Scientific Journal of Gdynia Maritime University

No. 128/23, 78–92
ISSN 2657-6988 (online)
ISSN 2657-5841 (printed)
DOI: 10.26408/128.05

Submitted: 27.10.2023
Accepted: 16.11.2023
Published: 29.12.2023

EVALUATION OF THE PROCESS OF CONDENSATION OF WATER VAPOUR IN THE PRESENCE OF DEPOSITS USING A SINGLE PIPE IN THE HEAT EXCHANGER OF A CONDENSING POWER PLANT AS AN EXAMPLE

Tomasz Hajduk
Gdynia Maritime University, 81-87 Morska St., 81-225 Gdynia, Poland,
Faculty of Mechanical Engineering, Department of Marine Power Plants
ORCID 0000-0002-3683-6542
e-mail: t.hajduk@wm.umg.edu.pl

Abstract: Additional thermal resistance caused by deposits accumulating on the heat exchange surfaces of condensers and in the regenerative exchangers of steam power plants most often results in a deterioration in the heat exchange process, which manifests itself, amongt other things, in a reduction in the thermal power of a given heat exchange device and an increase in the working pressure on the steam side. Moreover, such deposits sometimes form irregularities with a diverse geometric structure, hence the description of the water vapour condensation process is not always easy to interpret due to the coexistence of many phenomena at the same time. The article describes selected theoretical issues regarding the presence of deposits on heat exchange surfaces and presents the results of the author's own experimental research on heat transfer in the process of condensing water vapour on an example of a single pipe with deposits taken from the heat exchanger of a steam condensing power plant.

Keywords: condensing power plants, heat exchangers, deposits on heat exchange surfaces, condensation of water vapour.

1. INTRODUCTION

The presence of fouling on the heat exchange surfaces, on the water side and on the steam side, in condensers and regenerative exchangers of steam power plants is manifested by an increase in wall temperature on the steam side, amongst other things. This increase is a result of additional thermal resistance introduced by the accumulated deposits and leads to a deterioration in the water vapour condensation process. It manifests itself, amongst other things, in a decrease in the thermal efficiency of a given exchanger and a reduction in the mass flow of water vapour absorbed by the exchanger [Adamson 1981; Knudsen 1981; Butrymowicz 2001].

The above statement was supported by research performed by Butrymowicz and Gardzilewicz (1996), which showed a two-fold decrease in the mass flow of steam drawn by a low-pressure regeneration heater for the specific thermal resistance of deposits at a level of 0.0005 m^{2}K/W (condensing pressure 28.32 kPa: cooling water inlet temperature: 30.7°C; cooling water outlet temperature: 65.8°C; heat exchange surface area: 153 m$^{2;}$ heat transfer coefficient of the exchanger without deposits: 4,080 W/m^{2}K).

Additionally, fouling most often form irregularities with a diverse, often stochastic, geometric structure of the surface of the surface layer (Fig. 1), [Epstein 1999; Karabelas 2001; Kazi, Duffy and Chen 2002; Brahim, Augustin and Bohnet 2003; Hajduk 2018].

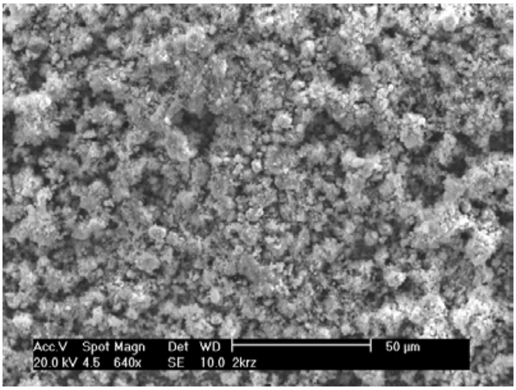

Fig. 1. Scanning photo of the surface layer of deposits accumulated on the heat exchange surface on the steam side of the regeneration exchanger of the KECKR02 domestic diesel power plant, magnified 640 times

Source: [Hajduk 2018].

The most common measure of this unevenness are roughness parameters describing the geometric structure of the surface [Kazi, Duffy and Chen 2002; Adamczak 2005; Kukulka and Devgun 2007; Wajs and Mikielewicz 2014]. An interesting fact is that the roughness of the heat exchange surface caused by deposits may, on the one hand, intensify the heat exchange process and, on the other hand, contribute to accelerating the thermal degradation of the exchanger [Knudsen 1981; Brodowicz and Markowski 1995; Rusowicz 2004; Butrymowicz and Hajduk 2006; Górski and Perepeczko 2013].

Brahim et al. (2003) investigated the influence of the shape of the geometric structure of the surface of deposits accumulated on the cooling water side of the heat exchanger on the time characteristics of the thermal resistance of deposits. The test results have proven that greater unevenness of the heat transfer surface on the side of the water cooling the exchanger results in a smaller increase in the value of thermal resistance of fouling over time. This occurs when surface irregularities go beyond the laminar layer and reach the turbulent core, causing disturbances in the laminar layer. This increases the amount of heat exchanged, which, in the case of a condenser, directly translates into the intensification of the condensation process. It should be noted, however, that although an increase in roughness causes an increase in the heat transfer coefficient, it is also attributable to an increase in flow resistance. This was confirmed by Cope's research on the influence of pipe roughness on the heat transfer coefficient from the inside of the pipe. Therefore, if smooth and rough pipes are compared in terms of the amount of heat transferred per unit of power needed to force the fluid through the pipe, it may emerge that smooth pipes are more advantageous.

An increase in the roughness of the heat exchange surface on the steam side may favour the formation of a thicker condensate layer, thus worsening the so-called "runoff conditions" during the water vapour condensation process.

This phenomenon occurs due to surface tension forces when the width of a finned duct is much smaller than the capillary constant of water vapour. It has an adverse effect on the operation of a single pipe because the layer of condensate remaining in the lower part, with a thickness approximately equal to the height of the fins, creates additional thermal resistance [Hobler 1986].

Moreover, the experiments of Förster and Bohnet (2002) using calcium sulphate ($CaSO_4$) as a deposit showed a significant shortening of the deposit formation time (so-called "fouling induction time") due to greater roughness of the heat transfer surface (Fig. 2).

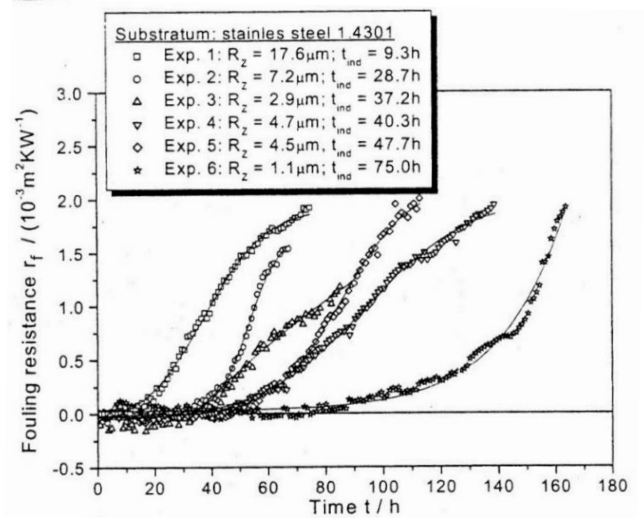

Fig. 2. Temporal characteristics of thermal resistance of $CaSO_4$ (r_f) for different values of roughness of the heat transfer surface, expressed by the mean maximum profile height (R_z)

Source: [Förster and Bohnet 2002].

The effect of the presence of deposits on the wall of a single pipe is sometimes taken into account by correcting the heat transfer coefficient α_V calculated for the condensing steam side. According to Kutateladze, the value of this coefficient should be corrected with correction factor ε depending on the pipe material and the condition of its surface. For example, the correction factor for a steel pipe covered with a thin layer of deposit is 0.67 [Michiejew 1953].

Webb (1981) states in one review paper that a significant amount of research on the intensification of vapour condensation concerns the condensation of organic and synthetic fluids, because in the case of water vapour condensation the main resistance to heat transfer lies on the cooling water side. Therefore, the heat transfer coefficient is not corrected for the water vapour side. However, if additional thermal resistance to conduction is created on the heat exchange surface on the water vapour side, due to the formation of a layer of metal oxides [Dobosiewicz 1996], then the change in the geometric structure of the surface of the surface layer, caused by the presence of these oxides, may partially eliminate the resulting thermal resistance.

Moreover, if the geometric structure of the top layer of deposits on the side of the condensing steam has a shape resembling corrugations, such a surface texture may lead to the intensification of water vapour condensation, similar to the *Gregorig effect* widely described in the relevant literature [Bonca and Butrymowicz 1994].

To sum up, the influence of contaminants accumulated on the heat exchange surfaces of heat exchange devices of steam power plants on the process of steam condensation is often not easy to interpret due to its specific nature, sometimes caused by the interference of a number of phenomena.

2. RESEARCH METHODOLOGY

Experiments were carried out at the SPOCZEWC research station located in the Robert Szewalski Heat Transfer Laboratory in the Heat Transfer Department of the Institute of Fluid-Flow Machinery of the Polish Academy of Sciences. This station was build according to the concept of Butrymowicz and Gardzilewicz (1996) and was then thoroughly modified according to the intentions of the author of this article.

The essential element of the station is a heat exchanger in which the water vapour condensation process can occur at pressure levels both lower and higher than that atmospheric pressure. A modern fully automatic Clayton type E-60 steam generator is the source of steam. The test station is supplied with steam from the low-pressure part of the steam system with a pressure of p_{LP} = 0.6 MPa. Precise regulation of the mass flow rate of the condenser cooling water is ensured by two Grundfos centrifugal pumps of type CRE 1-7 with integrated frequency converters and PI controllers and two Oventrop Hydrocontrol-R control valves equipped with mechanisms for smooth regulation of the valve opening degree (100 valve positions available).

The DAQ13 data acquisition system of the research station is characterised by a modular architecture enabling it to be modified if necessary. The system was configured using hardware (NI SCXI measurement transducer) and software (LabVIEW v.8.6) from National Instruments. The measurement tracks are equipped with the following transducers and sensors: flow meters for direct mass flow measurement (Endress+Hauser, Promass 40 E), absolute pressure transmitter (ZAP Pnefal, type 1151) and thermocouples (Czaki Thermo-Produkt, type K). In addition, the DAQ13 data acquisition system is equipped to calibrate temperature, pressure and mass flow measurement channels, which enables a high level of measurement accuracy. Details of the research site are presented in [Hajduk et al. 2006].

Experiments were carried out for a heat exchanger equipped with two pipes for which heat fluxes were measured simultaneously. One of them had an external heat exchange surface regularly covered with a thin layer of deposit of a rusty-bluish colour and a hard structure. This pipe is marked with the code DAG#12 (Fig. 3).

Fig. 3. Photo of a section of the pipe with DAG#12 deposits coming from the regeneration exchanger of the low-pressure condensing power plant Photo: Nikon D70S camera, MicroNikkor 105 mm-1:2.8D lens, tripod

Source: own archive.

The second pipe was a reference pipe (code REB#00), whose heat transfer surface was free of deposits. Moreover, the measured and calculated values for the pipe with deposits are marked with the subscript “F” (Fouled), while the pipe without deposits is described with the subscript “C” (Clean). Basic geometric dimensions of the tested pipes: measuring section length 1 m, internal diameter 12 mm, wall thickness 2 mm.

Heat rate absorbed by water (reference) in the pipe with deposits $\dot{Q}_{w,F}$:

$$\dot{Q}_{w,F} = \dot{m}_{w,F} \cdot c_{p,w}{}_{t_{wi,F}}^{t_{wo,F}} \cdot \left(t_{wo,F} - t_{wi,F}\right) \tag{1}$$

Heat rate absorbed by water (reference) in the pipe without deposits $\dot{Q}_{w,C}$:

$$\dot{Q}_{w,C} = \dot{m}_{w,C} \cdot c_{p,w}{}_{t_{wi,C}}^{t_{wo,C}} \cdot \left(t_{wo,C} - t_{wi,C}\right) \tag{2}$$

Heat rate released by water vapour (auxiliary) in the pipe with deposits $\dot{Q}_{v,F}$:

$$\dot{Q}_{v,F} = \dot{m}_{c,F} \cdot h_{fg}(t_k) + \dot{m}_{c,F} \cdot c_{p,\mathrm{v}}{}_{t_k}^{t_{vsh,o}} \left(t_{vsh,o} - t_k\right)_{p_k} \tag{3}$$

Heat rate released by water vapour (auxiliary) in the pipe without deposits $\dot{Q}_{v,C}$:

$$\dot{Q}_{v,C} = \dot{m}_{c,C} \cdot h_{fg}(t_k) + \dot{m}_{c,C} \cdot c_{p,\mathrm{v}}{}_{t_k}^{t_{vsh,o}} \left(t_{vsh,o} - t_k\right)_{p_k} \tag{4}$$

Mass flow rate of condensate from the pipe with deposits $\dot{m}_{c,F}$:

$$\dot{m}_{c,F} = \rho_{c,F}\left(t_{c,F}\right) \cdot \dot{V}_{c,F} \tag{5}$$

Mass flow rate of condensate from the pipe without deposits $\dot{m}_{c,C}$:

$$\dot{m}_{c,C} = \rho_{c,C}\left(t_{c,C}\right) \cdot \dot{V}_{c,C} \tag{6}$$

Volumetric flow rate of condensate from the pipe with deposits $\dot{V}_{c,F}$:

$$\dot{V}_{c,F} = \frac{V_{cF,stop} - V_{cF,start}}{\tau_F} \tag{7}$$

Volumetric flow rate of condensate from the pipe without deposits $\dot{V}_{c,C}$:

$$\dot{V}_{c,C} = \frac{V_{cC,stop} - V_{cC,start}}{\tau_C} \tag{8}$$

Mean logarithmic surface of heat transfer for the pipe with deposits $A_{log,F}$:

$$A_{log,F} = \frac{A_{o,F} - A_{i,F}}{ln\frac{A_{o,F}}{A_{i,F}}} \tag{9}$$

Mean logarithmic surface of heat transfer for the pipe without deposits $A_{log,C}$:

$$A_{log,C} = \frac{A_{o,C} - A_{i,C}}{ln\frac{A_{o,C}}{A_{i,C}}} \tag{10}$$

Mean logarithmic temperature difference for the pipe with deposits $\Delta T_{log,F}$:

$$\Delta T_{log,F} = \frac{t_{wo,F} - t_{wi,F}}{ln\left(\frac{t_k - t_{wi,F}}{\delta t_F}\right)} \tag{11}$$

Mean logarithmic temperature difference for the pipe without deposits $\Delta T_{log,C}$:

$$\Delta T_{log,C} = \frac{t_{wo,C} - t_{wi,C}}{ln\left(\frac{t_k - t_{wi,C}}{\delta t_C}\right)} \tag{12}$$

Overall heat transfer coefficient for the pipe with deposits k_F:

$$k_F = \frac{\dot{Q}_{w,F}}{A_{log,F} \cdot \Delta T_{log,F}} \tag{13}$$

Overall heat transfer coefficient for the pipe without deposits k_C:

$$k_C = \frac{\dot{Q}_{w,C}}{A_{log,C} \cdot \Delta T_{log,C}} \tag{14}$$

Where:

$(\ )_F$ – subscript "F: refers to the pipe with deposits,
$(\ )_C$ – subscript "C" refers to the pipe without deposits,

$(\ )_i$ – subscript "i" refers to the inlet part of the exchanger,
$(\ )_o$– subscript "o" refers to the outlet part of the exchanger,
A_o– external surface of the tested pipe [m^2],
A_i– internal surface of the tested pipe [m^2],
${c_{p,w}}_{t_{wi}}^{t_{wo}}$ – mean specific heat of water in the temperature ranges t_{wi} and t_{wo} [J/(kg·K)],
${c_{p,v}}_{t_{vsh,i}}^{t_{vsh,o}}$ – mean specific heat of steam in the temperature ranges $t_{vsh,i}$ and $t_{vsh,o}$ [J/(kg·K)],
δt – terminal temperature difference of the tested pipe [K],
$h_{fg}(t_k)$ – heat of phase change at condensation temperature t_k [kJ/kg],
p_k – water vapour condensation pressure [Pa],
τ – measurement time of condensate flow from the tested pipe [s],
t_k – water vapour condensation temperature [°C],
t_{vsh} – temperature of superheated steam [°C],
t_w – temperature of water cooling the tested pipe [°C],
$V_{c,start}$ – volume at the beginning of the measurement of the amount of condensate for the tested pipe [m^3],
$V_{c,stop}$ – volume at the end of the condensate measurement for the tested pipe [m^3].

The heat rate $\dot{Q}_w$ absorbed by water is a function of the following variables:

$$\dot{Q}_w = f\left(\dot{m}_w, t_{wi}, t_{wo}, c_p(t_{wi}), c_p(t_{wo})\right) \tag{15}$$

Formula (15) can be expressed in a generalised notation:

$$\dot{Q}_w = f(w_1, w_2, w_3, w_4, w_5) \tag{16}$$

Hence, the value of the maximum absolute systematic uncertainty, $\Delta\dot{Q}_w$, in the measurement of the heat rate absorbed by water, $\dot{Q}_w$, was determined based on Gauss's law of error propagation [Kotlewski and Mieszkowski 1972]:

$$\dot{Q}_w = \sqrt{\sum_{i=1}^{i=5}\left(\frac{\partial\dot{Q}_w}{\partial w_i}\cdot\Delta w_i\right)^2} \tag{17}$$

The heat rate, $\dot{Q}_v$, released by water vapour depends on the following variables:

$$\dot{\mathrm{Q}}_{\mathrm{v}} = \mathrm{f}\left(\dot{\mathrm{m}}_{\mathrm{c}}, \mathrm{h}_{\mathrm{fg}}, \mathrm{t}_{\mathrm{vsh}}, \mathrm{t}_{\mathrm{k}}, \mathrm{c}_{\mathrm{p}}(\mathrm{t}_{\mathrm{vsh}}), \mathrm{c}_{\mathrm{p}}(\mathrm{t}_{\mathrm{k}})\right) \tag{18}$$

Formula (18) can be written in a generalised notation:

$$\dot{Q}_v = g(v_1, v_2, v_3, v_4, v_5, v_6) \tag{19}$$

Like the $\dot{Q}_w$, rate, the value of the maximum absolute systematic uncertainty, $\Delta\dot{\mathrm{Q}}_{\mathrm{v}}$, in the measurement of the heat rate released by water vapour, $\dot{Q}_v$, was determined using the equation:

$$\Delta \dot{Q}_v = \sqrt{\sum_{i=1}^{i=6} \left(\frac{\partial \dot{Q}_v}{\partial v_i} \cdot \Delta v_i \right)^2} \tag{20}$$

The thermal resistance of the fouling, r_f, was determined using the differential method of direct determination of thermal resistance, i.e. as the difference between the total thermal resistance for the pipe with deposits, $r_{k,F}$, and the total thermal resistance for the deposit-free reference pipe, $r_{k,C}$:

$$r_f = r_{k,F} - r_{k,C} \tag{21}$$

which corresponds to the equation:

$$r_f = \frac{1}{k_F} - \frac{1}{k_C} \tag{22}$$

The heat balance coefficient HBC_F for measurements related to the pipe covered with deposits was calculated based on the equation:

$$HBC_F = \left| \frac{\dot{Q}_{v,F} - \dot{Q}_{w,F}}{\dot{Q}_{w,F}} \right| \tag{23}$$

By analogy, the heat balance coefficient HBC_C was determined for measurements related to the pipe without deposits:

$$\mathrm{HBC_C} = \left| \frac{\dot{\mathrm{Q}}_{\mathrm{v,C}} - \dot{\mathrm{Q}}_{\mathrm{w,C}}}{\dot{\mathrm{Q}}_{\mathrm{w,C}}} \right| \tag{24}$$

3. RESEARCH RESULTS

The measurement series of experimental tests included six measurement points, which were performed simultaneously for both pipes during continuous operation of the station over a time frame of several hours. After reaching a steady state at a given measurement point, an electronic test protocol was prepared. In the experimental plan, during the measurement series, the temperature of the cooling water at the inlet to the tested pipes and the condensation pressure in the heat exchanger were kept constant.

Moreover, the mass flow rate of cooling water in both tested pipes was kept constant within a single measurement point. The following values of cooling water mass flows were assumed *(ceteris paribus)* for the six measurement points:

- $\dot{m}_w \in$ (1,870 – nominal); 1,530; 1,070; 800; 700 and 600 kg/h), which corresponded to the following rotational speeds of the cooling water pumps;
- $n \in$ (2,650 – nominal); 2,150; 1,550; 1,200; 1,050 and 970 rpm).

3.1. Measurands

The mean values of the measured values are summarised in Tables 1 and 2.

Table 1. Mean values of the measurands for the streams drawn by cooling water for the pipes tested

Meas. #	n	n/n_{nom}	p_k	DAG#12			REB#00		
	[rpm]	[%]	[kPa]	$t_{wi,F}$	$t_{wo,F}$	$m_{w,F}$	$t_{wi,C}$	$t_{wo,C}$	$m_{w,C}$
				[°C]	[°C]	[kg/h]	[°C]	[°C]	[kg/h]
1	2650	100	135.4	19.04	27.19	1874.6	18.99	27.61	1861.0
2	2150	81	135.6	18.99	28.40	1534.8	18.94	28.92	1521.0
3	1550	58	135.4	19.04	31.09	1074.5	18.98	31.76	1062.2
4	1200	45	135.5	19.03	33.38	804.2	18.97	34.21	792.5
5	1050	40	135.7	19.03	34.57	703.5	18.98	35.47	691.6
6	970	37	135.5	19.04	36.03	604.8	18.99	37.03	592.9

Source: own research.

Table 2. Mean values of the measurands for the stream released by steam for the pipes tested

Meas. #	p_k	DAG#12				REB#00			
	[kPa]	τ_F		V_F [dm^3]		τ_C		V_C [dm^3]	
		[min]	[s]	start	stop	[min]	[s]	start	stop
1	135.4	4	14	0	2	4	2	0	2
2	135.6	4	45	0	2	4	20	0	2
3	135.4	5	22	0	2	4	48	0	2
4	135.5	5	36	0	2	5	18	0	2
5	135.7	6	10	0	2	5	38	0	2
6	135.5	6	19	0	2	6	0	0	2

Source: own research.

3.2. Calculated quantities

The values of the calculated quantities related to the measurements on the cooling water side and the steam side are shown in Tables 3 and 4. The properties of water and water vapour were determined using NIST Refprop SRD23 software.

Table 3. Values of quantities calculated on the cooling water side

Meas. #	t_k	DAG#12				REB#00			
	[°C]	$\Delta T_{log,F}$	$Q_{w,F}$	k_F	$r_{k,F}$	$\Delta T_{log,C}$	$Q_{w,C}$	k_C	$r_{k,C}$
		[K]	[kW]	[W/(m^{2}K)]	[m^{2}K/W]	[K]	[kW]	[W/(m^{2}K)]	[m^{2}K/W]
1	108.3	85.1	17.63	4679	0.000214	84.9	18.73	5047	0.000198
2	108.4	84.5	16.67	4453	0.000225	84.3	17.73	4814	0.000208
3	108.3	83.0	14.94	4064	0.000246	82.8	15.85	4385	0.000228
4	108.3	81.9	13.32	3674	0.000272	81.5	14.10	3962	0.000252
5	108.4	81.3	12.62	3507	0.000285	80.9	13.32	3771	0.000265
6	108.3	80.5	11.86	3330	0.000300	80.0	12.49	3575	0.000280

Source: own research.

Table 4. Values of the quantities calculated on the water vapour side

Meas. #	t_k	DAG#12			REB#00		
	[°C]	$V_{c,F}$	$m_{c,F}$	$Q_{v,F}$	$V_{c,C}$	$m_{c,C}$	$Q_{v,C}$
		[dm^3/s]	[kg/s]	[kW]	[dm^3/s]	[kg/s]	[kW]
1	108.3	0.0079	0.0075	16.91	0.0083	0.0079	17.75
2	108.4	0.0070	0.0067	15.08	0.0077	0.0073	16.53
3	108.3	0.0062	0.0059	13.36	0.0069	0.0066	14.94
4	108.3	0.0060	0.0057	12.78	0.0063	0.0060	13.50
5	108.4	0.0054	0.0052	11.61	0.0059	0.0057	12.71
6	108.3	0.0053	0.0050	11.34	0.0056	0.0053	11.94

Source: own research.

The values of thermal resistance of fouling, r_f, and HBC heat balance coefficients are included in Table 5. The arithmetic mean value of the specific thermal resistance of the deposits accumulated in the DAG #12 pipe was $r_{f,m} = 1{,}9 \cdot 10^{-5}$ m^{2}K/W.

Table 5. The value of the thermal resistance of the deposits, r_f, and the heat balance coefficients HBCF and HBCC for the research materials DAG#12 and REB#00 respectively

Meas. #	DAG#12		REB#00
	HBC_F	r_f	HBC_C
	[-]	[m^{2}K/W]	[-]
1	0.04	0.000016	0.05
2	0.09	0.000017	0.07
3	0.11	0.000018	0.06
4	0.04	0.000020	0.04
5	0.08	0.000020	0.05
6	0.04	0.000021	0.04

Source: own research.

The results of the calculation of measurement errors for heat rates are presented in Table 6.

Table 6. Maximum absolute and relative values of measurement uncertainty of heat rates on the cooling water side and on the steam side

Meas. #	**Cooling water**				**Water vapour**			
	$\Delta Q_{w,F}$	$\Delta Q_{w,F}/Q_{w,F}$	$\Delta Q_{w,C}$	$\Delta Q_{w,C}/Q_{w,C}$	$\Delta Q_{v,F}$	$\Delta Q_{v,F}/Q_{v,F}$	$\Delta Q_{v,C}$	$\Delta Q_{v,C}/Q_{v,C}$
	[W]	**[%]**	**[W]**	**[%]**	**[W]**	**[%]**	**[W]**	**[%]**
1	411	2.3	409	2.2	385	2.3	404	2.3
2	340	2.0	338	1.9	343	2.3	377	2.3
3	244	1.6	243	1.5	304	2.3	340	2.3
4	188	1.4	187	1.3	291	2.3	308	2.2
5	168	1.3	167	1.3	264	2.2	289	2.2
6	148	1.2	147	1.2	258	2.2	272	2.2

Source: own research.

Figure 4 shows changes in the steam flow for the pipe with deposits (DAG#12) and the pipe without deposits (REB#00) depending on the intensity of their cooling expressed by the value of the cooling water flow ($m_{w,max}$ @ meas. 1; $m_{w,min}$ @ meas. 6).

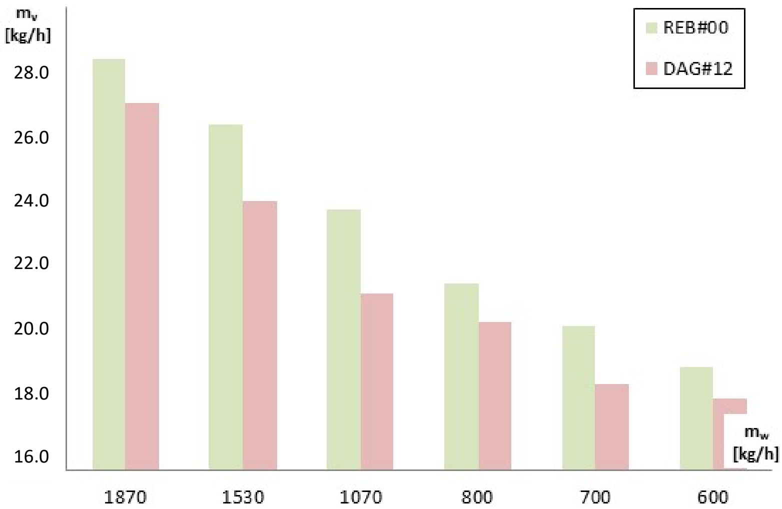

Fig. 4. The course of changes in the mass flow of water vapour, m_v, in the presence of deposit with a thermal resistance value of $1{,}9{\cdot}10^{-5}$ m^2K/W depending on the cooling intensity of the heat exchanger for the DAG#12 and REB#00 pipes tested

Source: own research.

Figure 5 illustrates the changes in thermal power loss on the steam side for the DAG#12 and REB#00 pipes tested, depending on the cooling intensity of the heat exchanger expressed as the ratio of cooling water streams (flux) at a given measurement point to the nominal flux $m_w/m_{w,nom}$ (max: $m_w/m_{w,nom}$ = 1,00 @ meas. 1; min: $m_w/m_{w,nom}$ = 0,32 @ meas. 6). The heat power loss was presented as the value of the difference between the heat flow at a given measurement point, while the nominal heat flow referred to the nominal heat flow ((Q_v - $Q_{v,nom}$)/$Q_{v,nom}$ = 0 @ meas. 1).

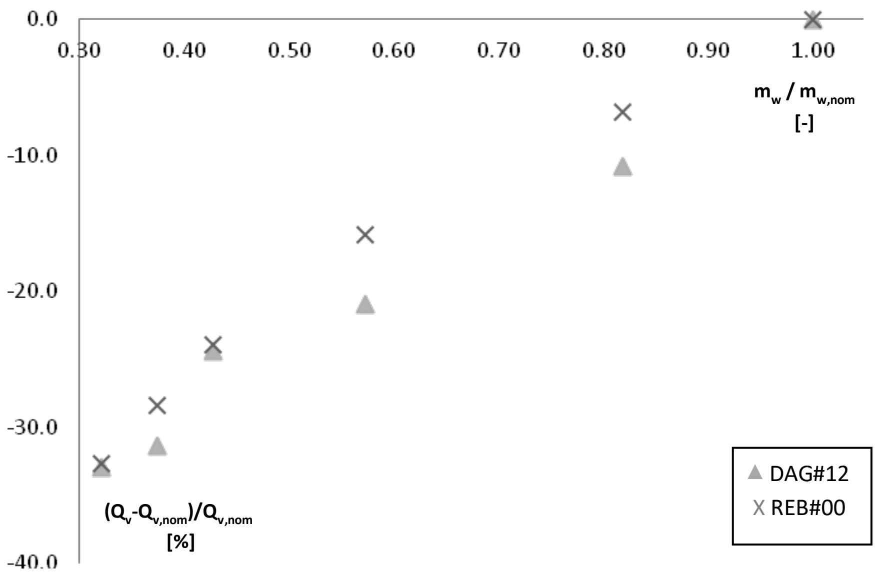

Fig. 5. Change in the relative loss of steam heat rate $Q_{v,F}$ and $Q_{v,C}$ in relation to their nominal values $Q_{v,Fnom}$ and $Q_{v,Cnom}$, respectively, depending on the cooling intensity of the heat exchanger $m_w/m_{w,nom}$, in the presence of deposit ($r_{f,m}$ = 1,9·10^{-5} m^2K/W)

Source: own research.

4. FINAL CONCLUSIONS

The comparative analysis of the results of the experimental studies, the aim of which was to assess heat transfer in the process of condensing water vapour in a single pipe with the heat exchange surface covered with deposits, supports the general rule that the presence of deposits on the heat exchange surface leads to thermal degradation of the heat exchanger.

This is indicated by the values of the mass flow of water vapour condensed in the pipe with deposits, which were always lower than those found in the pipe without deposits, for all cooling conditions in the heat exchanger (measurements 1–6). The relative loss of the mass flow of water vapour condensing in the pipe covered with deposits, with a thermal resistance value of $11{,}9 \cdot 10^{-5}$ m^2K/W, ranged from 5% to 11%. The value of the thermal resistance of deposits obtained is, according to the relevant literature, rather low.

The course of thermal power losses on the steam side, expressed as a relative value – as the difference between the heat flow at a given measurement point and the nominal heat flow related to the value of the nominal flow – was more less favourable for the pipe with deposits for each of the tested pipes. This is indicated by the percentage of heat power loss for the pipe with deposits from approx. 11% (cooling intensity $m_w/m_{w,nom} = 0.2$ @ meas. 2) to approx. 33% (cooling intensity $m_w/m_{w,nom} = 0.32$ @ meas. 6). For the pipe without deposits, the thermal power loss ranged from ca. 7% to ca. 33%. It should be mentioned that the effect of the presence of deposits was insignificant for lower values of the cooling water flow ($m_w/m_{w,nom} = 0.43$ @ meas. 4; $m_w/m_{w,nom} = 0.37$ @ meas. 5 and $m_w/m_{w,nom} = 0\ .32$ @ meas. 6).

It should also be mentioned that the values of heat rates, both on the cooling water and steam sides, did not differ significantly. This is indicated by the relatively low values of the heat balance coefficients HBCF and HBCC for the pipes tested. In thermal measurements, it is assumed that the test station is properly balanced if the heat balance coefficient HBC is less than 0.15 during the measurements. It is noteworthy that all the measurements showed the above inequality during the experimental studies (seven HBC coefficients reached values smaller than or equal to 0.05, four of them had values in the range from more than 0.05 up to 0.10, while one had a value greater than 0.10 and less than 0.15).

Moreover, one might add that the operation of modern heat exchangers in the thermal systems of condensing power plants has a profound sozological dimension. One of the determinants of the proper use of heat exchange devices is respect for the energy transferred by them. Therefore, it is important to pay attention to the technical condition of the heat exchange surfaces of these devices if they are to operate properly.

REFERENCES

Adamczak, S., 2005, *Normalizacja pomiarów struktury geometrycznej powierzchni Cz. 7. Ocena chropowatości i falistości powierzchni. Informacje podstawowe,* Mechanik, no. 5–6, pp. 492–495.

Adamson, W.L., 1981, *The Impact of Fouling on Condenser Design and Operation,* [in:] Marto, P.J. (ed.), Power Condenser Heat Transfer Technology, Publishing Co.

Bonca, Z., Butrymowicz, D., 1994, *Eksperymentalne badania wymiany ciepła w procesie skraplania freonu R22 na rurach gęstożebrowanych*, Prace Instytutu Maszyn Przepływowych, Wydawnictwo Instytutu Maszyn Przepływowych PAN, Gdańsk, vol. 97, no. 1–2, pp. 29–61.

Brahim, F., Augustin, W., Bohnet, M., 2003, *Numerical Simulation of the Fouling Structured Heat Transfer Surfaces*, ECI Conference on Heat Exchanger Fouling and Cleaning, Fundamentals and Applications, pp. 121–129.

Brodowicz, K., Markowski, M., 1995, *Metoda projektowania wymienników ciepła o małej wrażliwości na opory cieplne osadów*, IX Sympozjum Wymiany Ciepła i Masy, Augustów, pp. 177–186.

Butrymowicz, D., 2001, *Influence of Fouling and Inert Gases on the Performance of Regenerative Feedwater Heaters*, Archives of Thermodynamics, vol. 23, no. 1–2, pp. 127–140.

Butrymowicz, D., Gardzilewicz, A., 1996, *Analiza możliwości pomiaru oporu cieplnego zanieczyszczeń powierzchni wymienników ciepła*, Wydawnictwo Instytutu Maszyn Przepływowych PAN, Gdańsk.

Butrymowicz, D., Hajduk, T., 2006, *Zagadnienia degradacji termicznej wymienników ciepła*, Technika Chłodnicza i Klimatyzacyjna, R. XIII, no. 3(121), pp. 111–117.

Dobosiewicz, J., 1996, *Korozja mosiężnych rurek skraplaczy od strony pary wodnej*, Biuletyn Pro Novum, Energetyka, vol. L, no. 5(503), pp. 300(17)–301(18).

Epstein, N., 1999, *Particle Deposition and Its Mitigation*, [in:] Bott, T.R. (ed.), *Understanding Heat Exchanger Fouling and Its Mitigation*, Begell House, Inc., New York, USA, pp. 3–21.

Förster, M., Bohnet, M., 2002, *Modification of the Interface Crystal/Heat Transfer Surface to Reduce Heat Exchanger Fouling*, [in:] Müller-Steinhagen, H. (ed.), *Heat Exchanger Fouling. Fundamental Approaches & Technical Solutions*, Publico Publications, Essen, Germany.

Górski, Z., Perepeczko, A., 2013, *Okrętowe kotły parowe*, Wydawnictwo Akademii Morskiej w Gdyni, Gdynia.

Hajduk, T., 2018, *Research on Deposit Accumulated on Heat Exchange Surfaces in the Light of Thermal Degradation of Heat Exchange Aparatus of Steam Power Plants, Part I: Study of Real Sediments*, Polish Maritime Research, vol. 25, no. 1(97), pp. 99–107.

Hajduk, T., Karwacki, J., Butrymowicz, D., Szyrszyng, R., 2006, *Modyfikacja układu przepływowego stanowiska do badań oporów cieplnych zanieczyszczeń*, Instytut Maszyn Przepływowych PAN, no. 6774, Gdańsk.

Hobler, T., 1986, *Ruch ciepła i wymienniki,* WNT, Warszawa.

Karabelas, A.J., 2001, *Scale Formation in Tubular Heat Exchangers – Research Priorites*, Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, pp. 73–81.

Kazi, S.N., Duffy, G.G., Chen, X.D., 2002, *A Study of Fouling and Fouling Mitigation on Smooth and Roughened Metal Surfaces and a Polymeric Material*, [in:] Müller-Steinhagen, H. (ed.), *Heat Exchanger Fouling. Fundamental Approaches & Technical Solutions*, PP Publico Publications, Essen, Germany, pp. 65–72.

Knudsen, J.G., 1981, *Fouling of Heat Transfer Surfaces: An Overview*, [in:] Marto, P.J., Nunn, R.H., (eds.), *Power Condenser Heat Transfer Technology*, Hemisphere Publishing Co., pp. 375–424.

Kotlewski, F., Mieszkowski, M., 1972, *Pomiary w technice cieplnej*, WNT, Warszawa.

Kukulka, D.J., Devgun, M., 2007, *Fouling Surface Finish Evaluation*, Applied Thermal Engineering, vol. 27, pp. 1165–1172.

Michiejew, M., 1953, *Zasady wymiany ciepła,* PWN, Warszawa.

NIST Reference Fluid Thermodynamic and Transport Properties – Refprop, version 8.0, National Institute of Standards and Technology, USA.

Rusowicz, A., 2004, *Analiza powstających osadów w rurach skraplacza energetycznego*, XII Sympozjum Wymiany Ciepła i Masy, Kraków, pp. 753–761.

Wajs, J., Mikielewicz, D., 2014, *Effect of Surface Roughness on Thermal-Hydraulic Characteristics of Plate Heat Exchangers*, Key Engineering Materials, vol. 597, pp. 63–74.

Webb, R.L., 1981, *The Use of Enhanced Surface Geometries in Condensers: An Overview*, [in:] Marto, P.J., Nunn, R.H., (eds.), *Power Condenser Heat Transfer Technology*, Hemisphere Publishing Co., pp. 287–324.