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Analysis of fouling thermal resistance of feed-water heaters in steam power plants

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Abstract

In this paper a problem is discussed of thermal degradation of shell-and-tube recuperative feed-water heaters due to heat transfer surface fouling. Application (to an example steam cycle of power plant) of DIAGAR software system intended for the analysing of impact of degradation of a particular heat exchanger on whole recuperative system performance, is presented. On the basis of the systematically performed simulative calculations it was concluded that the degradation of feed-water heaters of the lowest extraction pressure results in an additional load imposed on the heaters of higher extraction pressures, moreover it was estimated that the degradation of the heaters due to presence of sediments of the thermal resistance reaching $5*10^{-4}$ m^2*K/W*, has resulted in the drop of turbine set efficiency by about 0.3%. Such drop depends on a given configuration of degree of degradation of particular feed-water heaters applied in a considered turbine set.*

Keywords: steam power unit, fouling, feedwater heaters

Introduction

Power efficiency improvement of steam power generation systems is today one of the most crucial research problems and concerns both stationary and marine power plants. The problems associated with their operation including thermal degradation of heat power systems resulting from the fouling of their surface, become more and more important.

In this paper is discussed the problem of impact of thermal degradation of heat exchangers working as a low-pressure heat recuperative system on power plant efficiency. An insufficient attention has been paid so far to the problem because it has been assumed that the problem of the fouling sedimented on heat transfer surface and thermal degradation associated with it, would not be very important due to the fact that the working medium of very high purity flows on both sides of such exchangers. As results from the authors' experience which has been gathered so far, the above presented opinion does not comply with real state of the matters, therefore the authors have initiated themselves a systematic research focused on the assessment of degree of thermal degradation of heat exchangers resulting from the fouling of heat transfer surface [1, 2]. In this paper the problem of the assessment of impact of surface fouling of low-pressure feed-water heaters on power plant efficiency, is discussed. The undertaken analysis deals with a supercritical power plant. Such choice is justified by the fact that in the power plant in question number of heaters forming heat recuperative system is the largest.

The below presented analysis is based on calculations which have to be very exact in view of a high precision necessary to determine differences in efficiency of considered cycle configuration variants as well as when various component devices are applied. For the planned scope of the investigations to compare impact of thermal degradation of different feed-water heaters on energy efficiency of a considered cycle, is necessary. For such analysis two quantities are in particular important, namely:

- specific heat consumption defined as the ratio of heat flux delivered to boiler's steam cycle and turbine's internal mechanical power;
- turbine's mechanical power.

The modeled object was thermally calculated and its characteristics were determined. To the calculations DIAGAR software was applied [3,4,5,6]. Some procedures of the program had to be so modernized as to make it possible to calculate the cycles in the range of supercritical steam parameters. Simultaneously, calculation procedures of heat exchangers were so modified as to make it possible to take into account set values of thermal resistance of the fouling sedimented on heat transfer surfaces. Values of thermal resistance were assumed on the basis of the experimental tests on thermal resistance of the fouling accumulated on pipes of one of Polish electric power plants, performed on the authors' original laboratory stand [7,8]. The work was aimed at determination of expected loss of efficiency resulting from fouling sedimented in such heaters. The loss was assessed in relation to the power plant of clean shell-and-tube feed-water heaters.

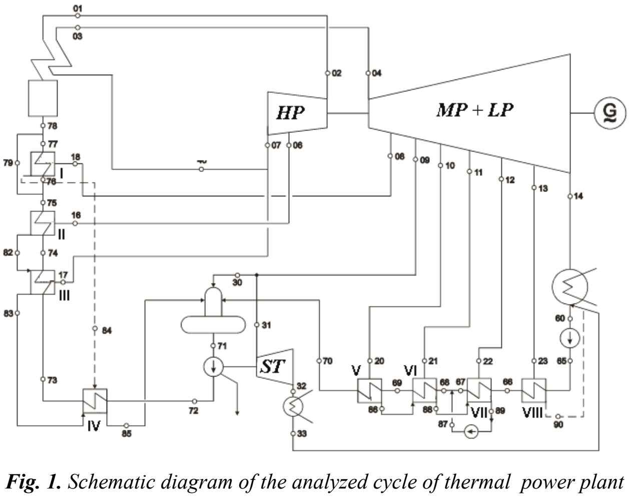

Fig. 1. Schematic diagram of the analyzed cycle of thermal power plant

Analyzed system of power plant

In this paper the influence of thermal degradation of low-pressure feed-water heaters on an example thermal cycle of steam power plant, was analyzed. In order to solve the problem to apply a complete model of thermal cycle is necessary; this is however a very complex problem. Therefore it was decided to use a thermal cycle model of steam power plant, which was at disposal of these authors. Results of the analysis can be considered representative in a wide range of operational parameters of steam power plants, both for marine and land applications.

In Fig. 1 schematic diagram of the considered example thermal cycle of 600 MW power, is presented. The remaining heat-and-flow data decisive of operational characteristics of the cycle are given below:

- Live steam pressure:	28.5 MPa
- Live steam temperature:	600 ^{o}C
- Interstage superheating temperature:	620 ^{o}C
- Condensation pressure:	5 kPa
- Degassing pressure:	1.15 MPa

The cycle is intended for co-operation with:

- electric drive of feed-water pump;

as well as

- turbine drive of feed-water pump.

Moreover from the heat-and-flow data the operational characteristics of the thermal cycle's component devices such as: pressure losses, condensate overcooling, ram temperature rise etc, were calculated.

Thermal degradation of heat exchangers

Thermal degradation of heat exchangers is defined as the lowering of exchanger's heat transfer efficiency, which results from operational factors such as: heat transfer surface fouling, inert gas presence, exclusion of a part of heat transfer surface from operation, or other operational causes. In this work special attention has been paid to the degradation caused by heat transfer surface fouling.

Fouling thermal resistance depends on definite operational conditions of heat exchanger. With a view of different chemical composition of water, e.g. in different electric power stations, sediments of a different thickness, structure and chemical compostion can be met. Moreover, even in a given electric power station thickness and structure of sediments can be different as results from observations of state of heat transfer surfaces. Hence any generalization of only once measured values of fouling thermal resistance for all heat exchangers operating in a given electric power station, is very uncertain.

Influence of thermal degradation resulting from the fouling of heat transfer surfaces as well as inert gas presence necessitates one-dimensional model of heat exchangers to be applied. In such model, elaborated by Butrymowicz and Trela [1,2], a change in heat transfer conditions along water flow through exchanger is taken into account. With a view of necessity of taking into account changes of the conditions in particular runs, such model can be considered quasi-two-dimensional. In the model, impact of inert gas content in condensating steam as well as bundle effect are to be taken into account locally. With a view of inert gas presence to take into account local change of the saturation temperature T_{vs} as well as the phase separation surface temperature T_b, is necessary.

In the analyzed model the following balance differential equations have been used for calculations:

$$\dot{m}_w c_{pw} \frac{dT_w}{dA_o} - k\left(T_w, T_b, T_{to}, \dot{m}_v\right) \cdot \left(T_{vs} - T_w\right) = 0 \qquad (1)$$

in which T_w stands for local temperature of water, and the heat transfer coefficient k is given by the following equation:

$$k = \frac{1}{\dfrac{1}{\alpha_v\left(T_b, T_{to}, \dot{m}_v\right)\beta\left(\dot{m}_v\right)} + R_a\left(T_b, \dot{m}_v\right) + R_{fo} + R_t \dfrac{D_o}{D_m} + \left(\dfrac{1}{\alpha_w\left(T_w\right)} + R_{fi}\right)\dfrac{D_o}{D_i}} \qquad (2)$$

where the following notation is assumed: α_v and α_w – heat transfer coefficients on the side of condensating steam and cooling water, respectively; T_{to} -pipe outer surface temperature; β -bundle coefficient; T_{vs} -steam saturation temperature; R_{fo}, R_{fi}, R_t -fouling thermal resistance of: outer surface, inner surface and pipe material, respectively; D_o, D_i, D_m -pipe outer, inner and logarithmic mean diameter, respectively; R_a -additional thermal resistance due to inert gas presence. Thermal resistance of inert gas layer can be determined by using Berman relation [10] well experimentally confirmed and often applied to numerical calculations [9].

In the case of the analyzed model the following set of equations is considered:

- the differential equation (1);
- and the following algebraic equations:
  - the equation which combines the heat transfer coefficient on the condensating steam side, α_v, the temperatures T_{to}, T_b, and the steam mass flow rate $\dot{m}_v$ [1,2],
  - the equation which expresses the thermal resistance, R_a, resulting from inert gas presence [2],
  - the energy balance equation for an elementary segment of heat exchanger pipe:

$$d\dot{m}_v \left[h_{fg} + c_{pv}\left(T_v - T_{vs}\right)\right] = \dot{m}_w c_{pw}\, dT_w \qquad (3)$$

  - the equation which combines local steam saturation pressure and air content:

$$p_{vs} = p_v \left(1 + 0.622 \frac{\dot{m}_a}{\dot{m}_v}\right)^{-1} \qquad (4)$$

where: $\dot{m}_a$ – air flow mass rate; p_v – steam partial pressure; h_{fg} – specific evaporation enthalpy, c_{pv} – specific heat capacity of superheated steam. On the basis of Eq. (4) local value of the steam saturation temperature T_{vs} can be determined provided distribution of the air flow mass rate $\dot{m}_a$ is known. Dependent variables are the following: the water temperature T_w, the steam mass flow rate $\dot{m}_v$, the phase separation surface temperature T_b, the pipe outer surface temperature Tto as well as the steam saturation temperature T_{vs}. All the quantities are local.

In the case of multi-run heat exchangers, heat transfer conditions on the condensating steam side undergo discrete changes, which is covered by the bundle coefficient β. Hence the solving of the set of equations in question is the most convenient if the differential equation (1) is written in the form of a difference equation. In the case of occurrence of the convective cooling of superheated steam (without condensation) the considered set of equations requires to be extended, that is shown in [11].

The one-dimensional model of heat transfer within recuperative heat exchanger is deemed most suitable for diagnostic analysis purposes because it makes it possible to take into account local effects important from the heat transfer point of view, at a relatively low input of numerical calculations.

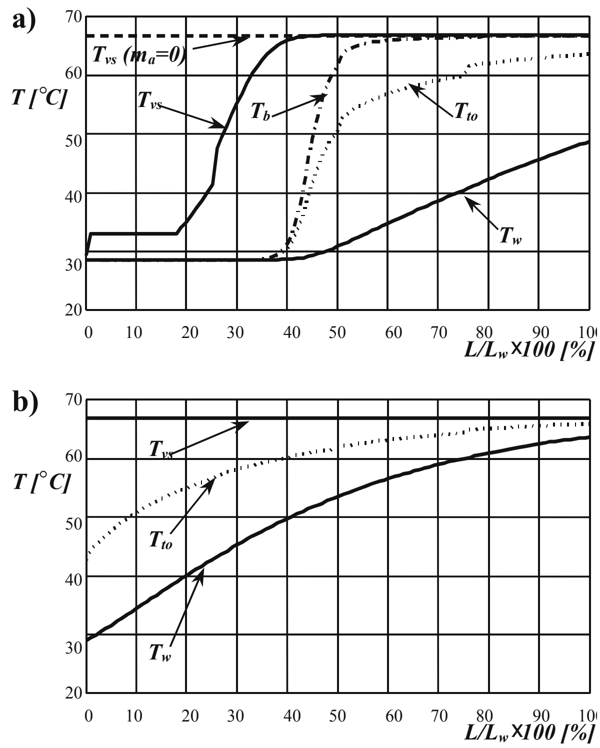

Fig. 2. Temperature distribution in XN1 heat exchanger along water path, Lw – the total water path length. In calculation it was assumed: ***a)*** *R_f =1.5 10^{-4} m^2K/W; = $1*10^{-3}$ kg/s;* ***b)*** *R_f = 0; $\dot{m}_a$ = 0.*

In Fig. 2 are given example results of calculations of a recuperative heat exchanger absorbing steam from the last extraction of LP turbine of 200 MW steam turbine set, i.e. the feed-water heater VIII. The heat exchangers of the lowest extraction pressure are vertically installed in condenser casings, and their heat transfer surface is made by a bundle of horizontal pipes. The basic geometrical quantities of the example feed-water heater VIII for which heat transfer calculations were performed are the following: – number of pipes: 670; – pipe outer diameter: 19 mm; – pipe inner diameter: 17 mm; – pipe material: brass; – active length: 3.642 m; – active breadth: 0.358 m; – number of vertical pipe rows: 52; – number of water runs: 4; – cross-section area of steam inflow to heat exchanger: 2.548 m×0.200 m; – pipe triangular spacing: 44 mm. For numerical calculations the following operational parameters of the heat exchanger were assumed: – water mass flow rate: 63.74 kg/s; -water inlet temperature: 28.51°C; – steam inlet temperature: 93.06 °C; – steam inlet pressure: 0.271 bar. In the calculations it was assumed – by analogy to power condensers – that the steam mass flow discharged from deaerators is associated with the minimum local condensation temperature higher by 2 ÷ 5 K than the water inlet temperature.

From the point of view of diagnostic assessment of heat exchanger operation its thermal degradation is revealed directly by the increasing of the so called ram temperature rise, i.e. difference between saturation temperature and heated water temperature at outlet from the exchanger.

As results from the presented diagrams, the distribution of the water temperature T_w in the exchanger working in air-presence conditions differs greatly from logarithmic one characteristic for operation of a clean exchanger free from presence of air. The air content in condensating steam, amounting to $\dot{m}_a = 1*10^{-3}$ kg/s which is equivalent to the ratio $\left(\dot{m}_a/\dot{m}_v\right)$ = 0.31% assumed for the calculations, results in a significant increase of the ram temperature rise $\delta T_o = T_{vs} - T_{wo}$. For the non-degraded exchanger the increase of the ram temperature rise equal to 3,5 K is obtained, and in the case of a significant degradation due to the heat transfer surface fouling the increase by about 18 K is obtained. On the basis of other analyses [2] it can be stated that the degradation resulting from dirtiness of heat transfer surfaces only causes the increase up to 9.0 K.

Tab. 1. Design data of low-pressure recuperative heat exchangers

Name of device	Values of geometrical characteristics of non-degraded recuperative heat exchangers	
	Heat transfer surface area [m2]	Number of pipes in a heater [pieces]
Recuperative heat exchanger VIII	435	704 – horizontal
Recuperative heat exchanger VII	900	3183 – vertical
Recuperative heat exchanger VI	1053	2874 – vertical
Recuperative heat exchanger V	1060	3183 – vertical

By taking into account the available research results on fouling thermal resistance [1,2,7,8,11], for purposes of this work two characteristic values of fouling thermal resistance can be proposed:

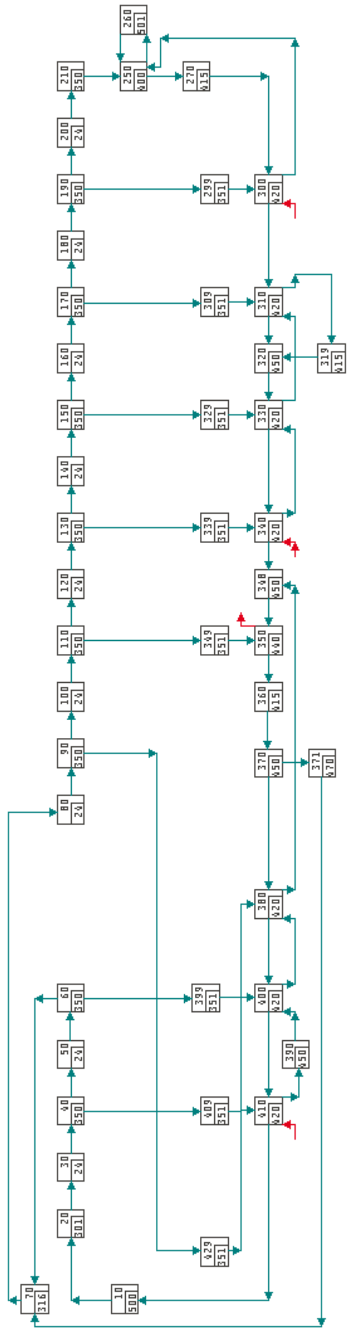

Fig. 3*. Numerical scheme of 600 MW steam turbine unit with electrical drive of feed-water pump .*
Notation: 10 -boiler; 15 – pipeline with valves; 20 – control stage; 30 – group of HP stages I; 40 -extraction I; 50 -group of HP stages II; 60 -extraction II; 70 – interstage superheater; 80 -group of IP/LP stages I; 90 -extraction III; 100 -group of IP/LP stages II; 110 -extraction IV; 120 -group of IP/LP stages III; 130 -extraction V; 140 - group of IP/LP stages IV; 150 -extraction VI; 160 -group of IP/LP stages V; 170 -extraction VII; 180 -group of IP/LP stages VI; 190 -extraction VIII; 200 -group of IP/LP stages VII; 210 – outlet diffuser TG; 250 – main condenser; 260 – condenser cooling; 270 – main condensate pump; 280 – degasifier steam condenser; 299 – pipeline of extraction VIII; 300 – recuperative heat exchanger VIII; 309 -pipeline of extraction VII; 310 -recuperative heat exchanger VII; 319 - condensate pump; 320 – water distribution; 329 -pipeline of extraction VI; 330 -recuperative heat exchanger VI; 339 -pipeline of extraction V; 340 -recuperative heat exchanger V; 348 – pipline; 349 -pipeline of extraction IV; 350 – odgazowywacz; 360 -pompa zasilająca; 370 - water distribution; 371 – injection of water to interstage superheater; 380 -recuperative heat exchanger III; 399 -pipeline of extraction II; 400 -recuperative heat exchanger II; 409 -pipeline of extraction I; 410 -recuperative heat exchanger I; 429 -pipeline of extraction III; 500 – auxiliary turbine; 505 – auxiliary turbine diffuser; 510 – auxiliary condenser; 520 - auxiliary condenser cooling; 530 – condensate auxiliary pump; 540 -pipeline

- $R_f = 2.5 \cdot 10^{-4} [m^2 \cdot K/W]$ -assumed to be typical for heat exchangers;
- $R_f = 5.0 \cdot 10^{-4} [m^2 \cdot K/W]$ -assumed to be relevant to heat exchangers of an elevated degree of degradation.

In this work the two above mentioned values of thermal resistance were used in assuming various degradation degree configurations of low-pressure recuperative feed-water heaters.

Calculations of the power plant thermal cycle

The presented heat-and-flow calculations of the cycle were performed with the use of DIAGAR software [3,4,5,6]. Each of the considered cycle configuration necessitates the same numerical scheme to be applied. The numerical scheme of the cycle for the electric drive of feed-water pump is presented in Fig. 3.

The estimated values of geometrical characteristics of feed-water heaters were based on the characteristics of the heaters installed in 200 MW power units, namely: the same inner and outer diameter and arrangement of pipes. The characteristic data of the heaters are given in Tab. 1.

The collected results of the calculations are graphically presented in Fig. 4. There are also given the applied configurations of thermal resistance levels of the fouling in LP recuperative system. The results indicate the losses revealed by introducing to the calculations the heat transfer surface fouling of feed-water heaters,

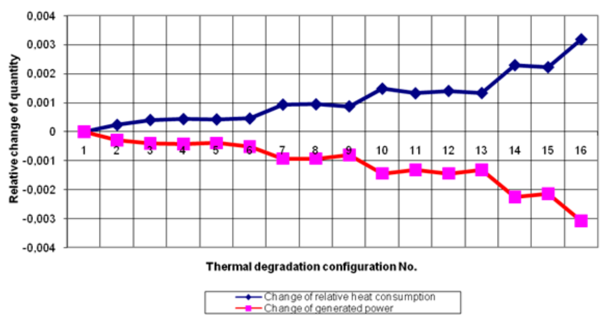

Fig. 4. *Changes of operational parameters of 600 MW power unit at various combinations of degree of degradation of low-pressure heat exchangers. Notation: 1 – clean exchangers; 2-moderately degraded heat exchanger VIII; 3 -moderately degraded heat exchanger VII; 4 -moderately degraded heat exchanger VI; 5 -moderately degraded heat exchanger V; 6 – maximum degraded heat exchanger VIII; 7 -maximum degraded heat exchanger VII; 8 -maximum degraded heat exchanger VI; 9 -maximum degraded heat exchanger V; 10 -moderately degraded heat exchangers VIII – V; 11 -maximum degraded heat exchangers: VIII and VII; 12 -maximum degraded heat exchangers: VIII and VI; 13 -maximum degraded heat exchangers: VIII and V; 14 -maximum degraded heat exchangers: VIII, VII and VI; 15 -maximum degraded heat exchangers: VIII, VII and V; 16 -maximum degraded heat exchangers: VIII – V.*

Tab. 2. *Ram temperature rise of low-pressure recuperative heat exchangers*

Name of device	Values of ram temperature rise [K]		
	$R_f = 0$	$R_f = 2.5\ 10^{-4}$ [m^2K/W]	$R_f = 5.0\ 10^{-4}$ [m^2K/W]
Recuperative heat exchanger VIII	1.62	3.55	5.72
Recuperative heat exchanger VII	0.63	3.82	7.88
Recuperative heat exchanger VI	0.73	4.08	7.92

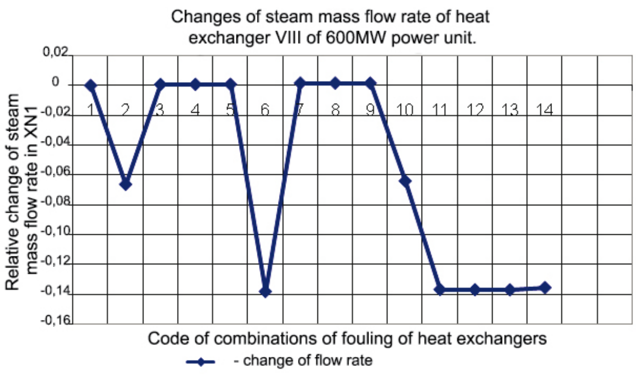

Fig. 5. Changes of steam mass flow rate of heat exchanger VIII at various combinations of degrees of degradation of low-pressure heat exchangers; code of degradation -see Fig. 4

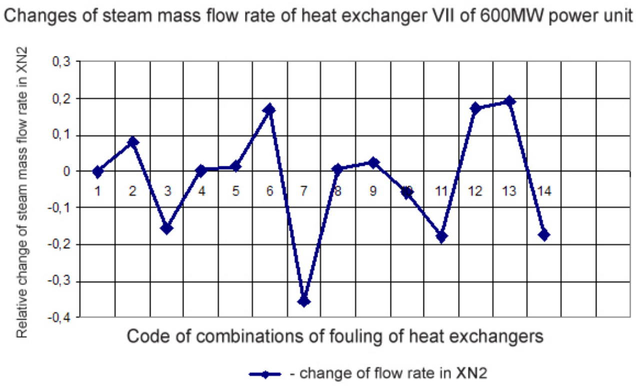

Fig. 6. Changes of steam mass flow rate of heat exchanger VII at various combinations of degrees of degradation of low-pressure heat exchangers; code of degradation -see Fig. 4

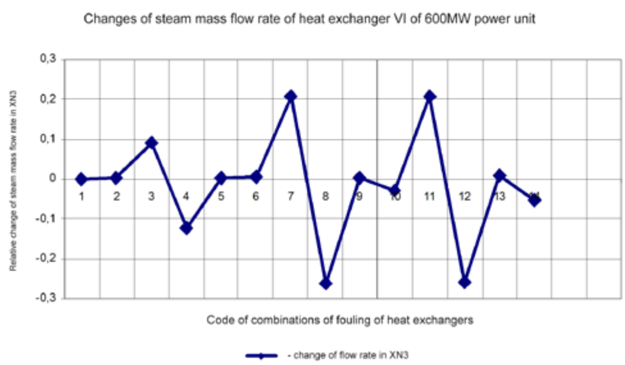

Fig. 7. Changes of steam mass flow rate of heat exchanger VI at various combinations of degrees of degradation of low-pressure heat exchangers; code of degradation -see Fig. 4

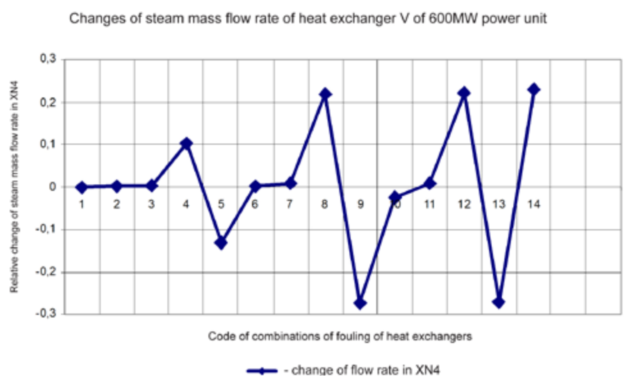

Fig. 8. Changes of steam mass flow rate of heat exchanger VI at various combinations of degrees of degradation of low-pressure heat exchangers; code of degradation -see Fig. 4

expressed in the form of relative changes in turbine-set mechanical power as well as relative changes in specific heat consumption.

As the presented results yield, in the case of the fouling of a single feed-water heater almost the same level of specific heat consumption increase and turbine-set power decrease is obtained for all the feed-water heaters and the typical value of R_{f1} (Case 2, 3, 4 and 5), and for the elevated value of R_{f2} certain diversity of the effects is observed since for the feed-water heater VIII (of the lowest extraction pressure) much lower changes are achieved. However, from the operational point of view the cases are more interesting of different configurations of the fouling of feed-water heaters, having a significant value of the thermal resistance R_{f2} - Case 10 through 13 -where for both the analyzed quantities the same level of changes was obtained. It allows to conclude that in the case of a significant degradation of the feed-water heater of the lowest extraction pressure, degradation of any of the remaining heaters of a higher extraction pressure does not play any important role. The greatest changes are obtained in the case 16 for degradation of three feed-water heaters. The changes in ram temperature rise in particular feed-water heaters are presented in Tab. 2. Along with the increasing of the fouling thermal resistance also the ram temperature rise increases, and in the conditions of the considered geometry especially high values of the ram temperature rise are obtained for feed-water heaters of higher extraction pressures.

In Fig. 5 through 8 the degradation effects in the form of changes in steam mass flow from recuperative extractions to feed-water heaters are presented for various configurations of degradation in the heaters. From the obtained results it can be generally concluded that the fouling of the feed-water heaters of lower extraction pressure values results in the increasing of thermal load imposed on the heaters of higher extraction pressure values because mas flow of steam sucked-in by the heaters then increases. Thermal degradation of a feed-water heater certainly makes that a smaller steam mass flows to it.

Heat transfer surface degradation of feed-water heaters impacts cycle's operational characteristics first of all by increasing the specific heat consumption. The calculation results given in Fig. 4 show that the highest impact on the worsening of cycle's efficiency (i.e. increasing the specific heat consumption) is introduced by the feed-water heater VIII, the first in the row of the recuperative feed-water system. It specially concerns moderate degradation levels, that leads already to necessity of cleaning them. This indicates that special attention should be paid just to cleanless of surface of the feed-water heater. Such conclusion is also confirmed by change in the steam mass flow passing through particular feed-water heaters, which constitutes one of the most important parameters deciding on performace of the exchanger itself.

Summary

In this paper theoretical analysis of impact of thermal degradation of recuperative low-pressure feed-water heaters on change of specific fuel consumption as well as output power of turbine set, has been presented. Degradation of feed-water heaters of the lowest extraction pressure is the most important. The revealed level of the changes should be considered significant.

In this stage of calculations of steam power plant cycles by means of DIAGAR software, carried out for the design load conditions and with taking into account cleanless of heat transfer surfaces of recuperative low-pressure feed-water heaters the following conclusions can be drawn:

1. Increasing the ram temperature rise is a direct effect of presence of sediments on surface of recuperative feed-water heaters (characterized by certain value of the thermal resistance Rf). In the range up to Rf = 0.0005 m2K/W, the resistance results in a multiple increase of ram temperature rise.

2. The maximum degradation of all recuperative low-pressure feed-water heaters (at Rf = 0.0005 m2K/W) may result in the specific heat consumption increase by about 0.3%. It is the value which can be deemed significant for operation of high-output power generation units.

3. Thermal degradation of feed-water heaters of the lowest extraction pressure is accompanied by an additional thermal load imposed on the feed-water heaters of higher extraction pressures. Hence in the feed-water heaters of higher extraction pressures greater relative changes in sucked-in steam mass flow rate than in the heaters of lower extraction pressures, are observed. The relative changes in mass flow rate of steam sucked-in by the heater of the lowest extraction pressure do not exceed 16%, whereas the changes for the heater of the highest extraction pressure are contained in the range up to about 30%. The above described effects directly result in a drop of turbine-set efficiency, hence in a relevant increase of specific fuel consumption.

Therefore it is necessary to fil up an important information gap in the area of real thermal resistance values resulting from the fouling of recuperative low-pressure feed-water heaters used in steam power plants as the available data in question have not a comprehensive character.

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