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An Influence of Fouling Gathered on the Heat Transfer Surfaces on the Heat Performance Characteristics of the Ship Steam Systems' Heat Exchangers

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ABSTRACT: Gathering of fouling on the heat transfer surfaces, both on the water side and the steam side of heat exchangers of the ship steam systems leads usually to the loss of their heat transfer capacities. This loss appears owing to the high value of the fouling heat resistance and is called the thermal degradation process. It is linked to falling the heat flux transported by a heat exchanger and decreasing total efficiency of the heat unit as well as increasing costs of operation and freight costs after all. The loss of thermal power of a heat exchanger does not only depend on the fouling thermal resistance but is also strictly correlated with the thermal quantities of a given heat exchanger, among others its overall heat transfer coefficient values at various operating modes. The paper presents the aforementioned phenomena and results of the author's own experimental studies, as well.

1 INTRODUCTION

Heat exchange apparatus within the thermal systems of steam power plants, i.e. condensers, low-pressure and high-pressure recuperative exchangers and the heating surfaces of boilers are exposed to the gathering of deposits on their heat exchange surfaces. The source of these sediments usually is the insufficient quality of the working medium and cooling media as well as the physicochemical processes are taken place, e.g. corrosion and erosive processes as several studies [7, 9, 14, 18] have suggested.

The fouling deposited on the heat exchange surfaces, above all, create additional thermal resistance in heat exchange processes [2, 12]. This state of affairs is highly associated with the loss of thermal power of the heat exchange apparatus, leading directly to its thermal (heat) degradation. Thermal degradation is the cause of, among others increase in value of the terminal temperature differences and deterioration of the vacuum degree in condensers [4]. The fouling thermal resistances values of the heat exchange surfaces of steam power plants' heat exchangers, presented in the literature, vary in a wide range. For instance, according to TEMA standards, the values of the deposits specific thermal resistances range from $8.8 \cdot 10^{-5}$ to $100 \cdot 10^{-5}$ m^2K/W [5, 12, 17, 21]. Furthermore, literature on the subject states that the value of thermal resistance of fouling is strongly influenced by the following things: the type of dissolved salts in the water, the surface condition, the construction material of the heat transfer surface, flow geometry, temperature and speed of the working media, i.e. the lower the wall temperature and the higher the speed the flow of water, the less susceptibility of the wall to the deposition of sediments on it [1].

Moreover, Cunningham's research [6] supported the thesis that in the case of the steam power plants condensers, the presence of inert gases within the steam space has a similar effect as the presence of fouling on the heat transfer surfaces of these exchangers. On the other hand, numerical research by Butrymowicz [4] shows a very important conclusion, i.e. the higher the value of the heat transfer coefficient of a given exchanger, the more sensitive the one is to the fouling presence on its heat exchange surface. The above constatation is a very crucial in the field of heat exchangers operation within the steam power plants, and particuliary in the ship power plants, e.g. steam condensers, because of their relatively high values of the heat transfer coefficients.

At the same time, deposits collected inside the heat exchanger tubes (the water side) initiate the process of obliteration. In particular, this phenomenon features the power condensers cooled with sea water [10, 21]. The reduce in cross-section area of the tubes due to formation of various types of particle-dispersion deposits (sulphate, carbonate and silica scale) and the biofouling as well (macro-deposits, e.g. mussels, crustaceans and microorganisms, e.g. bacteria, algae) increases the flow resistance with a simultaneous reduction in efficiency condenser cooling. And finally, this state of things leads to a reduction in the flow velocity of the condenser cooling medium with the consequence that there is an additional increase in the resistance to heat transmission. Hence, the presence of fouling leads not only to thermal degradation of a given exchanger, but it should be expressed stronger, to its heat-and-flow degradation [3, 13].

In view of above-presented results of the research, the issue of thermal degradation of heat transfer apparatus in the steam systems is a vital issue for their operation, e.g. due to the high values of heat transfer coefficients of these apparatus [3, 8, 11, 12, 15, 20].

2 A PHENOMENON OF HEAT EXCHANGERS THERMAL DEGRADATION

The heat output of a heat exchanger $\dot{Q}$ [W] can be expressed as follows,

$$\dot{Q} = \frac{\Delta T_{log}}{R_k} \tag{1}$$

where $\Delta T_{log,C}$, R_k represent the logarithmic temperature difference of the heat exchanger [K] and overall heat transfer resistance [K/W] respectively.

During the heat exchanger using within the normal operational time τ_{ope} i.e. beyond the fouling induction period τ_{ind}, the fouling thermal resistance R_f achieves a positive value. It is worth to mention that τ_{ind} is an initial period of heat exchanger operation in which the accumulated deposits constitute a form of microribs that enhance the heat transfer area and act simultaneously as turbulizers breaking the laminar boundary layer, resulting finally in increase of heat yielding through a given heat exchanger [8]

$$\tau_{ope} > \tau_{ind} \Rightarrow R_f > 0 \tag{2}$$

Thermal resistances are featured by additivity,

$$R_{k,F} = R_{k,C} + R_f \tag{3}$$

where $R_{k,C}$, $R_{k,F}$ constitute the heat transfer resistance of a heat exchanger without fouling (subscript "C" – Clean), and a heat exchanger with fouling (subscript "F" – Fouled).

The thermal resistance R is related to the overall specific thermal resistance r by means of a following relationship,

$$R = \frac{r}{A_{cal}} \tag{4}$$

where A_{cal} describes a calculating heat transfer surface given in square meters.

The overall thermal specific resistance r_k and the heat transfer coefficient k are related by the homographic function [2, 16, 19, 23],

$$r_k = \frac{1}{k} \tag{5}$$

The decrease in the heat capacity of the fouled heat exchanger is expressed as the difference between the following heat fluxes,

$$\Delta\dot{Q}_{loss} = \dot{Q}_C - \dot{Q}_F . \tag{6}$$

where $\dot{Q}_C$, $\dot{Q}_F$ mean the heat power of the exchanger without fouling and the heat power of heat exchanger with fouling respectively.

3 DESCRIPTION OF THE RESEARCH METHOD

The experimental research were carried out for a single tube of the L-P heat recovery exchanger from the steam system. The measurement of the thermal resistance of fouling was performed simultaneously for two types of tubes (Tab.1) i.e. for the tube with a heat exchange area covered with sediment (DKR#02) and for the tube without sediments constituting the reference tube (REB#00). The length of measuring section for two tubes was one meter. The inner diameter of tubes and their thickness equal 12 mm and 2 mm respectively. Fouled tube has got deposit on the outside (vapor side). Both reference and fouled tube are clean inside.

Table 1. Photos of the tested materials: fouled tube DKR#02 and reference tube REB#00

Tube with fouling DKR#02

Tube without fouling (reference) REB#00

[author's own photos, taken with a tripod by Nikon D70S camera with MicroNikkor 105mm-1:2.8D lens]

The heat flux took by the water in the tube with deposits $\dot{Q}_{w,F}$,

$$\dot{Q}_{w,F} = \dot{m}_{w,F} \cdot c_{p,w}\left(t_{wi,F}, t_{wo,F}\right) \cdot \left(t_{wo,F} - t_{wi,F}\right) \tag{7}$$

and the heat flux took by the water in the reference (model) tube $\dot{Q}_{w,C}$,

$$\dot{Q}_{w,C} = \dot{m}_{w,C} \cdot c_{p,w}\left(t_{wi,C}, t_{wo,C}\right) \cdot \left(t_{wo,C} - t_{wi,C}\right) \tag{8}$$

where $\dot{m}_w$, t_{wi}, t_{wo} and $c_{p,w}{}_{t_{wi}}^{t_{wo}}$ represent mass flow of the condenser water cooling [kg/s], temperature of water inlet to the tube [°C], temperature of water outlet from the tube [°C] and average value of specific heat of water in the temperature range from t_{wi} to t_{wo} [J/(kg·K)] respectively.

The fouling thermal resistance r_f has been determined on the basis of the differential method for the direct determination of thermal resistance, i.e. as the difference between a value of the overall thermal resistance of the heat transfer surface with deposits $r_{k,F}$ and a value of the overall thermal resistance for the heat transfer surface without deposits $r_{k,C}$,

$$r_f = r_{k,F} - r_{k,C} \tag{9}$$

The value of the maximum absolute systematic uncertainty of measurement δr_f for the fouling thermal resistance r_f was determined by means of the measurements uncertainties spreading law in accordance with the square error propagation rule [22],

$$\delta r_f = \left(\left(\left|\frac{\partial r_f}{\partial k_F}\right| \cdot \delta k_F\right)^2 + \left(\left|\frac{\partial r_f}{\partial k_C}\right| \cdot \delta k_C\right)^2\right)^{0.5} \tag{10}$$

Considering an operating mode of the heat transfer apparatus at a constant values of temperature difference and the heat exchange surface as well,

$$\Delta T_{log} = \Delta T_{log,C} \cong \Delta T_{log,F} \quad \wedge \quad A = A_C \cong A_F \tag{11}$$

and taking into account the following formulas (1) and from (3) to (5), heat power of the heat exchanger with fouling of the thermal resistance r_f can be evaluated as follows,

$$\dot{Q}_F = \left(\frac{1}{1 + r_f \cdot k_C}\right) \cdot \dot{Q}_C \tag{12}$$

So, the heat power loss for the fouled tube was calculated from the following relationship,

$$\Delta\dot{Q}_{loss} = \left(\frac{r_f \cdot k_C}{1 + r_f \cdot k_C}\right) \cdot \dot{Q}_C \tag{13}$$

The relative heat power loss of the tube with the fouling heat transfer surface was expressed by means the undermentioned RPL_t index,

$$RPL_t = \frac{\Delta\dot{Q}_{loss}}{\dot{Q}_C} \cdot 100\% = \frac{r_f \cdot k_C}{1 + r_f \cdot k_C} \cdot 100\% \tag{14}$$

On the other hand, the relative heat power loss of the tested heat exchanger with a partially fouled heat transfer surface was described by the RPL_{he} ratio,

$$RPL_{he} = \left(1 - \frac{\dot{Q}_F}{\dot{Q}_C}\right) \cdot 100\%. \tag{15}$$

Experimental studies were carried out on the SPOCZEWC test-bench 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 Butrymowicz and Gardzilewicz and then has been thoroughly modified mutatis mutandis according to own design of the author of this paper.

The basic component of the test-bench is a condenser in which there is a possibility of condensation at lower pressure, at the pressure equal to or higher than atmospheric one. The steam source for this test-bench is a modern, fully automated, once-through steam generator (Clayton, p=1.9 MPa, D=950 kg/h). The steam incoming to the test-bench stems from the low-pressure part of the system (p_{LP}=0.6 MPa). The test-stand cooling system is equipped with two circulation pumps (Grundfos, CRE17 type) with a precise control of the cooling water flow thanks to an integrated frequency converter, the PI controller and control valves with smooth positioning control (Oventrop, Hydrocontrol-R type). The data acquisition system was configured on the basis of a modular measuring transducer (NI, SCXI module) and software (LabVIEW v.8.6). It consists of the following sensors and transducers: K-type thermocouples (Czaki), absolute pressure transmitters (Pnefal, 1151 type), Coriolis flowmeters (Endress+Hauser, Promass40E type).

4 RESEARCH RESULTS

The series of measurements consisted of seven measurement points. The following parameters were kept at a constant level within the measuring series: both the inlet condenser cooling water temperature for tube with fouling and the one for tube without fouling (the reference one) at the level 19.00°C±0.05K, the condensing pressure of 135.0 kPa(a)±0.5kPa. Within a single measuring point, the water mass flow rates were kept constant in the fouled tube and in the reference tube, as well. In the experiment plan, the following values of cooling water mass flow rates were assumed, the same for both tubes (ceteris paribus), i.e. 1870, 1530, 1170, 1000, 800, 700 and 600 kg/h±5 kg/h , which were obtained at rotation speed of the cooling water pumps, respectively: 2650, 2150, 1700, 1450, 1200, 1050 and 970 rpm. After reaching the steady state in the measurement, an electronic test protocol was prepared. The average values of the measured values are presented in Table 2.

Table 2. The mean values of measured quantities for tested tubes DKR#02 and REB#00

Tested tube	No.	n [obr/min]	n/n_{max} %	$t_{wi,F}$ [°C]	$t_{wo,F}$ [°C]	$t_{wi,C}$ [°C]	$t_{wo,C}$ [°C]	p_k [kPa]	$m_{w,F}$ [kg/h]	$m_{w,C}$ [kg/h]
DKR#02 (F)	1	2650	93	19.05	25.79	19.01	27.59	135.6	1874.6	1859.9
REB#00 (C)	2	2150	76	19.03	26.93	18.99	28.93	135.7	1534.7	1519.8
	3	1700	60	19.03	28.69	18.99	30.99	135.5	1174.2	1160.7
	4	1450	51	19.06	29.88	19.01	32.35	135.4	1004.3	991.3
	5	1200	42	19.01	31.37	18.95	34.05	136.9	834.4	820.2
	6	1050	37	19.03	32.74	18.97	35.60	135.5	705.2	691.8
	7	970	34	19.01	34.09	18.96	37.03	135.7	602.9	594.0

[author's own research]

Table 3. The mean values of calculated quantities for tested tubes DKR#02 and REB#00

Tested tube	No.	t_k [°C]	$\Delta T_{log,F}$ [K]	$\Delta T_{log,C}$ [K]	$Q_{w,F}$ [kW]	$Q_{w,C}$ [kW]	k_F [W/m²K]	k_C [W/m²K]	$r_{k,F}$ [m²K/W]	$r_{k,C}$ [m²K/W]
DKR#02 (F)	1	108.4	85.9	85.0	14.59	18.63	3464	5019	0.000289	0.000199
REB#00 (C)	2	108.4	85.3	84.3	13.99	17.64	3342	4790	0.000299	0.000209
	3	108.3	84.4	83.2	13.09	16.26	3163	4475	0.000316	0.000223
	4	108.3	83.7	82.5	12.54	15.44	3054	4287	0.000327	0.000233
	5	108.6	83.3	81.9	11.90	14.47	2913	4044	0.000343	0.000247
	6	108.3	82.2	80.8	11.17	13.43	2768	3808	0.000361	0.000263
	7	108.4	81.6	80.0	10.49	12.54	2622	3586	0.000381	0.000279

[author's own study]

Table 4. 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 as well as RPL_t and RPL_{he} indexes

Tested tube	No.	r_f [m²K/W]	δr_f [m²K/W]	$\delta r_f/r_f$ [%]	$\partial r_f/\partial k_F$ [(m²K/W)²]	δk_F [W/(m²K)]	$\partial r_f/\partial k_C$ [(m²K/W)²]	δk_C [W/(m²K)]	RPL_t [%]	RPL_{he} [%]
DKR#02 (F)	1	0.000089	1.6E-05	17.8	-8.34E-08	168	4.0E-08	193	19.2	10.8
REB#00 (C)	2	0.000090	1.4E-05	15.8	-8.96E-08	139	4.4E-08	160	18.5	10.4
	3	0.000093	1.3E-05	13.6	-1.00E-07	109	5.0E-08	126	17.5	9.8
	4	0.000094	1.2E-05	12.5	-1.07E-07	95	5.4E-08	110	16.9	9.4
	5	0.000096	1.1E-05	11.5	-1.18E-07	80	6.1E-08	93	16.1	8.9
	6	0.000099	1.1E-05	10.8	-1.31E-07	69	6.9E-08	81	15.3	8.4
	7	0.000103	1.0E-05	10.2	-1.45E-07	61	7.8E-08	71	14.5	8.2

[author's own study]

The calculated values of analysed quantities are presented in Table 3. The properties of water and steam were received due to the NIST Refprop SRD 23 software, ver. 8.0.

The thermal resistance values of the fouling gathered on the heat transfer surface of the DKR#02 tube and the relative values and absolute values of the measurement uncertainty of the thermal resistance determination are presented in Table 4. This table also includes the value of the relative heat power loss of the fouled tube DKR#02 (the RPLt index was calculated for the average value thermal resistance $r_{f,m}=9.5\cdot10^{-5}$ m²K/W, average of measurements points from 1 to 7) and the value of the relative heat power loss for the heat exchanger equipped with one clean and one fouled tube (the RPL_{he} ratio).

Figure 1 shows the dependence of changes in the relative increase in the heat transfer coefficient k_F (DKR#02) and k_C (REB#00) on the condenser cooling intensity n/n_{max}. The highest value of $n/n_{max}=0.93$ has been achieved at the measurement point #1 a contrario the lowest value of $n/n_{max}=0.34$ has been gained at the measurement point #7). The model values for k_F and k_C were assumed at their minimum values $k_{F,min}$ and $k_{C,min}$, respectively. Figure 2 presents the course of changes in the relative heat power loss for the tube with fouling (the RPL_t) and for the heat exchanger (the RPL_{he}), as well.

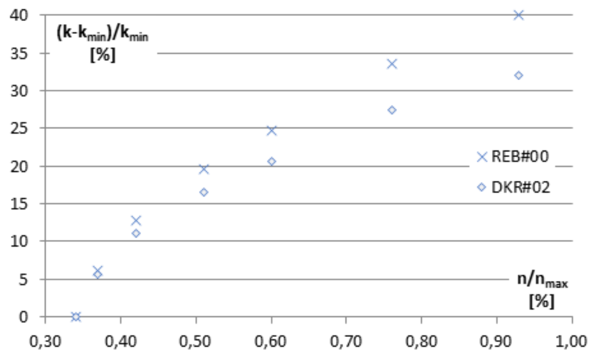

Figure 1. The changes in the relative increase in heat transfer coefficient k_F (DKR#02) and k_C (REB#00) with regard to their minimum values respectively $k_{F,min}$ and $k_{C,min}$ depending on the condenser cooling intensity n/n_{max}

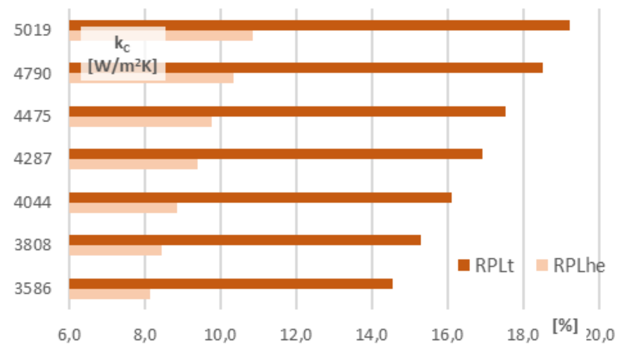

Figure 2. A comparison of changes in the heat power loss indexes the RPL_t and the RPL_{he} as a function of the heat transfer coefficient k_C (without fouling)

5 CONCLUSIONS

The experimental research was aimed at assessing the thermal power of heat exchangers in ship steam systems in relation to the thermal degradation caused by the presence of deposits on the heat transfer surfaces. Research results lead to the conclusion that the appearance of deposits has a significant negative impact on the thermal output of a heat exchange apparatus. It is indicated by the percentage of loss of thermal power of the fouled tube from 14.5% (the condition of the condenser operation at the minimum speed of the cooling water pump i.e. 34% of its maximum speed) to about 19% (the state of the condenser operation with the highest cooling intensification, i.e. 93% of the maximum rotational speed of cooling water).

From the operation point of view of the heat exchangers in steam systems, it is worth emphasizing that the research carried out by the author of this paper also supported an important thesis, i.e. the higher the value of the heat transfer coefficient of the heat transfer apparatus, the more sensitive it is to the presence of fouling on its heat transfer surface. Indeed, because taking into account the averaged value of the measured specific thermal fouling resistance of $9.5 \cdot 10^{-5}$ m²K/W, the largest decrease in heat output for the fouled tube was recorded at the first measuring point about 19% (at the highest value of the heat transfer coefficient for the model tube of approx. 5000 W/m²K) compared to the last measuring point (at the lowest value of the heat transfer coefficient for the model tube of approx. 3600 W/m²K) which was about 5 percentage points less.

When analyzing the test results for the entire heat exchanger, the relative decrease in its heat performance was about two times smaller than the one of the fouled tube, i.e. from 8.5% (at the lowest rotational speed of the cooling water pump) to ca. 11% (at the highest cooling-water pump speed). The scope of heat power loss of the tested exchanger in the whole range of its cooling intensity was approx. two and one-half percentage point. In addition, the intensification of the research condenser cooling greatly improved the heat transport conditions - a monotonic increase in the heat transfer coefficients for both the fouled and the clean tube. Therefore, the highest value of the relative increase in the heat transfer coefficient $\Delta k/k_{min}$ was scrutinize for the clean tube i.e. 40% and it was by 8 percentage points higher than for the fouled tube. The measured value of the fouling thermal resistance confirmed the compliance with the results presented in the research literature. The obtained level of the relative measurement uncertainty maximum value of the fouling thermal resistance indicates a high measuring accuracy of the test stand in the range of the measured values from 8.9 to 10.3 the $\delta r_f/r_f$ value was lower than 20%.

It is worth emphasizing that the operation subsoil of the modern ship steam systems has a deep sozologic dimension. In the case of heat exchangers operation of ship steam systems, the distinguishing feature of their proper use is the respect towards the energy transferred through them. Hence, *inter alios*, the care of the technical condition of the heat exchange surfaces in the steam systems is one of the substantial issues of their proper using. To put in briefly such an attitude and action prevents, in essence, the increase in shipping costs by sea.

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