Thermal-Hydraulics of Evaporating Tubes in the Forced Circulation Loop of a Steam Boiler

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Rifled evaporating tubes are applied in a steam boiler with the aim to increase the steam-water mixture turbulization and to prevent tubes' wall burnout. The evaporating rifled tubes and the working fluid forced circulation are applied in the steam boiler at the Thermal Power Plant "Kolubara B" that is being built by the Electric Power Utility of Serbia. In order to investigate operating characteristics of the steam boiler of such an advanced design, a simulation and analysis of complex coupled thermal processes on the furnace gas side and thermal-hydraulics in the evaporating tubes were performed for the whole range of the plant designed operating loads. In this paper the methodology for the hydraulic calculations of forced circulation loops is presented. The rifled tubes thermal-hydraulics is calculated. An acceptable temperature of the evaporating tube wall, even under a high void fraction of the coolant is obtained. It is shown that the rifled tubes prevent the occurrence of the critical heat transfer conditions. The obtained increased thermal safety margin of the rifled tubes is compared with the safety margin of the smooth tubes for uniform and variable heat loads between walls of the boiler furnace. The influence of the rifled tubes on the increase of the hydraulic resistance in the circulation loop is analyzed.

Keywords : *steam boiler, rifled tubes, forced circulation, thermal-hydraulic calculation.*

1. INTRODUCTION

The coal-fired 350 MW_e Thermal Power Plant "Kolubara B" is being built by the Electric Power Utility of Serbia. A tower design of the plant steam boiler is applied with an evaporator in a forced circulation loop. The forced flow is provided by circulation pumps added to the circulation loop. The circulation pumps pressure head contributes to the buoyancy forces between the descending subcooled water flow in the downcomer and the ascending two-phase mixture flow in the evaporating tubes. The sum of pumps head and buoyancy forces overcomes the friction and local pressure losses along the circulation loop. Also, rifled evaporating tubes with internal helical ribs are applied to increase steam-water mixture turbulization and prevent tubes' wall burnout. The sliding pressure is applied for the boiler load control.

A complete insight into two-phase flow and heat transfer conditions in evaporating tubes of the steam boiler is needed to design a reliable cooling of evaporating tubes walls, especially in the furnace zone of the evaporator, where the tubes are exposed to the highest heat fluxes. It is also necessary to predict the dependence of the evaporating tubes thermal-hydraulic conditions on the load change. The sliding pressure control during the load variation causes the change of the water and steam density ratio, which can lead to a substantial increase of the steam void fraction. The two-phase mixture velocity is increased as a consequence of the steam void fraction increase. Further, the liquid film accelerates and depletes on the evaporating tube inner wall, and there is a possibility for the wall dry-out and critical heat flux occurrence. The critical heat flux conditions are characterized by an abrupt increase of the tube wall temperature, with the possibility of the tube wall thermal damage. These conditions are also known under the name burnout.

In order to prevent the evaporating tubes burnout, the rifled tubes are applied in the furnace evaporating zone. The helical ribs and channels on the inner pipe wall induce the swirl flow and the centrifugal force that separates the liquid phase from the two-phase mixture towards the tube wall. This effect enhances the wall wetting and prevents the critical heat transfer occurrence even under the high steam void fraction flow conditions. Hence, compared with the classical tubes with the smooth inner wall, the rifled tubes enable reliable cooling even under the lower coolant mass fluxes. The lower coolant circulation rates require less energy consumption for the circulation pumps operation and reduce the plant operational costs.

The models of various levels of complexity regarding the process physical presentation and applied mathematical methods of solving have been applied to the thermal-hydraulic simulation and analyses of the steam boiler circulation loop and the two-phase flow in the evaporating tubes. Steady-state and homogeneous

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two-phase flow models were applied for the working fluid flow in the downcomer and evaporating tubes of the steam boiler circulation loops in [1,2]. The normative method for the steam boilers hydraulics calculation was developed in the former USSR [3]. This method provides reliable results in engineering applications since it is based on the basic principles of one-phase and two-phase flows and on experimental data bases regarding some important phenomena of two-phase flow in geometry of steam boiler components, such as water and steam slip in evaporating tubes or two-phase local and frictional pressure drops. More refined models of two-phase flow, applicable to the steam boiler evaporating tubes and based on the multi-fluid models of two-phase flow are presented for example in [4-6]. Experimental data and corresponding correlations on heat transfer coefficients, friction pressure loss coefficients and critical heat fluxes in smooth and rifled evaporating tubes of steam boilers are presented in [7,8].

In this paper detailed thermal-hydraulic simulation and analyses of the forced circulation loop of the large steam boiler that is being built in the Thermal Power Plant “Kolubara B” are performed. The rifled tubes in the furnace zone of the evaporating tubes are considered. Calculations are performed for designed load range from 40 % to 100 % of the full power. A dependence of all important thermal and hydraulic parameters in the evaporating tubes on the boiler load is predicted, such as working fluid flow rate, steam void fraction, circulation number, circulation velocity, critical heat flux along the evaporating tubes, etc. In order to demonstrate the influence of the rifled tubes on the increase of the evaporating tubes thermal margin against the burnout and on the circulation loop friction pressure drop, the comparative calculations with the evaporating smooth tubes are performed and compared. The heat loads of the evaporating tubes are the results of the zonal thermal calculation for the furnace and flue gas side of the steam boiler. The model based on integral and local mass, momentum and energy balances is developed for these purposes. The integral balances determine mass flow rates and pressure distributions between different tubes of the downcomer and riser, while the local balances enables prediction of flow parameters along individual tubes.

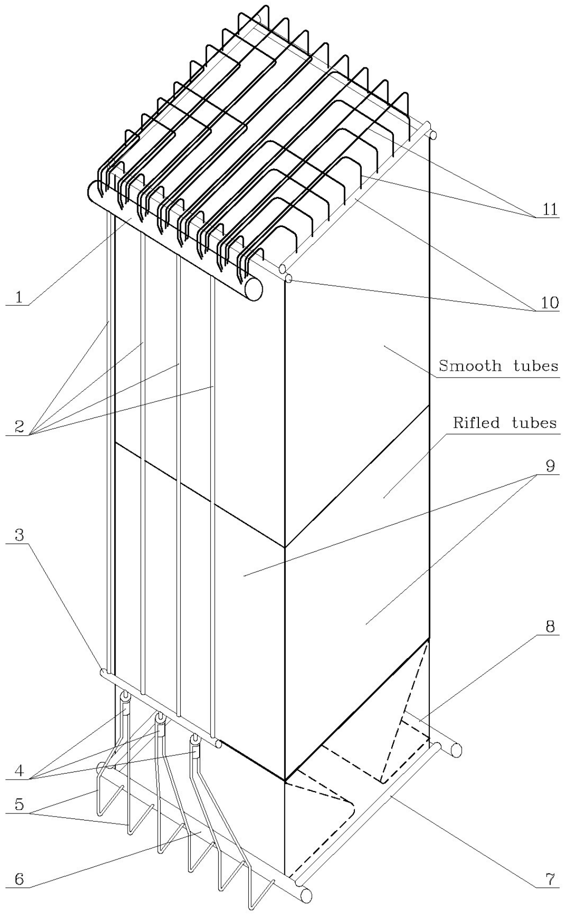
2. THE STEAM BOILER EVAPORATOR AT THE THERMAL POWER PLANT “KOLUBARA B”

The steam boiler at the 350 MW_e coal fired Thermal Power Plant “Kolubara B” is designed for the lignite with the low heating value of 6700 $kJkg^{-1}$. The steam boiler parameters are presented in Table 1 according to the specifications of the boiler designer “Combustion Engineering”.

The steam boiler evaporating tubes are heated by thermal radiation and convection. The evaporating tubes are through a drum, downcomer tubes, circulation pumps, connecting tubes and headers connected into the forced circulation loop. Figure 1 shows the main parts of the circulation loop as follows.

Table 1. Design parameters of the steam boiler at the Thermal Power Plant “Kolubara B”

Live steam generation,	D = 292 kgs^{-1}
Live steam pressure,	p_s = 18.0 MPa
Live steam temperature,	t_s = 540 ºC
Reheated steam mass flow rate,	D_r = 269 kgs^{-1}
Reheated steam pressure,	p_{rs} = 4.0 MPa
Reheated steam temperature,	t_{rs} = 540 ºC
Steam pressure at the reheater inlet,	p_r = 4.2 MPa
Steam temperature at the reheater inlet,	t_r = 330 ºC
Feedwater temperature,	t_{fw} = 250 ºC



1 – Boiler drum; 2 – Downcomer tubes (suction section); 3 – Header; 4 – Circulation pumps; 5 – Downcomer tubes (discharge section); 6 – Header of the furnace front wall; 7 – Header of the furnace lateral wall; 8 – Header of the furnace rear wall; 9 – Evaporating tubes; 10 – Upper headers; 11 – Connecting tubes

Figure 1. Scheme of the steam boiler evaporator

Water flows through four downcomer non-heated tubes from the boiler drum (1) to the header (3), and then to the circulation pumps (4). Three circulation pumps are mounted, where two operate and one is in reserve. The discharge section of the downcomer tubes, from the pumps to the header of the front furnace wall, consists of six tubes (5), where two tubes are connected to each circulation pump. The header of the front furnace wall (6) is connected with the headers of the lateral walls (7), where these are connected via the header of the rear furnace wall (8). The membrane type evaporating tubes (9) are mounted on the furnace walls and in the convective channel. At the top of gas channel they are introduced into the upper headers (10). From each upper header the steam-water mixture flows to the boiler drum through eight connecting tubes (11). The inner side of the furnace wall tubes is rifled, while it is smooth in tubes of the convective channel. Dimensions of the smooth tubes are Ø38.1 x 4.191 mm. The rifled tube is depicted in Figure 2, and its dimensions are presented in Table 2.

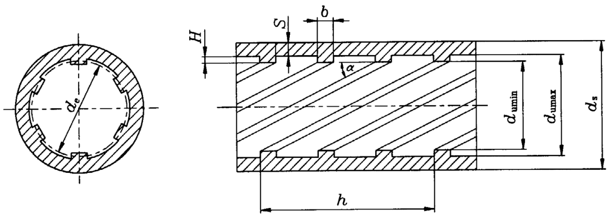


Figure 2. Characteristic dimensions of the rifled tubes

Table 2. Characteristic dimensions of the rifled tubes

Outer tube diameter,	d_s = 0.0381 m
Minimum tube wall thickness,	s = 0.0034845 m
Maximum inner diameter,	d_{umax} = 0.031131 m
Minimum inner diameter,	d_{umin} = 0.028931 m
Equivalent inner diameter,	d_e = 0.029718 m
Rib height,	H = 0.0011 m
Thread angle,	α = 30º
Thread height/step,	h = 0.15 m
Rib width,	b = 0.0035 m
Number of ribs,	z = 6
Characteristic number of the thread height, $F_B = \dfrac{2\pi d_e}{gh^2} = 2.658\ s^2/m^2$	
Rib efficiency,	$\eta_R = \dfrac{H}{d_e} z = 0.222$

3. MODELLING APPROACH

The scheme of the boiler evaporator circulation loop is shown in Figure 3.

The corresponding dimensions are presented in Table 3. As it is shown in Figure 3, the furnace (z_I) and convective (z_{II}) sections are separately treated along the boiler evaporating walls. Each section is discretized with corresponding number of control volumes for the calculation of flow and thermal parameters. The following assumptions are introduced in the modeling of the circulation loop flow: only steady-state conditions are considered, the downcomer tubes are adiabatically isolated, the water and steam densities in the downcomer and evaporating tubes are equal to the corresponding saturation densities determined by the drum pressure.

Table 3. Heights and lengths of evaporator's sections

Total height of the downcomer tubes, z_{dc} = 75.570 m
Height of the non-heated section of the riser tubes, z_{nh} = 5.600 m
Height of the evaporator in the furnace, z_I = 39.500 m
Height of the evaporator in the convective section, z_{II} = 30.470 m
Length of the suction section of the downcomer tubes, l_1 = 67.426 m
Length of the discharge section of the downcomer tubes, l_2 = 8.815 m
Non-heated length of the riser tubes in the front and rear wall, l_{nh} = 12.800 m
Non-heated length of the riser tubes in the lateral walls, l_{nh} = 5.372 m
Evaporator length in the furnace front and rear walls, l_3 = 40.200 m
Evaporator length in the furnace lateral walls, l_3 = 39.500 m
Evaporator length in the boiler convective section, l_4 = 30.274 m
Length of the connecting tubes at the furnace front wall, l_5 = 8.870 m
Length of the connecting tubes at the furnace rear wall, l_5 = 29.770 m
Average length of the connecting tubes at the furnace lateral walls, l_5 = 21.474 m

3.1 Mass balance

According to the presented scheme of the evaporator circulation loop in Figure 3, the integral mass balance of the fluid flow is written as:

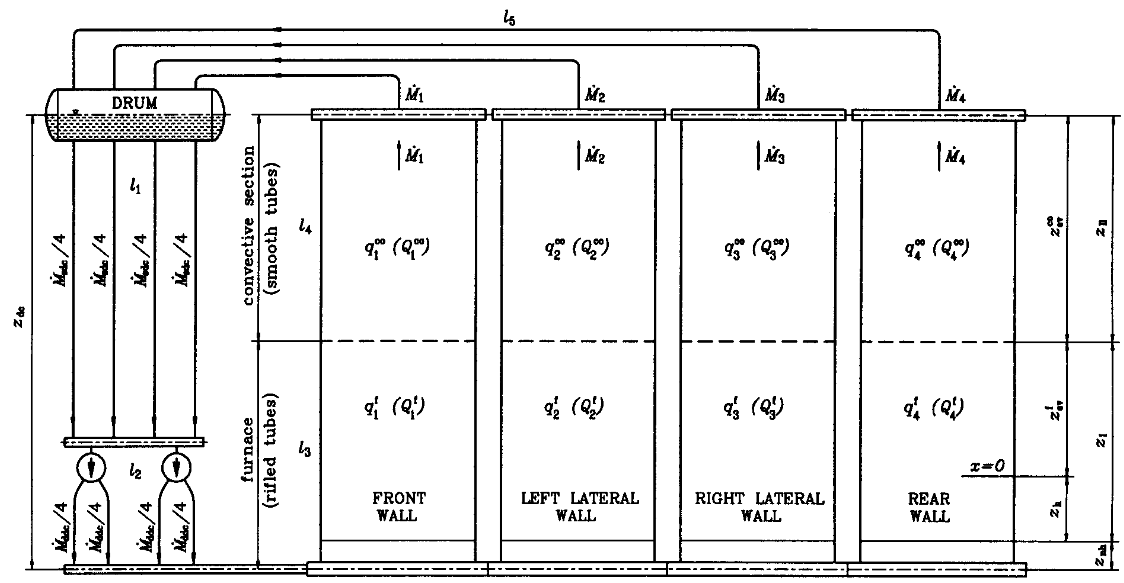


Figure 3. Scheme of the evaporator circulation loop

$$\dot{M} = \dot{M}_{\text{sdc}} = \dot{M}_{\text{ddc}} = \dot{M}_1 + \dot{M}_2 + \dot{M}_3 + \dot{M}_4 = \text{const}\,, \quad (1)$$

where $\dot{M}_{\text{sdc}}$ is the total mass flow rate in the suction section of the downcomer tubes, $\dot{M}_{\text{ddc}}$ is the total mass flow rate at the discharge section of the downcomer tubes, $\dot{M}_1$ is the mass flow rate in the evaporating tubes in the front wall, $\dot{M}_2$ is the mass flow rate in the evaporating tubes in the left lateral wall, $\dot{M}_3$ is the mass flow rate in the evaporating tubes in the right lateral wall and $\dot{M}_4$ is the mass flow rate in the evaporating tubes in the rear wall. Due to a different geometry of the evaporating walls, the corresponding mass flow rates are also different, except for the left and right lateral wall where the flow rates are equal ($\dot{M}_2 = \dot{M}_3$), since these two walls are symmetrical.

The local mass balance for the working fluid (water or steam-water mixture) flow in a tube of constant cross section is presented in the following simple form:

$$\frac{\mathrm{d}(\rho w)}{\mathrm{d}z} = 0\,, \quad (2)$$

which results in the constant mass velocity (mass flux) along the tube.

3.2 Energy balance

The integral energy balance of the absorbed heat rates on the evaporating walls in the furnace and in the convective channel is expressed as:

$$\int_{l_3} q^{\text{f}} O\,\mathrm{d}z + \int_{l_4} q^{\text{co}} O\,\mathrm{d}z = Q^{\text{f}} + Q^{\text{co}}\,, \quad (3)$$

where q^{f} is the heat flux on the evaporating walls in the furnace, q^{co} is the heat flux on the evaporating walls in the convective section, O is the perimeter of the furnace cross section, Q^{f} is the heat load transfered to the evaporating walls in the furnace and Q^{co} is the heat load transfered to the evaporating loads in the convective section.

The heat load transferred to the evaporator in the furnace and in the convective boiler section is divided to each wall according to its geometry. In case of a uniform heat load distribution between the evaporating walls, which should exist due to the tangential firing of the furnace, the heat load is linearly dependent on the surface of the corresponding wall. But, a non-uniform heat load distribution can be encountered in practice and it is considered in this paper too.

Therefore, the absorbed heat load on the evaporating wall in the furnace Q_{iw}^{f} (whose height is $z_{\text{h}} + z_{\text{ev}}^{\text{f}}$, as depicted in Fig. 3) and in the convective section $Q_{\text{iw}}^{\text{co}}$ (which height is $z_{\text{ev}}^{\text{co}}$, Fig. 3), is calculated as:

$$\sum_{\text{iw}=1}^{4} Q_{\text{iw}}^{\text{f}} + \sum_{\text{iw}=1}^{4} Q_{\text{iw}}^{\text{co}} = Q^{\text{f}} + Q^{\text{co}}\,. \quad (4)$$

Steam generation in the observed wall at the outlet of the furnace is

$$\left(D_{\text{out}}^{\text{f}}\right)_{\text{iw}} = \frac{Q_{\text{iw}}^{\text{f}} - \dot{M}_{\text{iw}}\Delta h_{\text{d}}}{r}\,, \quad (5)$$

where r is the latent heat of evaporation, while Δh_{d} is the water subcooling in the drum that is calculated as the difference of saturated water enthalpy and the enthalpy of water at the economizer outlet divided by the circulation number

$$\Delta h_{\text{d}} = \frac{h' - h_{\text{out}}^{\text{eco}}}{\kappa}\,. \quad (6)$$

Steam mass flow rate at the outlet of the convective section, and at the same time at the outlet of the evaporator is determined by the following expression:

$$\left(D_{\text{out}}^{\text{co}}\right)_{\text{iw}} = \left(D_{\text{out}}\right)_{\text{iw}} = \left(D_{\text{out}}^{\text{f}}\right)_{\text{iw}} + \frac{Q_{\text{iw}}^{\text{co}}}{r} \quad (7)$$

The steam flow quality x at the inlet and outlet of the evaporator sections is calculated according to the generated steam mass flow rates. The boiling boundary in the tubes of the furnace walls ($x = 0$) is defined by the calculation of the heated height z_{h}, as presented below. Steam flow quality at the evaporator outlet from the furnace, and at the same time at the evaporator inlet in the convective section is determined as:

$$\left(x_{\text{out}}^{\text{f}}\right)_{\text{iw}} = \left(x_{\text{in}}^{\text{co}}\right)_{\text{iw}} = \frac{\left(D_{\text{out}}^{\text{f}}\right)_{\text{iw}}}{\dot{M}_{\text{iw}}}\,, \quad (8)$$

while the quality at the outlet of the convective section, i.e. at the outlet of the whole evaporator of the observed wall is determined as:

$$\left(x_{\text{out}}^{\text{co}}\right)_{\text{iw}} = \left(x_{\text{iz}}\right)_{\text{iw}} = \frac{\left(D_{\text{out}}^{\text{co}}\right)_{\text{iw}}}{\dot{M}_{\text{iw}}}\,. \quad (9)$$

Relation between the steam flow volume fraction and the flow quality is given by the following relation:

$$\beta = \frac{1}{1 + \dfrac{\rho''}{\rho'}\left(\dfrac{1}{x} - 1\right)}\,. \quad (10)$$

One of the main tasks in the prediction of gas-liquid two-phase flow is the slip calculation (the ratio between the gas and liquid phase velocity). The same task can be formulated as the prediction of the gas phase volume fraction in the two-phase mixture – the void fraction. Depending on the two-phase flow modeling approach, this task is solved by the application of the empirical correlations for the void or slip [9], by the application of the drift flux models [4,9], or through the prediction of the gas – liquid phase interface friction in the multifluid models of two-phase flow [4-6]. The slip between the gas and liquid phase is more exaggerated at lower pressures and lower two-phase flow mass velocities. At higher pressures and two-phase flow fluxes that are characteristics of the steam boilers at the power plants, the slip is less important and in some cases even the homogeneous two-phase flow model could be applied with satisfactory reliability of the calculation [1,2,9]. In this paper, the slip between the phases and its influence on the steam void fraction in two-phase flow is taken into account, especially because of the operating conditions on lower loads and the corresponding lower sliding pressures. The steam void fraction is determined according to the experimental data base developed for the two-phase flow in steam boiler tubes [3]. According to the approach in [3] the void (φ) is calculated as the product of the steam flow volume fraction and the corresponding empirical slip coefficient (C):

$$\varphi = C\beta \,. \tag{11}$$

Slip coefficient in (11) is determined from the experimental chart in Figure 4, where C depends on the pressure p, steam flow volume fraction β and two-phase flow mixture velocity w_m. Mixture velocity is predicted by the circulation velocity ($w_o = \dot{m}_{iw} / \rho'$) with the following relation:

$$w_m = w_o \left[1 + x \left(\frac{\rho'}{\rho''} - 1 \right) \right]. \tag{12}$$

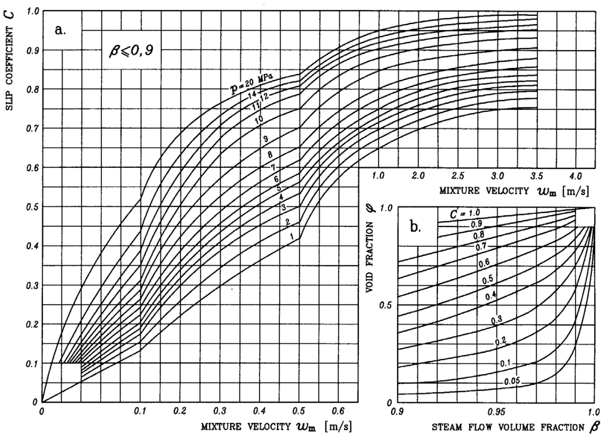


Figure 4. Chart for the slip coefficient *C* prediction

3.3 Momentum balance

The integral momentum balance equation in case of forced calculation along the circulation loop is written in the following form:

$$\int_{l_1} \frac{dp}{dz} dz + \int_{l_2} \frac{dp}{dz} dz + \int_{l_3} \frac{dp}{dz} dz + \int_{l_4} \frac{dp}{dz} dz + \int_{l_5} \frac{dp}{dz} dz = \Delta p_p \,, \tag{13}$$

where l_i denotes the length of the i-th section of the circulation loop (Fig. 3). The circulation pump head (Δp_p), as well as its efficiency (η_p) are defined by the empirical pump characteristics in Figure 7.

The local momentum balance for a tube flow is expressed as the pressure change that consists of the friction pressure drop, local pressure drops, gravitational pressure change and acceleration pressure change:

$$\Delta p = \frac{dp}{dz} = \left(\frac{dp}{dz} \right)_{fr} + \left(\frac{dp}{dz} \right)_{loc} + \left(\frac{dp}{dz} \right)_{g} + \left(\frac{dp}{dz} \right)_{a} . \tag{14}$$

The friction pressure drop for one-phase flow is:

$$\left(\frac{dp}{dz} \right)_{fr} = \lambda_1 \frac{l_i}{d_e} \frac{\dot{m}^2}{2\rho'} . \tag{15}$$

The friction pressure drop in case of steam-water two-phase flow is calculated by applying the two-phase flow multiplier Φ_{vo}

$$\left(\frac{dp}{dz} \right)_{fr} = \Phi_{vo} \left(\frac{dp}{dz} \right)_{lo} , \tag{16}$$

where

$$\left(\frac{dp}{dz} \right)_{lo} = \lambda_1 \frac{l_i}{d_e} \frac{\dot{m}^2}{2\rho'} , \tag{17}$$

is the pressure drop calculated for the liquid phase which fills the whole cross section of the flow channel in which two-phase flow exists. The two-phase flow multiplier (Φ_{vo}) represents the coefficient of the increase of the one-phase friction pressure drop due to the two-phase flow. It depends on various two-phase flow parameters, such as phases' thermo-physical characteristics and pressure level, quality, two-phase flow mass velocity etc. Several empirical approaches are developed for its determination. In this paper, the empirical relation of Friedel [7] is applied for the calculation of two-phase multiplier for flows in smooth tubes and steam flow qualities lower than critical ones (here the criticality denotes the phenomenon of the Critical Heat Transfer)

$$\Phi_{vo} = A + 3.43 x^{0.685} (1-x)^{0.24} \left(\frac{\rho'}{\rho''} \right)^{0.8} \left(\frac{\mu''}{\mu'} \right)^{0.22} \cdot \left(1 - \frac{\mu''}{\mu'} \right)^{0.89} Fr_1^{-0.047} We_1^{-0.0334} , \tag{18}$$

where

$$A = (1-x)^2 + x^2 \left(\frac{\rho'}{\rho''} \frac{\lambda_1}{\lambda_2} \right). \tag{19}$$

The Froud number is calculated as:

$$Fr_1 = \frac{\dot{m}^2}{g d_u \rho'^2} , \tag{20}$$

and the Weber number as:

$$We_1 = \frac{\dot{m}^2 d_u}{\rho' \sigma} . \tag{21}$$

The following correlations are applied for the friction coefficients for water λ_1 and steam λ_2 [7]

$$\lambda_i = \left(0.86859 \ln \frac{Re_i}{1.964 \ln Re_i - 3.8215} \right)^{-2}$$

for $Re_i > 1055$, (22)

$$\lambda_i = \frac{64}{Re_i} \quad \text{for} \quad Re_i \le 1055, \tag{23}$$

where

$$Re_i = \frac{\dot{m} d_u}{\mu_i} \quad \text{and} \quad i = 1,2. \tag{24}$$

For two-phase flows with steam qualities higher than critical ones, the Beattie's correlation is applied [7]:

$$\Phi_{vo} = \left[1 + x\left(\frac{\rho'}{\rho''} - 1\right)\right]^{1.8} \left(\frac{\rho''}{\rho'}\right)^{0.8} \left(\frac{\mu''}{\mu'}\right)^{0.2}. \tag{25}$$

The critical steam flow quality is determined by the Kastner correlation [8]:

for $0.49 \le p \le 2.94$ MPa

$$x_{cr} = 25.6\left(q \cdot 10^3\right)^{-\frac{1}{8}} \dot{m}^{-\frac{1}{3}} \left(d_u \cdot 10^3\right)^{-0.07} e^{0.1715p}, \tag{26a}$$

for $2.94 \le p \le 9.80$ MPa

$$x_{cr} = 46\left(q \cdot 10^3\right)^{-\frac{1}{8}} \dot{m}^{-\frac{1}{3}} \left(d_u \cdot 10^3\right)^{-0.07} e^{-0.0255p}, \tag{26b}$$

for $9.80 \le p \le 19.60$ MPa

$$x_{cr} = 76.6\left(q \cdot 10^3\right)^{-\frac{1}{8}} \dot{m}^{-\frac{1}{3}} \left(d_u \cdot 10^3\right)^{-0.07} e^{-0.0795p}. \tag{26c}$$

where the heat flux q is expressed in [kW/m^2], mass velocity $\dot{m}$ in [kg/m^{2}s], inner tube diameter d_u in [m] and operating pressure p in [MPa].

The two-phase multiplier for rifled tubes flow the modified Friedel correlation is applied [7]:

$$\Phi_{vo} = A + 6.0x^{1.2}(1-x)^{0.41}\left(\frac{\rho'}{\rho''}\right)\left(\frac{\mu''}{\mu'}\right)^{0.4} \cdot \left(1 - \frac{\mu''}{\mu'}\right) Fr_1^{-0.05} We_1^{-0.033}, \tag{27}$$

where

$$\lambda_i = \frac{8.60 \cdot 10^4}{Re_i^{1.13}} + 0.0220. \tag{28}$$

In [7] the empirical correlation for the critical steam flow quality is developed, which depends not only on the pressure, mass velocity and heat flux, but also on the geometry of the rifled tube

$$x_{cr} = 1 - \left\{\frac{\eta_R^{1.46}}{q} F_B^{0.5} \frac{\dot{m}}{\rho'}\left(\frac{p_{cr} - p}{p_{cr}}\right)^{0.275}\right\}^{-1} \cdot \left\{r\rho''^{0.5}\left[\sigma(\rho' - \rho'')g\right]^{0.25}\right\}^{-1}, \tag{29}$$

where the heat flux q is expressed in [kW/m^2].

Pressure drop due to the local resistance to one-phase flow is calculated as:

$$\left(\frac{dp}{dz}\right)_{loc} = \sum \zeta_{li} \frac{\dot{m}^2}{2\rho'}, \tag{30}$$

while for two-phase flow it is calculated as:

$$\left(\frac{dp}{dz}\right)_{loc} = \sum \zeta'_{li} \frac{\dot{m}^2}{2\rho'}\left[1 + x\left(\frac{\rho'}{\rho''} - 1\right)\right]. \tag{31}$$

Local drag coefficients for one-phase flow (ζ_{li}) and local drag coefficients for steam-water two-phase flow (ζ'_{li}) are determined according to [3].

Gravitational pressure change is calculated as:

$$\left(\frac{dp}{dz}\right)_g = g\rho h. \tag{32}$$

Density of the steam and water mixture in the evaporator tubes is calculated as:

$$\rho_m = \phi\rho'' + (1-\phi)\rho'. \tag{33}$$

The height of the riser tubes up to the boiling boundary (z_h) is calculated from the heat balance as presented in [3]. Namely, if the water boiling starts at the height within the first section (z_I), then the water enthalpy at that level is:

$$\left(h'_{pz}\right)_{iw} = h_d + \frac{Q_{iw}^f (z_h)_{iw}}{\dot{M}_{iw} z_I}, \tag{34}$$

where h_d is water enthalpy in the drum that is equal to the difference of the saturated enthalpy h'_d and enthalpy of water subcooling in the drum ($h_d = h'_d - \Delta h_d$). At the same time, the water enthalpy at the incipient of boiling, in case of the natural circulation is determined by:

$$\left(h'_{pz}\right)_{iw} = h'_d + \frac{\partial h'}{\partial p} g\rho'\left(z_{dc} - (z_{nh})_{iw} - (z_h)_{iw}\right) - \frac{\Delta p_{dc} + (\Delta p_{nh})_{iw} + (\Delta p_h)_{iw}}{g\rho'}. \tag{35}$$

The second term on the right hand side of (35) represents the increase of the saturated water enthalpy due to the hydrostatic pressure increase and its reduction due to the pressure drops up to the boiling boundary. By equalizing (34) and (35) the expression for the heating height up to the boiling boundary is obtained in the following form:

$$(z_h)_{iw} = \frac{\Delta h_d + \frac{\partial h'}{\partial p} g\rho'}{\frac{Q_{iw}^f}{\dot{M}_{iw} z_I} + \frac{\partial h'}{\partial p} g\rho'} \cdot \frac{\left(z_{dc} - (z_{nh})_{iw} - \frac{\Delta p_{dc} + (\Delta p_{nh})_{iw} + (\Delta p_h)_{iw}}{g\rho'}\right)}{\frac{Q_{iw}^f}{\dot{M}_{iw} z_I} + \frac{\partial h'}{\partial p} g\rho'}. \tag{36}$$

In case of forced circulation the heating height is obtained in the same way as (36). Due to the circulation pump operation (36) is modified in the following form:

$$(z_h)_{iw} = \frac{\Delta h_d + \frac{\partial h'}{\partial p} g\rho'}{\frac{Q_{iw}^f}{\dot{M}_{iw} z_I} + \frac{\partial h'}{\partial p} g\rho'} \cdot$$

$$\cdot \frac{\left(\frac{\Delta p_p - \Delta p_{dc} - (\Delta p_{nh})_{iw} - (\Delta p_h)_{iw}}{g\rho'} \right)}{\frac{Q_{iw}^f}{\dot{M}_{iw} z_I} + \frac{\partial h'}{\partial p} g\rho'} . \quad (37)$$

The heating height is calculated iteratively, since it is necessary first to guess this height in order to calculate the pressure drop along that section $(\Delta p_h)_{iw}$. Hence, the evaporating height in the furnace is calculated as $\left(z_{ev}^f\right)_{iw} = z_I - (z_h)_{iw}$, while in the boiler convective section it is $z_{ev}^{co} = z_{II}$, where z_I is the height of the furnace section and z_{II} is the height of the convective section of the evaporator (Fig. 3).

Pressure drop due to the two-phase flow acceleration in the evaporator with natural or forced circulation has negligible value in regard to other pressure changes, hence, it is assumed

$$\left(\frac{dp}{dz}\right)_a \cong 0 . \quad (38)$$

The control heat calculation of the boiler and the presented thermal-hydraulic calculation of the evaporator are performed by the developed FORTRAN computer code.

4. RESULTS AND DISCUSSION FOR THE FORCED CIRCULATION LOOP

In order to obtain the absorbed heat loads on the corresponding evaporating walls in the boiler furnace (q_{iw}^f, Q_{iw}^f) and in the convective section (q_{iw}^{co}, Q_{iw}^{co}), it is first necessary to perform the heat calculation of the steam boiler. Results of the calculation for four boiler loads ($\bar{D} = 1.0$; $\bar{D} = 0.8$; $\bar{D} = 0.6$ and $\bar{D} = 0.4$) are presented in [10].

Thermal-hydraulic calculations of the circulation loop are conducted for loads of 100 %, 80 %, 60 % and 40 % of the nominal full load. Results of the forced circulation hydraulics are presented in Figure 5 for both rifled tubes (which corresponds to the boiler design condition) and for the smooth tubes (analyzed comparative design) in the furnace. It is shown that the decrease of the boiler load, and at the same time the decrease of the evaporator operating pressure, leads to the increase of the fluid mass flow rate and the slight increase of the pressure drop in the evaporator.

In order to better envisage the changes of corresponding two-phase flow parameters, depending on the boiler load ($\bar{D}$), the changes of some parameters are depicted in Figure 6. The increase of the mass flow rate ($\dot{M}_{be}$), and corresponding mass velocity ($\dot{m}$), with the boiler load decrease, is the consequence of the higher water density at the lower pressure. If the circulation velocity is determined from the mass velocity ($w_o = \dot{m}/\rho'$), a slight change of its dependence upon the boiler load is observed. Also, the average values of void fractions in the furnace (ϕ_{av}^f) and convective section (ϕ_{av}^{co}) of the evaporator show a weak dependence upon the boiler load, which indicates that the sliding pressure dependence on the boiler load change is adequatly chosen. The decrease of the steam void fraction at the evaporator outlet with the decrease of the boiler load is caused by two effects. One is the decrease of the heat load, and the other is the thermodynamic effect of the increase of the latent heat of evaporation with a pressure decrease.

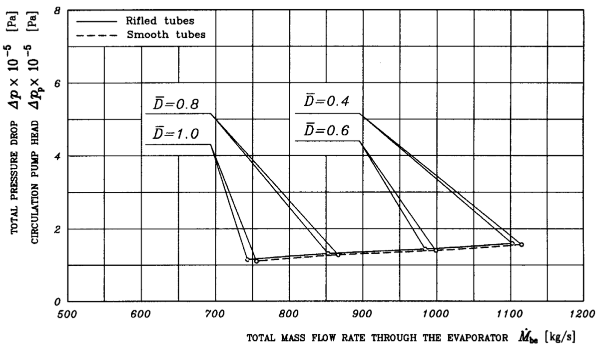


Figure 5. Hydraulic characteristic of the evaporator with the forced circulation in cases of rifled and smooth tubes

The circulation number of evaporator (κ) increases with the boiler load decrease, and it is equal to the reciprocal value of the steam quality at the evaporator outlet (x_{out}). On the basis of the presented dependence of the circulation number on the steam boiler load in Figure 6, and with the aim at the fast prediction of the evaporator thermal-hydraulic parameters in the operation, the following relation is derived:

$$\kappa = 26.521 - 53.703D + 41.075D^2 - 11.292D^3 , \quad (39)$$

where the fluid total mass flow rate through the evaporator is calculated as:

$$\dot{M}_{be} = \kappa D , \quad (40)$$

and the steam quality at the evaporator outlet is predicted by:

$$x_{out} = \frac{1}{\kappa} , \quad (41)$$

where D is dry steam production in the evaporator and it is known for defined steam boiler load.

It is also interesting to present the calculated hydraulic characteristics of the circulation loop for different boiler loads on the pump operating characteristic that is provided by the pump manufacturer (Fig. 7). It is shown that the fluid volumetric flow through the evaporator for all boiler loads are very close, and the coefficient of the pump efficiency is nearly maximal in the whole operating range and has a value of 80 %; hence, it can be concluded that the adequate circulation pump is chosen.

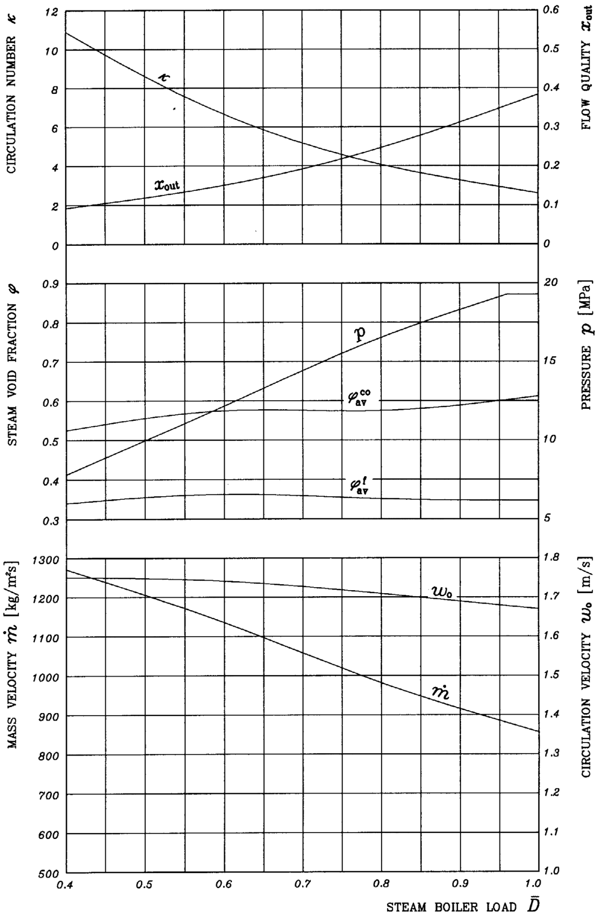


Figure 6. Calculated thermal-hydraulic parameters for the evaporator forced circulation

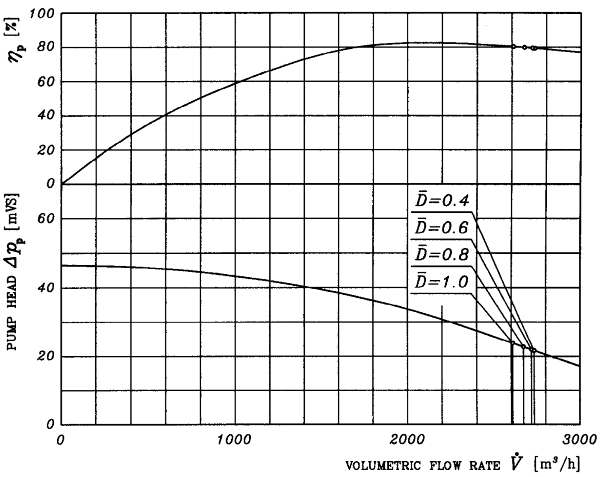


Figure 7. Evaporator operating points dependence on the boiler load

In order to compare the evaporator hydraulic characteristics in cases of furnace evaporating tubes with rifled and smooth inner surfaces, the same calculations are performed with smooth furnace tubes for above stated boiler loads. Obtained results are depicted in Figure 5 with dotted line. Comparing the results it is shown that the pressure drop in case with rifled tubes in the furnace is at maximum 3.7 % higher than in case with smooth tubes for the nominal boiler load. At first sight so small difference is not expected, since the increase of frictional pressure drop in the rifled tube, compared with the smooth tube, is approximately 90 % higher in case of two-phase flow with 30 % quality and the same other flow parameters [7]. But, observing the whole circulation loop, the length of the rifled tubes is only 25 % of the total circulation loop length, and the existence of the thread ribs on the tube inner surface leads to the adjustment of the fluid mass flow rate; for nominal boiler load the flow rate is reduced by 1.7 % and for the minimal load 1.1 %. Hence, due to the friction pressure drop dependence on the velocity square, and relatively small length of the rifled tubes, the increase of frictional pressure drop in the circulation loop is practically negligible.

5. THE INFLUENCE OF RIFLED TUBES ON THE INCREASE OF THE MARGIN TOWARDS BURN-OUT

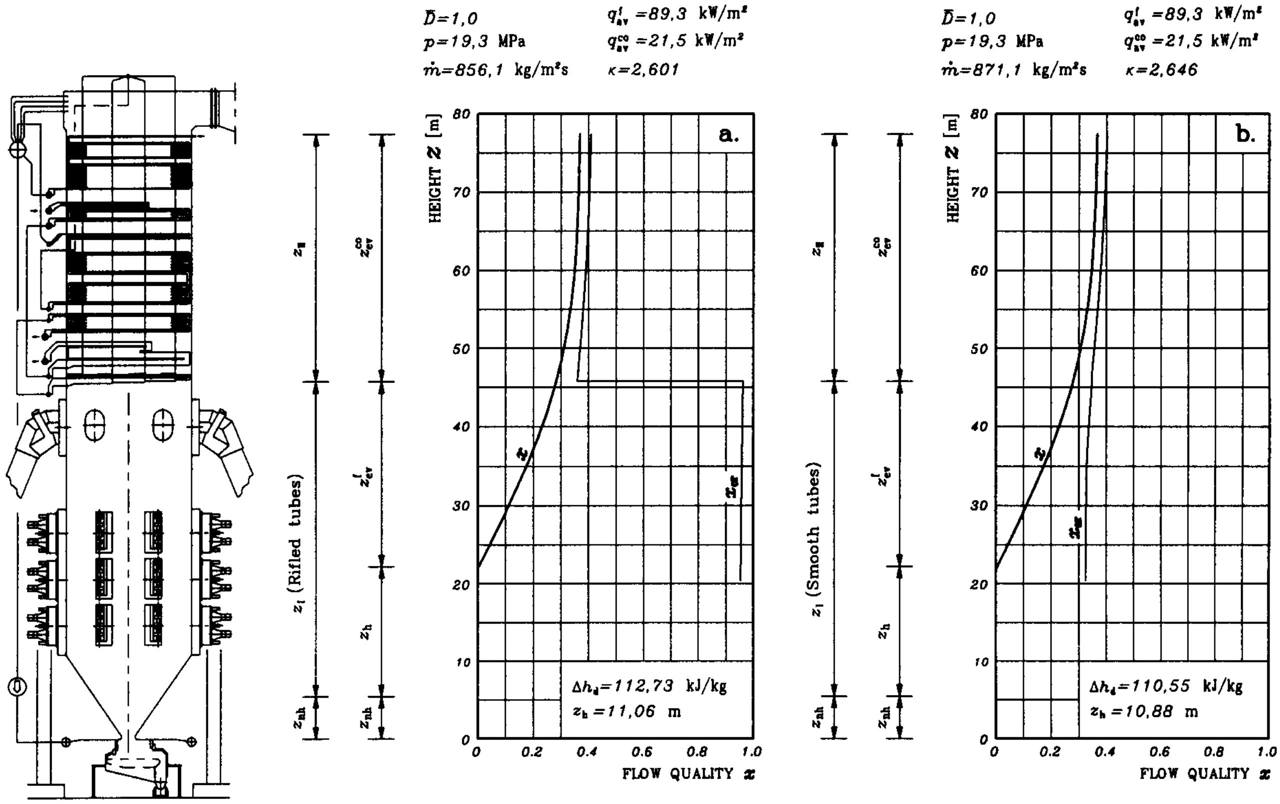
Critical steam flow quality is a characteristic of the burnout (or critical heat transfer phenomenon) when the dry-out of the liquid film on the tube wall occurs, while water drops are still present in the steam flow. As stated, the burnout is characterized with the rapid drop of the heat transfer coefficient and tube wall temperature increase. In zones with high temperatures of the combustion products in the furnace, the burnout leads to tube wall thermo-mechanical damage. Hence, a thermal-hydraulic design of the steam boiler must provide that the flow quality values along the evaporator are lower than the critical ones.

In order to increase the margin towards the burnout occurrence, the rifled tubes with the thread ribs on the inner side are applied in the design of the steam boiler of the TPP "Kolubara B". The advantage of the rifled tubes is more pronounced in case of higher pressures and higher heat loads.

Figure 8 shows the change of the flow quality and its critical value along the evaporator height under the forced circulation and uniform heat load between evaporator walls. It is shown that the quality does not exceed the critical values, whereas the margin towards the critical values is much higher in the rifled than in smooth tubes.

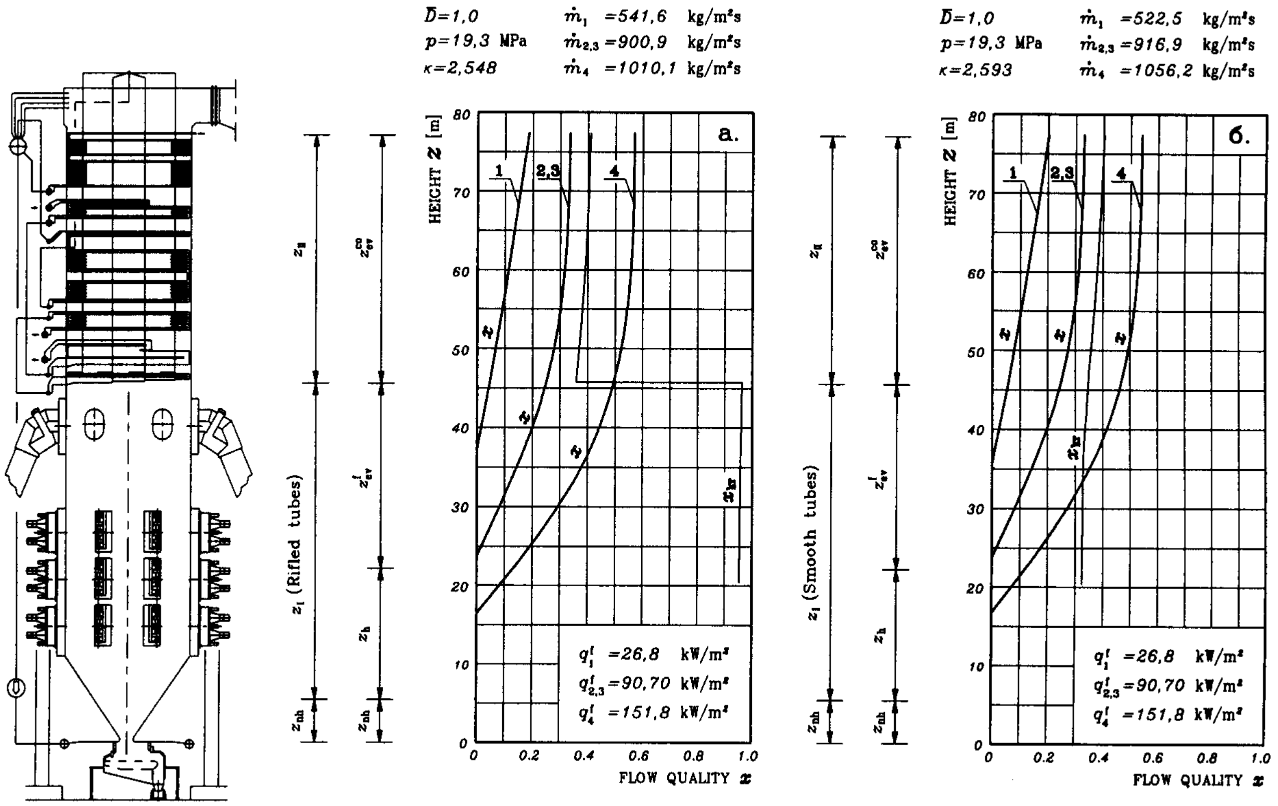
Presented results are obtained under the uniform heat load distribution between the evaporator walls. In real operating conditions the non-uniformity of the furnace heat loads exists and it is more or less exaggerated. The non-uniformity of the heat load exists due to some burners malfunction or switch off, or because of a different fouling of the walls. Because of that, some tubes or all tubes at some furnace wall receive different heat load in real operating conditions than those average value determined by the boiler heat calculation.

In order to get insight into the influence of the heat load non-uniformity on the thermal-hydraulic parameters of the steam and water two-phase flow, the thermal-hydraulic calculation is performed under the assumption that the front wall receives 70 % lower heat load than average, the rear wall receives 70 % higher heat load, and the load of the lateral walls is symmetrical and equal to the average load. The change of the flow quality along the evaporator height is shown in Figure 9 for both rifled and smooth tubes in the evaporator furnace. Also, the heat fluxes and the mass velocities in furnace walls are shown. The greatest quality occurs in the rear wall since it absorbs the greatest heat load.



a) Rifled tubes in the evaporator; b) Smooth tubes in the evaporator

Figure 8. Change of the flow quality along the evaporator height under the forced circulation and uniform heat load between the evaporator walls



a) Rifled tubes in the evaporator; b) Smooth tubes in the evaporator (1 – front wall, 2,3 – lateral walls, 4 – rear wall)

Figure 9. Change of the flow quality along the evaporator height under the forced circulation and non-uniform heat load between the evaporator walls

The application of the rifled tubes provides the satisfactory protection against the burnout for the evaporator section in the furnace (Fig. 9a). In the rear evaporator wall in the convective gas channel the quality exceeds its critical value. But, in that section the heat fluxes have low values and the tubes burnout could not occur.

In case of the smooth tubes application in the furnace section, the critical steam quality is achieved in zones of high temperatures, which would lead to tubes burnout. Hence, this analysis shows the necessity of the application of the rifled tubes in the furnace section of the evaporator in order to provide evaporators safety against the burnout.

6. CONCLUSIONS

The thermal-hydraulic model of the working fluid flow and evaporation within the forced circulation loop of the large steam power plant boiler is developed. The model is based on both integral and local mass, momentum and energy balances of one-phase and two-phase fluid flow. The integral balances define the distribution of the working fluid flow rate and pressure between several downcomer and riser sections. The local balances define the thermal-hydraulic parameters change along tubes of observed downcomer and riser sections. Developed model is applied to the simulation and analyses of the boiler forced circulation loop thermal-hydraulics. The applied geometry and operating conditions correspond to the steam boiler of the Thermal Power Plant “Kolubara B” under construction. Two-phase flow and steam generation in evaporating tubes in the furnace zone are modeled and analyzed for both rifled and smooth inner tubes’ surfaces. The influence of the rifled tubes on the enhancement of the thermal margin against the critical heat transfer occurrence and on the circulation loop hydraulics due to the increased flow resistance is determined. Presented results show the main thermal and hydraulic parameters in the evaporating tubes for the whole range of designed operating conditions from 40 % to 100 % of the full boiler load. The change of the flow quality and critical quality, under which the critical heat transfer occurs, are compared for rifled and smooth evaporating tubes, as well as for the uniform and non-uniform heat load distribution between evaporator walls. The presented results are a necessary support for the steam boiler design. The major findings are as follows.

(a) According to the simulation results in Figure 6, the circulation velocity and steam void fraction practically do not change with the boiler load, which confirms that the sliding pressure dependence on the boiler load is adequately defined.

(b) Calculated hydraulic characteristics of forced circulation loops with rifled and smooth tubes in the evaporating tubes in the furnace section are analyzed. The results show that in case of the smooth tubes the circulation mass flow rate is increased by only 1.7 % under the 100 % load and 1.1 % under the 40 % load compared to the case with the rifled tubes. Hence, it can be concluded that the rifled tubes practically do not influence the evaporator’s hydraulic characteristics and they do not increase the energy consumption of the circulation pumps.

(c) The conducted analyses show that the rifled tubes prevent the critical heat transfer conditions in the furnace zone under all operating conditions and under supposed non-uniform heat load distribution between the evaporator’s walls. In case of non-uniform heat load the critical heat flux can be reached in the convective zone of the most loaded evaporating tubes. But, in this zone the heat fluxes are lower compared to the furnace zone and they can not lead to the tubes burnout.

(d) In case of the smooth tubes application in the evaporator furnace zone, Figure 9b shows that the critical heat transfer conditions can be reached in the furnace zone under non-uniform heat load distribution between the evaporating walls. This fact justifies the need for the rifled tubes implementation in the furnace zone of the evaporator.

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NOMENCLATURE

C	slip coefficient
D	dry steam production [kgs^{-1}]
d_e	equivalent tube diameter [m]
d_u	inner diameter of smooth tubes [m]
g	gravity [ms^{-2}]
z	tubes height [m]
h	specific enthalpy [kJkg^{-1}]
Δh_d	water subcooling in the drum [kJkg^{-1}]
l_i	length of the circulation loop section [m]
$\dot{m}$	mass velocity $\dot{m} = w_o \rho'$ [kgm^{-2}s^{-1}]
$\dot{M}$	mass flow rate [kgs^{-1}]
p	pressure [MPa]
p_{cr}	critical pressure [MPa]
Δp	total pressure change [Pa]
Δp_p	circulation pump pressure head [Pa]
q	heat flux [kWm^{-2}]
Q	absorbed heat load [kW]
r	latent heat of evaporation [kJkg^{-1}]
w	velocity [ms^{-1}]
w_o	circulation velocity [ms^{-1}]
x	quality [kgkg^{-1}]
x_{cr}	critical quality [kgkg^{-1}]

Greek symbols

β	steam flow volume fraction
φ	steam void fraction
Φ_{vo}	two-phase flow multiplier
κ	circulation number $\kappa = \dot{M} / D$
λ	friction coefficient
μ	dynamic viscosity [Nsm^{-2}]
ρ	density [kgm^{-3}]
σ	surface tension [Nm^{-2}]
ζ_1	local drag coefficient for one-phase flow
ζ'_1	local drag coefficient for two-phase flow

Subscripts

av	average value
be	boiler evaporator
d	boiler drum
dc	downcomer tubes
ddc	discharge section of the downcomer tubes
eco	economizer
ev	evaporating section of the riser tubes
h	heated section of the riser tubes
in	inlet
iw	i-th wall of the evaporator
m	two-phase flow mixture
nh	non-heated section of the riser tubes
out	outlet
sdc	suction sectiPrimeon of the downcomer tubes
1	water
2	steam
I	evaporator in the furnace part
II	evaporator in the conventive part

Superscripts

co	convective section
f	furnace section
’	saturated water
”	saturated steam

ТЕРМОХИДРАУЛИКА ИСПАРИВАЧА ЛОЖИШТА ПАРНОГ КОТЛА СА ПРИНУДНОМ ЦИРКУЛАЦИЈОМ

Драган Туцаковић, Владимир Стевановић, Титослав Живановић

Електропривреда Србије је у току 2005. године наставила радове на изградњи ТЕ Колубара Б снаге 350 MW, који су започети још давне 1989. године. Сам котао је торањске конструкције, а за испаривач котла предвиђено је циркулационо коло са принудним струјањем. Осим тога, екранске цеви у ложишту су ожљебљене са унутрашње стране, што представља јединствени случај у котловима термоелектрана у Србији. Да би се извршиле термохидрауличке анализе овог испаривача урађени су термички прорачуни котла, као и хидраулички прорачуни испаривача за различита радна оптерећења. У овом раду приказана је методологија хидрауличког прорачуна испаривача са принудном циркулацијом. На основу добијених резултата, за споменути котао, извршена је анализа утицаја ожљебљења цеви на повећање хидрауличког отпора у циркулационом колу. С обзиром да ожљебљене цеви обезбеђују сигурнији рад испаривача, одржавајући ниску температуру зида цеви и при високим садржајима паре, вршено је њихово упоређивање са глатким цевима за случајеве равномерног и неравномерног топлотног оптерећења у ложишту.