Work Verification of the Energy Steam Boiler Evaporator in the Power Plant "Kostolac B"

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Within Electric Power Utility of Serbia 1991, a thermal power plant "Kostolac B", power 2 × 350 MW, started. Based on the examination of steam boiler pipe system, it is confirmed that it is in a "bad" shape. Its most jeopardized parts are evaporating heating surfaces. Considering that there are areas within the evaporator that suffered more damage, wherefore the steam boiler is out of production, it is necessary to settle the reasons for those damages. In order to locate the jeopardized areas of the evaporator, we carried out a detailed hydraulic calculation. A thermal calculation of steam boiler was also carried out, considering that the evaporator is also located in the convective heating surfaces part. By settling the most jeopardized evaporator locations – a horizontal part of support tubes, it was enabled to make certain reconstructions during the capital repair, in regard of changing their inclination angle, which would provide safer work of the evaporator, and steam boiler as a whole.

Keywords: *steam boiler, evaporator, hydraulic calculation, thermal calculation, inclination angle of tubes.*

1. INTRODUCTION

Within the thermal power plant "Kostolac B" there are two steam boilers made for combustion of lignite with a lower heating value of 7326.9 kJ/kg. The basic characteristics of a steam boiler are as follows:

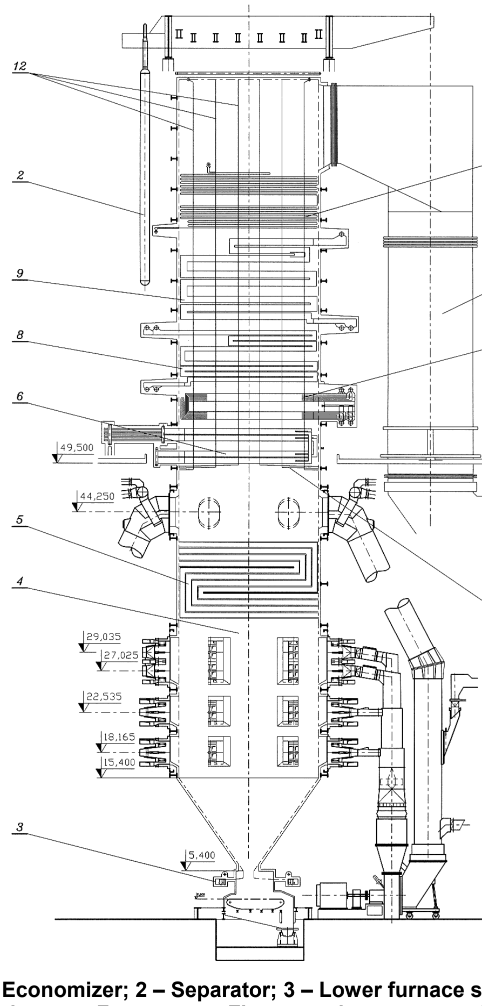
- Main steam mass flow rate, D = 277.8 kg/s;
- Main steam pressure, p_s = 18.6 MPa;
- Main steam temperature, t_s = 540 ºC;
- Reheated steam mass flow rate, D_r = 248.8 kg/s;
- Reheated steam pressure, p_{rs} = 4.375 MPa;
- Reheated steam temperature, t_{rs} = 540 ºC;
- Feed water temperature, t_{nv} = 255 ºC

A simplified steam boiler disposition is shown in Figure 1.

Flue gases made by coal combustion in the furnace (4) are streaming over the third superheater stage (6), the second reheater stage (7), the second superheater stage (8), the first reheater stage (9) and economizer (1), and then turn into the sheet duct (10), in the outlet of which two air preheaters are situated, after which flue gases are released into the atmosphere.

Feed water is guided into the economizer (1). After heating, water gets into the mixing box in which it is mixed with water from the separator (2). This mixture arrives into the circulation pump, which then suppresses it into the ring header (3) that supplies furnace screen (4) having the form of a membrane wall.

One part of the tube screen on the furnace outlet is distributed into support tubes (11,12) on which the superheater and reheater stages and economizer are hanged. The convective gas channel is also screened with tubes.



1 – Economizer; 2 – Separator; 3 – Lower furnace screen headers; 4 – Furnace; 5 – First superheater stage; 6 – Third (output) superheater stage; 7 – Second (output) reheater stage; 8 – Second superheater stage; 9 – First reheater stage; 10 – Sheet duct; 11 – Horizontal part of support tubes; 12 – Vertical part of support tubes duct

Figure. 1. Disposition of the steam boiler in TP Kostolac B

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The mixture of water and steam from furnace screen tubes, support tubes and convective channel screen tubes is lead, over the system of headers, which are located on the top of the convective channel, into the separator (2) where the phase separation is conducted, so water, after mixing with feed water, enters in the described circle, while steam is lead for superheating.

Hydraulic calculations for the evaporator of the examined boiler were made for two work cases: for the designed conditions and for the existing state. Calculation for the designed conditions is based on thermal calculation made for parameters adopted by the boiler manufacturer and for garant fuel. Under the existing conditions, the case is considered in which the heat load disposition by furnace height is changed in relation to the designed, that is, when the highest temperature zone is located in the upper furnace part (risen flame).

With that, calculations for real conditions are made for improved quality fuel, with a lower heating value of 8373.6 kJ/kg.

Thermal and hydraulic steam boiler calculations were based on the Normative method shown in [1-3] which gave the best results for the boilers in Serbia.

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 an example in [4,5]. Experimental data and corresponding correlations of heat transfer coefficient, friction pressure loss coefficient and critical heat flux in smooth evaporating tubes of steam boilers are presented in [6,7].

2. TECHNICAL DESCRIPTION OF THE EVAPORATOR

From the ring header, Ø273 × 28 mm dimensions, water is lead via connecting tubes into inlet headers of the front, rear, left and right evaporator wall. Depending on the inlet header size, each is fed with one, two or tree connecting tubes in which the diaphragms are placed, and whose geometry is given by the manufacturer.

The boiler evaporator is a very complex construction and is divided into panels that function as inlet headers. The front and rear evaporator wall have seven pairs of headers each from which tubes are exiting, forming the adequate panels, while side walls are made of eight headers from which tubes exit, forming eight adequate panels (Fig. 2).

Within the front and rear furnace wall panel, there are tubes used as support tubes, and pipes that make the front and rear convective channel screen wall. Besides, some tubes of both kinds are bent around different openings. Sidewall tubes screen the furnace and convective gas channel and there are two kinds: straight, and bent around the opening. Therefore, the evaporator is made of a large number of tubes that are geometrically and hydraulically different. Because of that, the panels shown are divided into an appropriate number of elements consisting of tubes with the same geometrical characteristics. The front and rear walls are divided into 41 elements each, and left and right wall into 21 elements each.

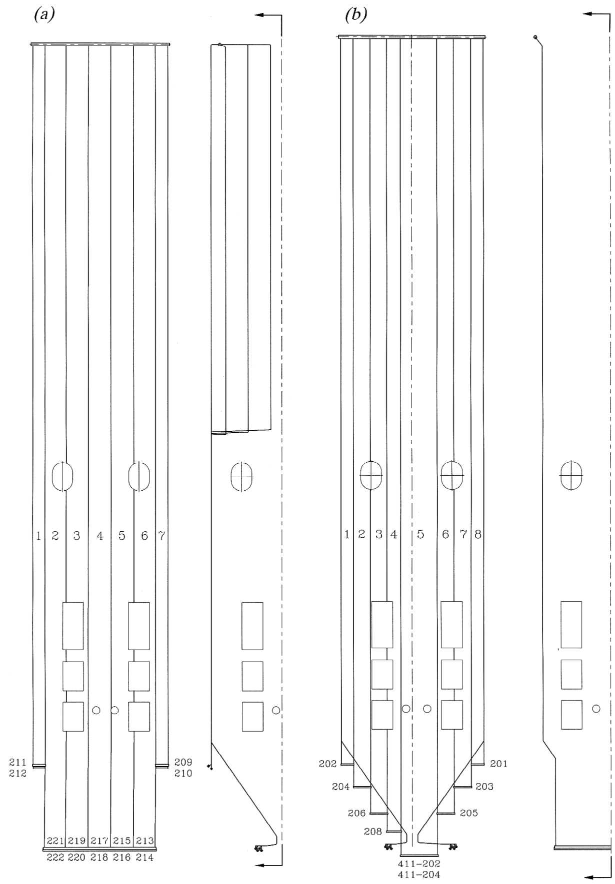


Figure 2. Division of evaporator riser tubes: (a) front and rear wall of the evaporator and (b) left and right side wall of the evaporator

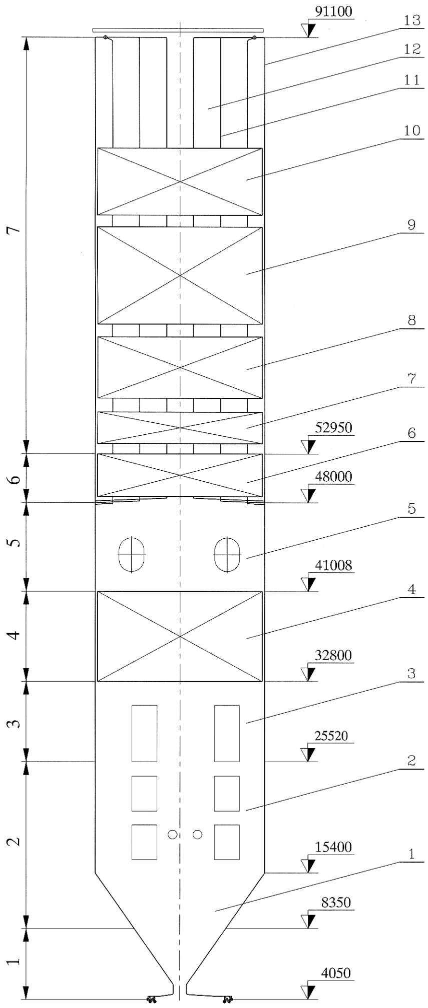
Every element of the evaporator is divided vertically into seven stages depending on heat load distribution, which is gained from thermal calculation of steam boiler and the appropriate geometry (Fig. 3).

3. HYDRAULIC CALCULATION OF THE EVAPORATOR

The principle of the hydraulic calculation of the evaporator is based on definition of the flow rate of every element for the given and adopted conditions (geometrical characteristics and heat load distribution) in that manner that in each of them the pressure drop between the ring header and the separator is the same. The procedure of the hydraulic calculation [2,3] consists of determining the hydraulic characteristics of complex elements of the circulation loops with a grapho-analytical method. In order to solve complex hydraulic calculation procedures, the own code was developed using the iteration method.

End results of the hydraulic calculation for the evaporator are shown in Table 1.

Results given by boiler designer are given in column 4, while the results of calculation made by the Faculty of Mechanical Engineering in Belgrade, and relating to the burning of garant fuel and nominal load, are shown in column 5. Results for the existing work conditions of the boiler with improved fuel and risen flame are shown in columns 6, 7 and 8, for 3 different boiler loads.



1 – Furnace funnel; 2 – Primary furnace zone; 3 – Secondary furnace zone; 4 – First superheater stage; 5 – Furnace recirculation zone; 6 – Third superheater stage; 7 – Second reheater stage; 8 – Second superheater stage; 9 – First reheater stage; 10 – Economizer; 11 – Support tubes; 12 – Guide chamber; 13 – Outlet tube screen

Figure 3. Distribution of riser part of the evaporator to stages in accordance with heat loads

4. CALCULATION RESULTS ANALYSIS

Based on the results shown for hydraulic calculation, it is found that the flow rate value of circulation water is different from that calculated in the same steam boiler working conditions. Water flow rate at the steam separator outlet (Table 1, column 5) is around 20 % greater than the designed one (Table 1, column 4) which happens during the boiler work. The lower amount of heat given to the evaporator resulted in an increase of circulation number and therefore the increase of water amount in the separator. In order to bring the flow down to the designed values, it would be most efficient to set a circulation pump at the appropriate number of rotations and a possibility of its regulation. It is possible, as an alternative, to install the appropriate regulating valve in the downcomer part of the evaporator.

Circulation water flow rate through the evaporator is similar for each furnace working conditions and the steam boiler load.

If the boiler is working with a risen flame, heat given to the evaporator is less, which brings to the decreased steam quality on its outlet. Decrease of the boiler load also causes the decrease of the steam quality on the evaporator outlet for two reasons. First, a lower heat quantity is given to evaporator, and second, there is the increase of the latent heat of evaporation at lower pressures, because the boiler is working with a sliding pressure.

After the shown calculation, all the relevant parameters necessary for the security check of an evaporator, defined by these conditions, are: cavitation effect, absence of circulation halt and direction change, stability of hydraulic characteristics, absence of flow pulsation and normal temperature regime of heating tubes. All of these conditions have been obtained, except the temperature regime of heating tubes for support tubes in a part of the third superheater stage.

Table 1. Results of hydraulic calculation of the evaporator

Name	Nomenclature	Units	Lower heating value of fuel [kJ/kg]				
			7326.9		8373.6		
			Boiler load [%]				
			Designer's results	Designed conditions	Risen flame		
			100	100	100	85	70
1	2	3	4	5	6	7	8
Circulation water flow rate through evaporator	D_c	kg/s	360.000	416.673	405.909	404.457	415.698
Water flow rate at separator outlet	D_w	kg/s	88.000	159.481	166.942	213.192	267.236
Steam flow rate at separator outlet	D_s	kg/s	272.000	257.192	238.967	191.265	148.462
Steam quality in separator	x	kg/kg	0.756	0.617	0.589	0.473	0.357
Circulation number	κ	–	1.324	1.621	1.699	2.114	2.801
Pressure in separator	p_s	bar	202.3	202.3	202.3	183.8	150.0
Temperature in separator	t_s	ºC	367	367	367	359	342
Absorbed heat load in evaporator	Q_i	kW	256,461	240,653	210,699	191,587	171,399

Namely, during a certain combination of regime parameters (pressure, mass velocity, heat flux, steam quantity and pipe geometry) coefficient of heat transfer during the streaming of two-phase flow is quickly reduced due to the contact interruption between the inner tube wall and liquid faze, when there is a fast increase of wall temperature. Under high void conditions, the liquid film depletes on the evaporating tube inner wall, with a possibility of wall dry-out and occurrence of critical heat transfer conditions. The basic parameter of the critical heat transfer is steam quality at the spot of appearance of a wall dry-out and its critical steam quality (CSQ). The methodology for the calculation of the steam quality, as the appropriate increase in the inner wall temperature, is shown in [8] in more details.

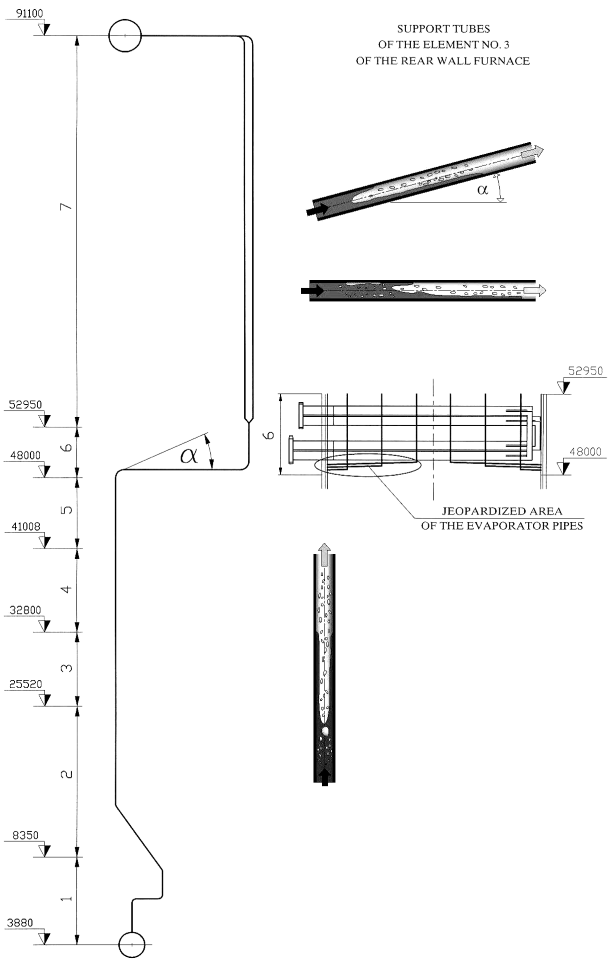


Figure 4. Tube geometry of evaporator element with greatest heat load

For this analysis, we have chosen support tubes with the largest heat load, which are a part of the element no. 3 of the rear wall furnace (Fig. 4). The change of the critical steam quality (CSQ), for the vertical (inclination angle of 90^o) and for the upper (US) and lower (LS) stream of the inclined tubes, of the chosen evaporator element for three different boiler loads during real working conditions, is shown in Fig. 5. Besides, steam quality values on the inlet and outlet of the fifth and sixth stage can be found in the diagram, where it's visible that the critical heat transfer during the boiler load of 70 % takes place in the horizontal part of the support tubes. Namely, with reducing the boiler load, the pressure in the evaporator is also reduced, which leads to the increase of the critical steam quality in the vertical tubes, but reduces the difference of the critical steam quality between the upper and the lower stream of the inclined tubes. Based on the shown diagrams, it is clear that regardless of the boiler load, there is always a danger of stratification in the inclined part of the support tubes, which causes the difference in cooling on the tube volume.

Diagrams also show temperature increase values of the inner (Δt_{wi}) and outer (Δt_{wo}) surface of the tube wall, based on which we can conclude that tube wall temperature increases with load (pressure) reduction at the spot of appearance of a wall dry-out, so that the boiler load of 70 % and appropriate pressure can be in a "danger area".

5. CONCLUSION

Based on the analysis results of the hydraulic calculation of the evaporator of the given steam boiler, we can conclude the next:

Total evaporator flow rate, for the same conditions, is 20 % greater than the flow rate predicted by the designer. Because of that, the circulation number in the evaporator is 1.62, compared to the designed, being 1.32, which resulted in a greater amount of water in the separator.

Boiler works using the fuel of improved quality with a risen flame in furnace, which is suitable to the existing conditions, brings changes to the evaporator's working conditions, in such manner that the critical heat transfer conditions are worsen. That happens more often with reduced boiler load. Variable boiler work regimes lead to deterioration of the listed conditions, and to variable heat loads and tensions in tube material, which results in its fatigue and eventually its damage.

Critical heat transfer conditions are more dangerous for horizontal tubes, which have a temperature difference between the upper and lower stream. Variable regimes, in this case, have even greater effect on fatigue of the materials than they do on vertical tubes. For the given boiler, this is presented in the inclined area of support tubes at the furnace outlet, and on the inlet of the third superheater stage, respectively. This was confirmed in practice – most of the time, boiler's break down was due to damage caused by thermal changes on the support tubes in the considered area. Increase of inclination improves the conditions of the tubes, so it is recommended to increase the inclination angle for this part of tubes. Besides, it would be good to use a material of better quality for this part of support tubes.

ACKNOWLEDGMENT

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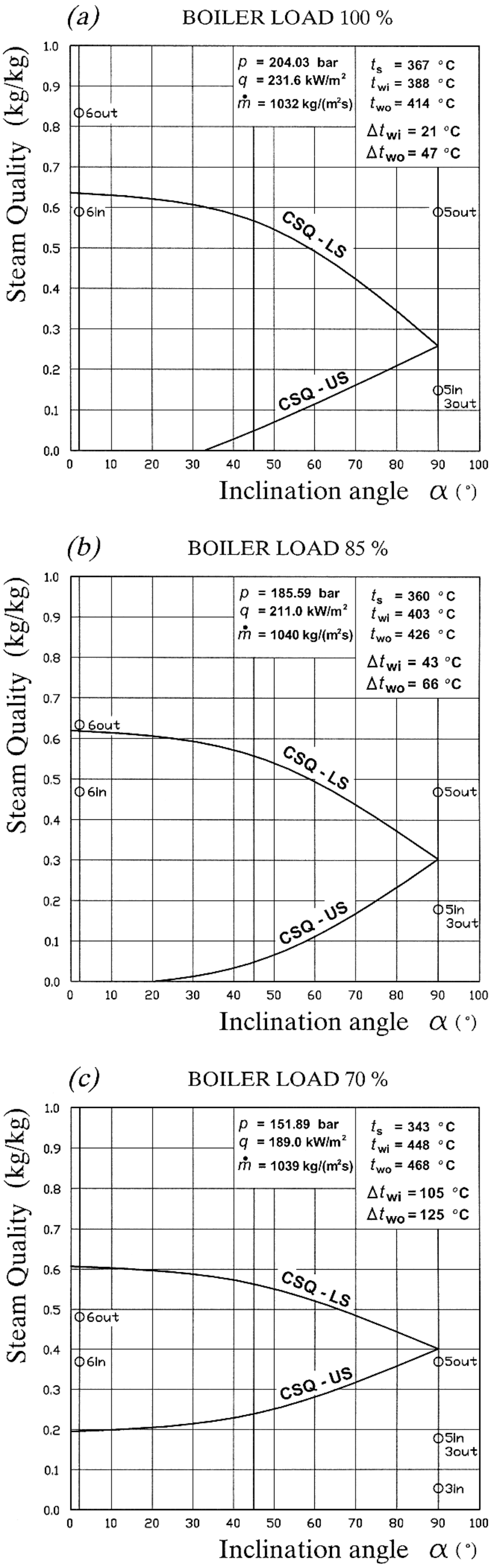

***p* – pressure; *q* – Heat flux; t_s – Saturation temperature; t_{wi} – Inner wall temperature; t_{wo} – Outer wall temperature; $\dot{m}$ – Mass velocity**

Figure 5. Change of the critical steam quality (CSQ) based on the inclination angle of tubes and an appropriate boiler load: (a) results for 100 % boiler load, (b) results for 85 % boiler load and (c) results for 70 % boiler load

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ПРОВЕРА СИГУРНОСТИ РАДА ИСПАРИВАЧА ЕНЕРГЕТСКОГ ПАРНОГ КОТЛА У ТЕ „КОСТОЛАЦ Б“

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У оквиру Електропривреде Србије 1991. године пуштена је у рад Термоелектрана „Костолац Б“ снаге 2 × 350 MW. На основу досадашњих испитивања цевног система парног котла утврђено је да се он налази у „лошем“ стању. Његови најугроженији делови су испаривачке грејне површине. С обзиром да постоје области у испаривачу које су се чешће оштећивале, због чега је парни котао испадао из погона, неопходно је утврдити разлоге због којих је долазило до тих оштећења. Да би се лоцирала угрожена подручја испаривача, спроведен је детаљан хидраулички прорачун. Такође је извршен термички

прорачун парног котла у целини, с обзиром да се испаривач налази и у зони конвективних грејних површина. Утврђивањем најугроженије локације испаривача – хоризонтални део носећих цеви, омогућено је да се при капиталном ремонту изврше одређене реконструкције у смислу промене њиховог угла нагиба, што би у крајњем обезбедило сигурнији рад, како испаривача тако и котла у целини.