2$, without considering the influence of heat transfer coefficient h on this value.

However, the thermal stress concentration coefficient at opening edges in complex-shaped components depends on the heat transfer coefficient h. Since the boiler drum has openings for downcomers, intense stress concentration occurs at the opening edge. The stress concentration coefficient at the opening edge is strongly influenced by the heat transfer coefficient on the inner surface of both the boiler drum and downcomer. The value of α_T is lower at point P1 compared to point P2, regardless of the heat transfer coefficient value (Fig. 9). When $h = 10000$ W/(m$^2 \cdot$ K), the stress concentration coefficient is $\alpha_T = 1.53$ at point P1 and $\alpha_T = 1.86$ at point P2.

The thermal stress concentration coefficient was calculated using:

$$\alpha_T = \frac{\sigma_{\phi,1}}{\sigma_{\phi,0}} \tag{18}$$

where $\sigma_{\phi,1}$ denotes circumferential stress at the opening edge at point P1, and $\sigma_{\phi,0}$ represents circumferential stress on the inner surface of the boiler drum or pressure vessel without openings.

The total circumferential and equivalent stress at the opening edge at points P1 and P2 can be approximated linearly as a function of heating rate. At the beginning of boiler start-up, when gauge pressure equals zero, the absolute values of circumferential and equivalent stresses at points P1 and P2 are equal (Fig. 10 [Figure 10: see original paper]). Total circumferential and equivalent stresses presented in Fig. 11 were computed considering both pressure and thermal loads.

The circumferential stress at points P1 and P2 for pressure $p = 10.87$ MPa can be approximated by a linear function with respect to v_T:

$$\sigma_\phi = a + b v_T \tag{19}$$

Allowable boiler drum heating rates v_{T1} and v_{T2} can be determined from the condition $\sigma_\phi = \sigma_a$, where circumferential stress σ_ϕ is given by Eq. (19). The same method can be applied to other complex-shaped components. The issue of optimizing heating and cooling of thick-walled boiler components was presented in [5] without assuming quasi-steady state. However, the developed method for determining allowable cooling or heating rates is simple and can be easily applied in practice.

Components with Non-uniform Circumferential Temperature Distribution

Uneven temperature distribution occurs on the drum circumference due to the presence of water and steam zones. During boiler start-up, steam condenses on the inner surface of the drum in the steam zone. During condensation, the upper part of the drum heats very quickly because the heat transfer coefficient is extremely high. In the bottom portion occupied by water, the heat transfer coefficient from water to the inner drum surface is low, and heating of the lower part occurs much more slowly. Different heating rates between upper and lower portions create a temperature difference between the top and bottom of the drum, with the upper part being hotter. A similar phenomenon occurs in horizontal pipeline sections and superheater headers.

At the beginning of boiler start-up, pipes are cold and water vapor condenses on pipe walls. Water accumulates at the bottom of horizontal piping and other pressure components, especially when drainage elements malfunction. This occurs when horizontal pipelines and superheater headers are plastically deformed. In later start-up stages, superheated steam flows through the upper pipe section while saturated water remains at the bottom, causing very large temperature differences on the pipe circumference—reaching up to about 200 K. This often leads to permanent pipe and header deformation, which further impairs proper drainage. Stresses caused by temperature non-uniformity on the pipe circumference may exceed stresses caused by temperature difference across the pipeline thickness. The current European Standard EN 12952-3 does not account for thermal stresses caused by temperature differences on cylindrical element circumference. This paper analyzes heating of horizontal thick-walled components considering different heat transfer coefficient values in water and steam zones (Fig. 12 [Figure 12: see original paper]).

Calculations assume fluid temperature increases linearly with time, eventually forming a quasi-steady state temperature field. This paper examines thermal and strength loads on the OP-210M steam boiler drum during startup. Figure 13 [Figure 13: see original paper] illustrates time changes of water level, with the zero level defined as $H_w = 10$ cm below the horizontal plane through the drum axis. Water temperature in the evaporator increases, raising water specific volume and water level in the drum. After about 12000 seconds, excess water discharges from the drum, lowering the level to near zero. Time variations of saturation pressure in the drum are depicted in Fig. 14 [Figure 14: see original paper].

Measured time variations of drum temperature at seven points (shown in Fig. 12) indicate significant temperature differences on the drum circumference, especially during initial heating. The maximum temperature difference between top and bottom does not exceed 50 K. However, such large temperature differences cause high thermal stresses. Analysis shows large temperature differences on the perimeter during initial heating stages, creating additional thermal stresses. Current boiler standards do not account for stresses due to temperature differences on horizontal element perimeters, considering only quasi-stationary stresses caused by temperature differences across wall thickness.

Assuming a quasi-stationary temperature field in the pressure element, we account for temperature differences on the component circumference caused by different heat transfer coefficients on the inner surface. Time variations of circumferential stress at the drum-downcomer junction edge at points P1 and P2 are shown in Figs. 15 and 16. Analysis reveals that absolute thermal stress values at point P2 are much greater than at point P1. The heat transfer coefficient in the water area exceeds 1000 $W/(m^2 \cdot K)$. Due to higher compressive thermal stress and lower tensile stress from pressure at point P2, the permissible drum heating rate should be determined based on stresses at point P2. Changes in circumferential and equivalent stress at points P1 and P2 as functions of heating rate are shown in Fig. 17. Figure 17a shows gauge pressure $p = 0$, while Fig. 17b [Figure 17: see original paper] shows pressure equal to the design value of 10.87 MPa.

During boiler start-up, the heat transfer coefficient in the water zone is lower than in the steam zone. In such cases, the maximum allowable heating rate is determined by stresses at point P1. As previously explained, faster heating of the upper drum portion results from vapor condensation on the inner surface, where the heat transfer coefficient is very high. After about 15000 seconds, the temperature difference between top and bottom reduces considerably (Fig. 14). Thermal stress concentration coefficients can be quickly determined using FEM, even for very complex shapes. For flat, cylindrical, and spherical walls with one-dimensional temperature distribution, the thermal stress concentration coefficient does not depend on the heat transfer coefficient value. However, for complex-shaped structural components, the heat stress concentration coefficient depends on the heat transfer coefficient on their inner surface.

For a typical practical case [7] with $h_w = 1000$ $W/(m^2 \cdot K)$ and $h_s = 10000$ $W/(m^2 \cdot K)$, the permissible heating rate at the beginning of boiler start-up (when $p = 0$) is approximately 3 K/min. Currently, boiler manufacturers recommend a permissible drum heating rate of about 1 K/min. The allowable heating rate of 3 K/min derived from FEM analysis (Fig. 17a) is significantly higher than the manufacturer-prescribed value, potentially allowing boiler start-up time to be reduced by a factor of three.

Conclusions

The assumption of a quasi-steady temperature field significantly simplifies determination of allowable fluid temperature change rates during heating and cooling of thick-walled boiler components. This paper has analyzed quasi-steady state temperature fields and stresses in the drum-downcomer junction, considering different heat transfer coefficients in water and steam spaces. The presented graphs allow determination of heating rate limits at the beginning of boiler start-up (when pressure equals 0) and at the end of starting (when pressure equals nominal pressure). Permissible drum heating rates, assuming both circularly symmetric and asymmetric temperature fields in the drum, are several times higher than limit values recommended by boiler manufacturers.

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Note: Figure translations are in progress. See original paper for figures.