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EVALUATION OF THE THERMAL DEGRADATION PROCESS WITHIN THE STEAM POWER PLANTS HEAT EXCHANGERS

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Abstract

The consequence of the deposits presence on the heat transfer surface of the following heat exchangers: shell-and-tube condensers, the regenerative feed water exchangers, are commonly the loss of heat exchanger capacity, owing to the high value of heat conductive resistance of fouling. The process is more often defined as the thermal (heat) degradation of a given heat transfer device. The symptoms of heat degradation are usually defined as the difference between the values of thermal-flow parameters for the current and the reference state. Moreover, this process always entails an increase in the cost of energy conversion, leading to increasing the emission output of greenhouse gases consequently increasing the environmental degradation. It is worth mentioning that deposits settled on the waterside of heat transfer surface could also initiate the process of tubes obliteration. This phenomenon is characteristic for condensers cooled by seawater in particular. Reducing internal diameter of any single heat exchanger pipe by the deposits cause the rise in flow resistance and also reduce the condenser cooling water pump capacity. Ultimately, it leads to the reduction in water flow rate, resulting in an additional increase in the resistance of heat transfer. Furthermore, reducing the flow rate of cooling water causes the enhancement of the fouling settling rate. The paper describes the above-mentioned phenomena and presents the quantitative determinants of the thermal degradation description for heat exchangers based on the results of the author's own experimental research.

Keywords: *steam power plants, fouling of heat transfer surface, thermal degradation determinants*

1. Introduction

The presence of deposits on heat transfer surfaces of the heat exchangers (e.g. condensers, heat recovery exchangers) is associated with the loss of their thermal performance, which is increasingly being regarded as thermal or heat degradation of a given heat exchanger. Among other things, the thermal degradation causes the rise of the terminal temperature difference value and is the reason of condenser vacuum deterioration [5, 7, 9, 20]. The process of the heat exchanger thermal degradation finally leads to a reduction in the efficiency of a given power unit. This always entails an increase in the cost of energy conversion, and consequently leading to increased environmental degradation, for instance, through higher than necessary energy consumption [7, 18].

The thermal resistance values of deposits, which gather on the heat transfer surfaces of the steam power plants heat exchangers, presented in the literature, have varied over in a wide range. For example, according to TEMA standards (Tubular Exchanger Manufacturers Association), the values of the specific thermal resistances of fouling r_f within the steam power plants heat exchangers are changed in the range of $8.8\cdot10^{-5}\div10\cdot10^{-4}$ m^{2}K/W [6, 13, 17, 19]. Besides, literature of the subject teaches that the value of the thermal resistance of fouling is strongly influenced by the following factors: the type of dissolved salts in water, the material of construction and condition of heat transfer surface, kind of flow, the temperature and speed of the working media. For example, the lower the temperature of pipe wall and the higher the speed water flow are, the lower the tendency of the wall to gather the deposits [1, 2, 13].

It should be added that in the case of steam power plants condensers, the presence of inert gases in their steam space has a similar effect as the presence of fouling on their heat transfer surfaces (Cunningham research [8]). Furthermore, Butrymowicz numerical studies [5] have shown that the effect of non-condensable gases is particularly acute for the steam power plants condensers, featured by high values of the overall heat-transfer coefficient, e.g. marine steam condensers. Moreover, these studies have led to a very important conclusion i.e. the higher the value of a heat exchanger heat transfer coefficient, the more sensitive heat exchanger is to fouling presence on its heat transfer surface. The above-mentioned conclusion has a direct impact on the operation of steam power plants heat exchangers and on marine heat exchangers in particular. Taking into account the design guidelines, including minimizing the dimensions of heat exchangers in the marine steam power plants, these heat exchangers are characterized by relatively high values of heat transfer coefficients. Therefore, compared to the results arising from the research of the author [5], the problem of thermal degradation heat exchange apparatus in steam plants is very vital for the operation of these exchangers, because of their high values of heat transfer coefficients [12, 13, 15, 16].

It is worth mentioning that the fouling gathered inside the heat exchanger pipes (the waterside) initiate the process of pipes obliteration. This phenomenon is particularly characteristic for the seawater-cooled condensers [19]. By reducing the pipe light, the deposits cause an increase in flow resistance while simultaneously reducing the condenser cooling efficiency. Ultimately, this leads to a reduction in water flow rate, resulting in an additional increase in the resistance of heat transfer. Thus, it can be concluded that the presence of fouling on the surface of the heat exchange not only leads to thermal degradation, but more broadly to *the thermal-and-flow degradation* of a given heat exchanger.

2. Thermal degradation of heat exchangers

Heat transport Q_C through the heat transfer surface free of fouling A_C with a predetermined logarithmic temperature difference $LMTD_C$ is described by the following relation (C as the *clean*):

$$\dot{Q}_C = \frac{1}{r_{k,C}} \cdot A_C \cdot LMTD_C . \tag{1}$$

Heat output Q_F at $LMTD_F$ of the heat exchanger in which heat transfer surface A_F is covered with fouling of thermal resistance r_f is determined in analogy to (1) (F as the *fouled*):

$$\dot{Q}_F = \frac{1}{r_{k,F}} \cdot A_F \cdot LMTD_F , \tag{2}$$

where:
$r_{k,C}$ – overall thermal specific resistance of heat exchanger without fouling [m^2K/W],
$r_{k,F}$ – overall thermal specific resistance of heat exchanger with fouling [m^2K/W].

Given that:

$$r_{k,F} = r_{k,C} + r_f \;\wedge\; r_f > 0 \;\Rightarrow\; \frac{1}{r_{k,C} + r_f} < \frac{1}{r_{k,C}} , \tag{3}$$

and also considering the heat exchanger operating condition at a constant $LMTD$ and slightly different heat exchange surfaces, i.e.:

$$LMTD_C = LMTD_F \;\wedge\; A_C \cong A_F , \tag{4}$$

with regard to formulas (1) and (2), the final conclusion was brought to the inequality:

$$\dot{Q}_F < \dot{Q}_C . \tag{5}$$

The inequality (5) indicates that the loss of thermal efficiency of heat exchanger due to the presence of deposits on its heat transfer surface, at the same time ensures the phenomenon of the heat exchanger thermal degradation.

2.1. Thermal degradation determinants

1. *Cleanliness factor CF* of a heat exchanger – is expressed as a ratio of the overall heat-transfer coefficients k_F and k_C or as a ratio of the overall thermal specific resistances $r_{k,C}$ i $r_{k,F}$:

$$CF = \frac{k_F}{k_C} = \frac{r_{k,C}}{r_{k,F}}, \tag{6}$$

During heat exchangers operation, factor *CF* should not be less than 0.65 [1].

2. *Loss of thermal power LP* factor of a heat exchanger – determines the relative loss of its thermal output between its two operational states, i.e. the heat exchanger in the state of fouled heat transfer surface and the heat exchanger in the state of "clean" heat transfer surface and is as follows:

$$LP = \left(1 - \frac{\dot{Q}_F}{\dot{Q}_C}\right) \cdot 100\%. \tag{7}$$

Taking into consideration the heat exchanger operation conditions expressed by equalities in (4), the *LP* factor can be redefined to formula designating the heat exchanger susceptibility to fouling:

$$LP = \frac{k_C \cdot r_f}{1 + k_C \cdot r_f} \cdot 100\%. \tag{8}$$

For thermal resistances values $r_f \in \langle 1E\text{-}05\text{-}5E\text{-}04\ m^2K/W \rangle$, the LP factor reaches values in range of 4-20% when an overall heat-transfer coefficient $k_C \leq 500\ W/m^2K$, whereas the LP factor achieves values greater than 45% when an overall heat-transfer coefficient $k_C \geq 10\,500\ W/m^2K$ [4].

3. Research methodology

1. The specific thermal resistance of fouling r_f was determined by the differential method of designating the fouling resistance. In this method, the fouling thermal resistance is calculated as the difference between total specific thermal resistance of heat exchanger with fouled heat transfer surface $r_{k,F}$ and total specific thermal resistance of heat exchanger with heat transfer surface free of deposits $r_{k,C}$ (subscript "F" – with fouling, subscript "C" – without fouling):

$$r_f = r_{k,F} - r_{k,C} = \frac{1}{k_F} - \frac{1}{k_C}. \tag{9}$$

2. The heat flux received by the water flowing through the fouled tube (with deposits) $\dot{Q}_{w,F}$:

$$\dot{Q}_{w,F} = \dot{m}_{w,F} \cdot c_{p,w}{}_{twi,F}^{two,F} \cdot (t_{wo,F} - t_{wi,F}). \tag{10}$$

3. The heat flux received by the water flowing through the "clean" tube (without deposits) $\dot{Q}_{w,C}$:

$$\dot{Q}_{w,C} = \dot{m}_{w,C} \cdot c_{p,w}{}_{twi,C}^{two,C} \cdot (t_{wo,C} - t_{wi,C}), \tag{11}$$

where:
$\dot{m}_w$ – condenser water cooling mass flow [kg/h],
$c_{p,w}{}_{twi,C}^{two,C}$ – mean specific heat of water in the range of temperatures from t_{wi} to t_{wo} [J/(kg·K)],
t_{wi} – inlet temperature to the tube [°C],
t_{wo} – outlet temperature from the tube [°C].

4. The maximum absolute systematic uncertainty of measurement Δr_f for the fouling specific thermal resistance r_f, has been defined by means of Gauss's law of measuring uncertainties propagation [14]. Taking into account relation (9), the Δr_f value was calculated as follows:

$$\Delta r_f = \sqrt{\left(\frac{\partial r_f}{\partial k_F}\cdot \Delta k_F\right)^2 + \left(\frac{\partial r_f}{\partial k_C}\cdot \Delta k_C\right)^2} = \sqrt{\left(-\frac{1}{k_F^2}\cdot \Delta k_F\right)^2 + \left(\frac{1}{k_C^2}\cdot \Delta k_C\right)^2}\ . \tag{12}$$

Experimental tests were carried out for three heat exchanger tubes derived from different domestic steam power plants. Measurement of fouling specific heat resistance was taken simultaneously for two pipes, i.e. for the tested pipe with the fouled heat transfer surface and for pipe without fouling regarded as a reference pipe. The length of each pipes was 1 m. Research materials were labelled by the following Roman numerals: II (d_o = 17.5 mm, d_i = 13.5 mm), VII (d_o = 17.5 mm, d_i = 13.5 mm) and XII (d_o = 16.0 mm, d_i = 12.0 mm). The pipes II and VII have originated from low-pressure heat recovery exchangers but the XII one has originated from high-pressure heat recovery exchanger. Fig. 1 shows photos of research materials II, VII and XII.

Fig. 1. Photos of the research material: II, VII, XII. They were taken with a tripod by Nikon D70S camera with MicroNikkor 105 mm-1:2.8D lens, settings: ISO-200, diaphragm – 32, time – 1s

3.1. Test-bench

Experimental studies were performed on the SPOCZEWC test-bench, which is located in the Laboratory of the Heat Transfer Department of The Szewalski Institute of Fluid-Flow Machinery of Polish Academy of Sciences. This test-bench was made according to the idea of researchers [3], and then has been thoroughly modified according to own design of the author of the paper. The basic component of this test-rig is a condenser in which there is a possibility of condensation at lower pressure, at the pressure equal to or higher than atmospheric pressure. The main dimensions of the heat exchanger are as follows: length 1070 mm, outer diameter 405 mm, wall thickness 9 mm. In addition, the condenser was equipped with six sight-glasses for visual inspection of the liquefaction process. The steam source for this test-bench is a modern high-pressure, fully automated, flow-through steam generator manufactured by Clayton (p_n = 18 bar(g), D_n = 950 kg/h). The steam incoming to the test-bench comes from the low-pressure part of the steam system, i.e. p_{l-p} = 5 bar (g). In order to adjust the fine mass flow of the condenser cooling water, flowing through the tested pipe and the pipe without deposits, two vertical multistage centrifugal pumps with integrated frequency converter and PI regulator (CRE1-7 type manufactured by Grundfos, speed of rotation: 360-2840 rev/min) were used. Moreover, to obtain high fidelity of flow rate control, two flow regulating valves equipped with an adjustable pre-setting mechanism (Hydrocontrol R type, made by Oventrop) were used [more in 10, 11].

Data acquisition system DAQ13 of that testing-bench was generally configured on National Instruments devices such as: hardware (modular transducer NI SCXI) and software (programming environment of LabVIEW v.8.6). The measuring system was composed of the following sensors and transducers: thermocouples type K (manufactured by Czaki Thermo-Product, 14 pieces, with an analogue signal), one absolute pressure transmitter with the local reading 1151 type (manufactured by ZAP Pnefal, with a digital signal), two flow-meters for direct measuring of mass flow Promass 40E type (made by Endress+Hauser, with a digital signal). In addition, the data acquisition system was equipped with a system to calibrate measuring channels of temperature, pressure and mass flow. As a result, the accuracy of the test stand measurements has been highly improved. Moreover, the measuring system has got a modular structure which allows for its easier validation and also increases its versatility.

3.2. Research results

Within the each series of measurement, such parameters were maintained at constant level:

- inlet temperature of condenser cooling water for tested pipe with fouling, 19.00°C±0.05 K,
- inlet temperature of condenser cooling water for pipe without fouling, 19.00°C±0.05 K,
- mass flow of condenser cooling water in the tested pipe with fouling, 600 kg/h ±5 kg/h,
- mass flow of condenser cooling water in the pipe without fouling, 600 kg/h ±5 kg/h,
- condensing pressure, 135.0 kPa(a) ±0.5 kPa.

After reaching steady state within the measurement series, an electronic test protocol was prepared by DAQ13 data acquisition system. Tab. 1 shows mean values of measured quantities.

Tab. 1. The mean values of measured quantities for tested tubes: II, VII, XII

Tested tube	$t_{wi,F}$	$t_{wo,F}$	$t_{wi,C}$	$t_{wo,C}$	p_k	$m_{w,F}$	$m_{w,C}$
	[°C]	[°C]	[°C]	[°C]	[kPa(a)]	[kg/h]	[kg/h]
II	19.01	34.09	18.96	37.03	135.7	600.5	598.4
VII	19.01	33.56	18.95	37.37	135.4	602.3	598.0
XII	19.04	36.03	18.99	37.03	135.6	602.4	597.3

The values of physical quantities calculated by using the differential method to designate the specific thermal fouling resistance of tested tubes are included in Tab. 2. Calculations were based on the formulas (1), (2), (9)-(12) and were supported by NIST software such as the Refprop SRD 23 ver. 8.0 program to compute the steam and water properties.

Tab. 2. The values of calculated quantities for tested tubes: II, VII, XII

Tested tube	t_k	$LMTD_F$	$LMTD_C$	$Q_{w,F}$	$Q_{w,C}$	k_F	k_C	$r_{k,F}$	$r_{k,C}$
	[°C]	[K]	[K]	[kW]	[kW]	[W/m^{2}K]	[W/m^{2}K]	[m^{2}K/W]	[m^{2}K/W]
II	108.4	81.6	80.0	10.49	12.54	2621	3586	0.00038	0.00028
VII	108.3	81.8	79.8	10.16	12.77	2574	3664	0.00039	0.00027
XII	108.3	80.5	80.0	11.86	12.49	3330	3575	0.00030	0.00028

The values of specific thermal fouling resistances accumulated on the heat transfer surfaces for tested tubes II, VII and XII were presented in Tab. 3. In addition, Tab. 3 contains values of an absolute and a relative measurement uncertainty of designating the specific thermal resistance of fouling as well as quantitative description of thermal degradation for tested pipes illustrated by values of thermal degradation determinants such as CF and LP indicators.

Tab. 3. The values of the fouling specific heat resistances (r_f) and the values of absolute (Δr_f) and relative ($\Delta r_f/r_f$) measuring uncertainty of specific heat resistances for tested tubes: II, VII, XII,

Tested tube	r_f	Δr_f	$\Delta r_f/r_f$	$\partial r_f/\partial k_F$	Δk_F	$\partial r_f/\partial k_C$	Δk_C	CF	LP
	[m^{2}K/W]	[m^{2}K/W]	[%]	[(m^{2}K/W)^2]	[W/(m^{2}K)]	[(m^{2}K/W)^2]	[W/(m^{2}K)]	[–]	[%]
II	0.00010	1.0E-05	10	–1.0E-07	61	8.0E-08	71	0.73	16
VII	0.00012	1.0E-05	9	–1.5E-07	61	7.4E-08	72	0.70	21
XII	0.00002	8.0E-06	41	–1.5E-08	70	7.8E-08	71	0.93	5

4. Conclusions

The performed experimental studies designed to evaluate the steam power plant heat exchangers thermal degradation arisen due to the deposits presence on their heat transfer surfaces showed that for the two tested tubes II and VII, the process of thermal degradation was significant. It was evidenced by two heat degradation determinants i.e. cleanliness factor *CF* value, 0.73 and 0.70 and the percentage loss of their thermal power, 16 and 21% (LP factor) respectively. While the deposit of XII had a minor negative effect on its heat transfer process (CF = 0.93, LP = 5%).

It is worth noting that measured values of the fouling specific heat resistances were in accordance with the results presented in the literature. Achieved level of the maximum relative uncertainty of the measuring thermal resistances of tested deposits has shown a high measurement accuracy of the testing-bench in the range of specific thermal resistances values greater than 1E-04 m^{2}K/W ($\Delta r_f/r_f$ value was less than or equal to 10% for the II and the VII fouling, whereas in case of the XII fouling, the value gained a less satisfactory level of 40%).

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