Open Access Journal

Journal of Power Technologies 91 (1) (2011) 41–48

journal homepage:papers.itc.pw.edu.pl

Thermal Resistance of Steam Condensation in Horizontal Tube Bundles

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Abstract

This paper presents calculated heat transfer coefficients for water vapor condensation on horizontal tubes. The influence of vapor, cooling water and noncondensable gases properties on heat transfer process are presented. Two types of condensers are analyzed. One of them is a power plant condenser; the second is an absorption refrigerator condenser where the working pair is water and Lithium Bromide (LiBr). Results for tubes selected from different places in tube banks from both types of condensers are compared.

1. Introduction

Shell and tube heat exchangers are used as absorption refrigerator condensers as well as power plant condensers. The Institute of Heat Engineering has been conducting research into shell and tube power plant condensers for many years now. One result of these investigations is a numerical model and related computer program for these types of condenser. The program simulates the behavior of steam vapor in a condenser using the Finite Elements Method (FEM). Two types of condenser will be compared in this paper: the well-known power plant condensers [1] and the absorption refrigerator condenser. At issue is whether the numerical model that has been developed can be used to simulate the work of absorption refrigerator condensers.

The heat transfer rate for a single tube can be represented by Equation 1. Total thermal resistance R is a sum of partial resistances which represent:

- water side thermal resistance R_w;
- fouling thermal resistance R_d;
- wall resistance R_p;
- condensate thermal resistance R_v;
- noncondensable gases thermal resistance R_a.

$$\dot{Q} = \frac{A}{R}(t_v - t_w) \tag{1}$$

where

$$R = \frac{d_o}{d_i}R_w + R_d + R_p + R_a + R_v \tag{2}$$

The heat transfer to water flowing inside a tube is called internal forced convection. The correlation [2] of heat transfer coefficient h_w for turbulent flow inside a tube is defined by Equation 3.

$$h_w = 0.023 \cdot \frac{k_w}{d_i} Re^{0.8} Pr^{0.4} \tag{3}$$

The fouling deposit formation has a significant impact on heat transfer in shell and tube condensers. The investigation [3] determined that fouling deposit mean heat transfer coefficient h_d in power plant condensers is h_d=25 kW/(m^{2}K). In individual pipes this value was changed from 10 kW/(m^{2}K) to 35 kW/(m^{2}K). Sometimes, in condensers without automatic cleaning, after a long period of last manual

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cleaning the value of the heat transfer coefficient decreases to 0.5 kW/(m^{2}K).

The wall resistance for tube R_p depends on thermal conductivity of pipe k_p and pipe dimensions. Equation 4 represents wall thermal resistance.

$$R_p = \frac{d_o ln\left(\frac{d_o}{d_i}\right)}{2 \cdot k_p} \tag{4}$$

The amount of noncondensable gases should be minimized. The best solution is to eliminate all noncondensable gases. In working plants it is impossible to have steam without these gases. In power plant condensers and absorption refrigerator condensers noncondensable gases consist of O_2, N_2, and H_2. To calculate heat transfer coefficient of noncondensable gases ha the correlation presented in Equation 5 [4] was used.

$$h_a = \tag{5}$$
$$a\frac{D_a}{d_o}Re^{1/2}\left(\frac{p}{p-p_s}\right)^b p^{1/3}\left(\rho_s\frac{r}{t_v}\right)^{2/3}\frac{1}{(t_v-t_c)^{1/3}}$$

where

$$a = 0.52;\ b = 0.7 \,\mathrm{for}\, Re < 350$$

$$a = 0.82;\ b = 0.6 \,\mathrm{for}\, Re \geq 350$$

In shell and tube condensers the liquid phase (condensate) fully wets the cold surface of the pipe. For this type of film condensation, the classic Nusselt correlation for the heat transfer coefficient on the steam side [5] can be used. Equation 6 represents correlation for heat transfer coefficient on steam side h_v on a single tube.

$$h_v = 0.725\frac{k_c}{d_o}\left[\frac{r' \cdot (\rho_c - \rho_s) \cdot g \cdot d_o^3}{(t_s - t_t) \cdot k_c \cdot \upsilon_c}\right]^{0.25} \tag{6}$$

During condensation on a tube bundle, the condensed water from the above tubes inundates the tubes below. The mean heat transfer $\tilde{h}_v$ coefficient of the first N rows can be expressed by Equation 7 [6], because the temperature difference between the tube wall temperature and the saturated temperature in both condenser types is under 14 K.

$$\frac{\tilde{h}_v}{h_v(N=1)} = N^{-1/6} \tag{7}$$

The heat transfer coefficient obtained from the Nusselt correlation does not take steam velocity into consideration. If in a shell and tube condenser steam is flowing with a significant value, Equations 8, 9, 10 [7] will be used to calculate heat transfer coefficient h_v for the tube bundle.

$$h_v = \sqrt{h_{sh}^2 + h_{grav}^2} \tag{8}$$

where

$$h_{sh} = 1.26\left(\frac{1}{X_{tt}}\right)^{0.78} h_L \tag{9}$$

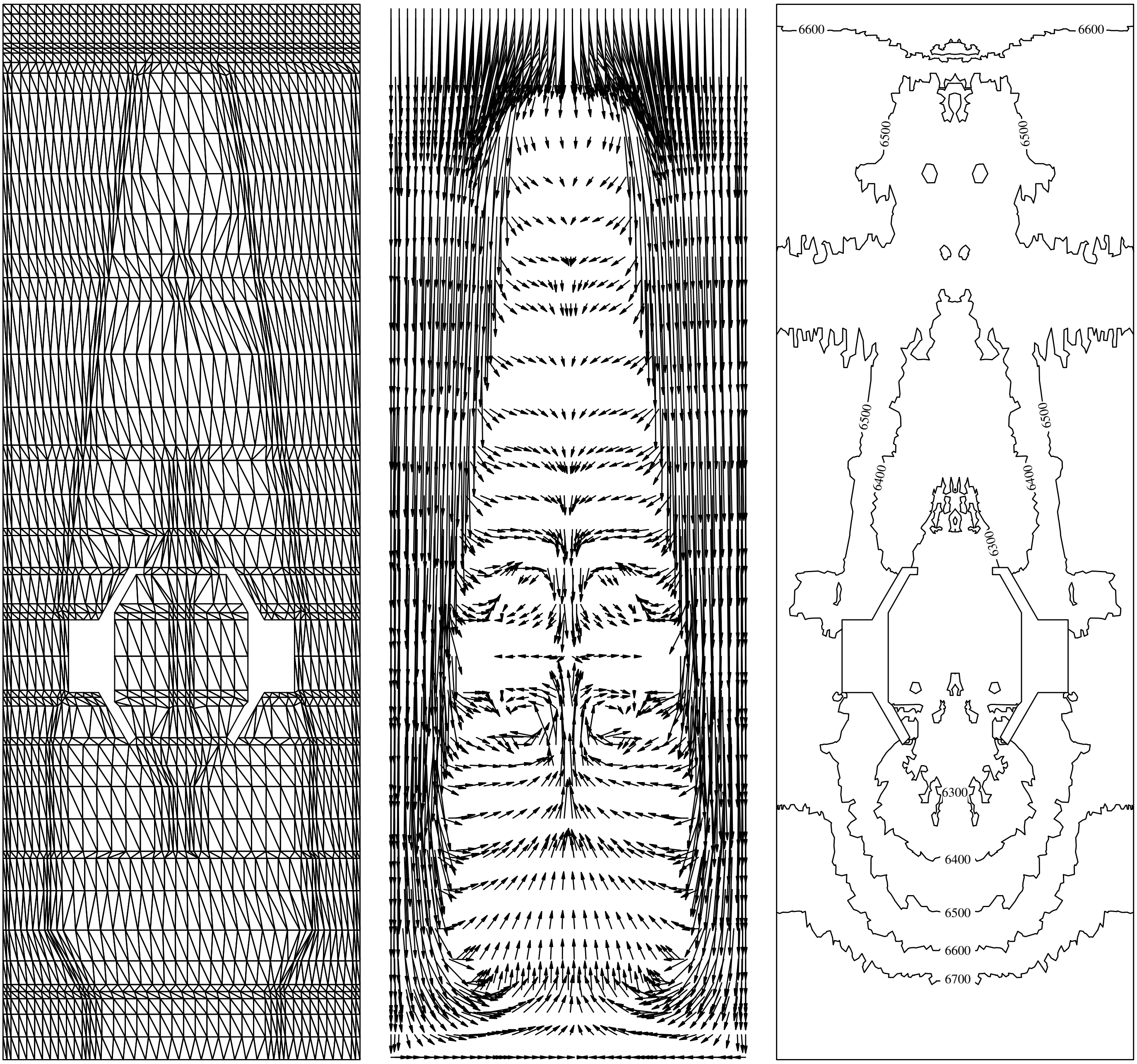
$$h_{grav} = h_v(N=1)\left[\frac{\Gamma(N-1) + \Gamma(N)}{\Gamma(N)}\right]^{\gamma} \tag{10}$$

2. Description of condensers

The power plant condenser selected here is a typical condenser operating at Belchatow, Poland. The experimental data in terms of the pressure and temperature for this condenser are available [8], so the predictions can be compared against the experimental data to validate the proposed numerical simulation [9, 10]. The grid used for the simulation are presented in Fig. 1 [11]. The velocity vector plot and contour map of steam pressure are shown in Fig. 1, too.

The absorption refrigerator condenser is a typical condenser used in one-stage absorption refrigerators where the working pair is water and Lithium Bromide (LiBr). The condenser and generator have the same working area. Cooling water at first cools the absorber and then flows into the condenser. The geometric and operating parameters for both types of condensers are given in Table 1.

Fig. 1a shows the grid used for simulation. The grid shows the tube bundle border. Fig. 1b presents steam velocity vectors. The vectors show the directions and values of steam velocity. It is possible to obtain the steam velocity direction and value on every tube. Fig. 1c represents steam pressure changes inside the condenser. In the area without tubes the



a) Grid used for the simulation b) Steam velocity vectors c) Contour map of steam pressure

Figure 1: Simulation results of condenser for power generation

Table 1: Ranges of parameters for condensers

Parameter:	Absorption refrigerator condenser	Power plant condenser
Tube outside diameter [mm]	25.4	24
Tube wall thickness [mm]	0.7	1
Tube length [mm]	6820	9940
Number of tubes	145	3220
Number of passes	1	1
C.W. Inlet Temperature [^{o}C]	33.33	22.4
C.W. Outlet Temperature [^{o}C]	36.63	30.2
Cooling water flow [kg/h]	424490	8813000
Steam inlet pressure [Pa]	7442	6620

velocity and pressure remain almost constant. In the tube bundle the pressure and steam velocity decrease significantly.

3. Results

To compare two types of condensers it is important to know what kind of properties impact heat transfer in heat exchangers. Then it is possible to compare the influence of these properties on heat transfer. These properties are:

- pipe material;
- cooling water velocity;
- cooling water inlet temperature;
- steam inlet velocity;
- tendency to fouling deposit formation;
- amount of noncondensable gases.

There are other properties (i.e. roughness) which are not discussed in this paper.

Fig. 2 shows the relationship of cooling water velocity to thermal resistance. The influence of this parameter is similar in both types of condenser. If cooling water velocity increases, waterside thermal resistance R_w will decrease.

If waterside thermal resistance R_w decreases, total thermal resistance R decreases too. Fig. 2 illustrates that steam side thermal resistance R_v increases insignificantly when velocity increases. This situation is a consequence of the gentler cooling water temperature gradient. Cooling water mean temperature is lower so condensate film thickness is larger and condensate thermal resistance increases.

The faster flow of cooling water means lower thermal resistance, but in working condensers water velocity cannot be increased *ad infinitum*, because it has other unwanted effects. When water flows faster, corrosion and pressure drop will increase. When water from the sea or a lake is used in the condenser, the velocity of cooling water should be between 1.2 and 2.4 m/s. The higher value is for cleaner water. On the other side, when velocity is low, fouling deposit formation will be faster. Optimizing water velocity is of key importance for shell and tube condensers.

Fig. 3 illustrates the relationship between cooling water inlet temperature and thermal resistance. In both types of condenser, when the inlet temperature of cooling water rises, both thermal resistances waterside R_w and steam side R_v will decrease. The thermal resistance on the steam side decreases due to the lower temperature difference between the steam and tube surface temperatures. Condensate film thickness is smaller. But this phenomenon has a very unwanted effect – the heat transfer rate decreases to a great extent. In working plants the engineer has to choose between lower thermal resistance or a lower heat transfer rate. In most known plants the inlet temperature of cooling water is 10 to 15 K lower than the steam vaporization temperature.

Fig. 4 illustrates the relationship between vapor velocity and thermal resistance. Vapor velocity has a significant influence on the heat transfer process. When velocity increases, steam side thermal resistance decreases. This phenomenon is a consequence of smaller condensate film thickness. Inflowing steam with faster velocity does not let film form a

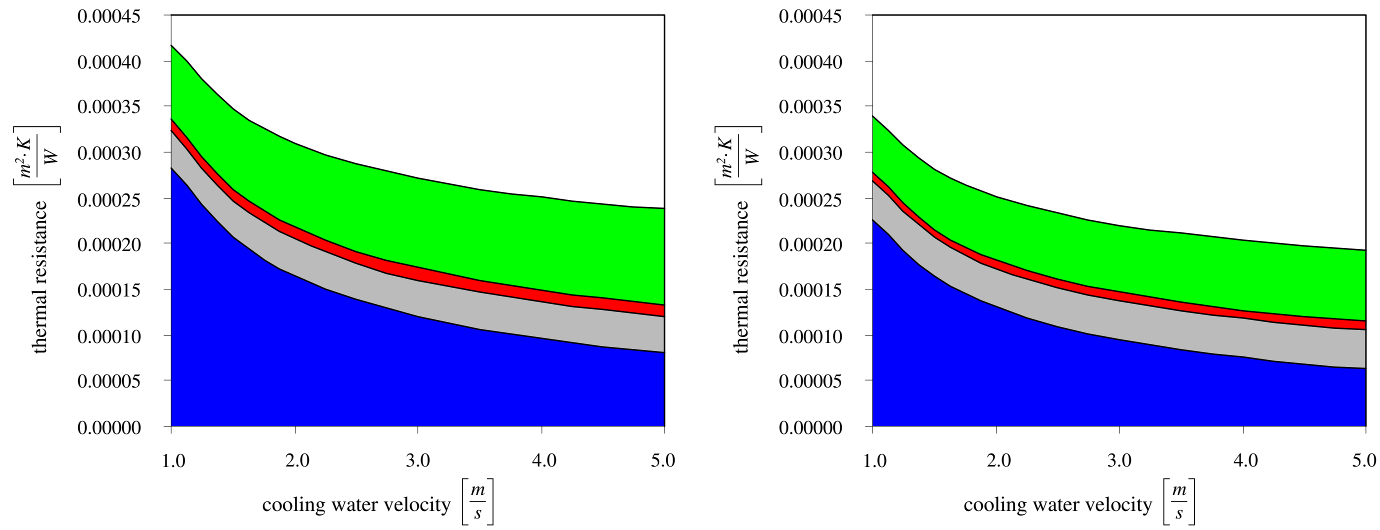
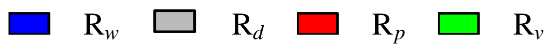


Figure 2: Relationship of cooling water velocity to thermal resistance. Absorption refrigerator condenser – left side. Power plant condenser – right side



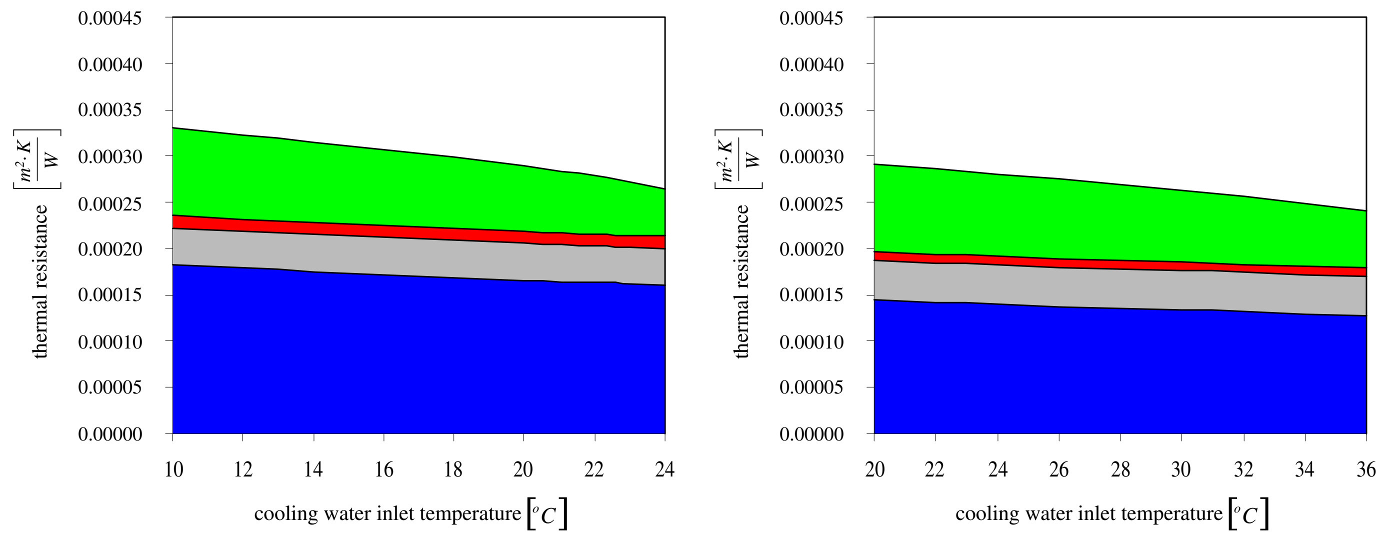
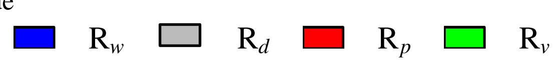


Figure 3: Relationship between cooling water inlet temperature and thermal resistance. Absorption refrigerator condenser – left side. Power plant condenser – right side



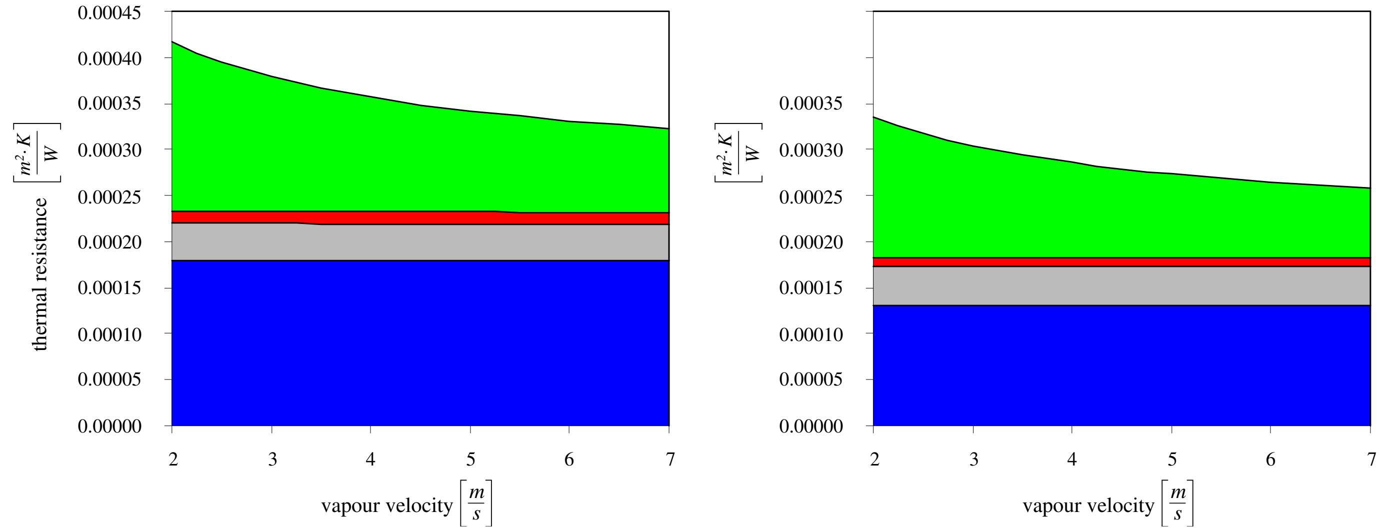


Figure 4: Relationship between vapor velocity and thermal resistance. Absorption refrigerator condenser – left side. Power plant condenser – right side

R_w R_d R_p R_v

larger thickness. In power plant condensers vapor velocity is very high (sometimes higher than 10 m/s), but in absorption refrigerator condensers this phenomenon does not occur.

Fig. 5 shows that noncondensable gases mass fraction has a very important influence on thermal resistance during the condensation process. When noncondensable gases mass fraction increases, thermal resistance will increase too. When gases mass fraction rises to 0.1 %, gases mass fraction thermal resistance will be higher than all other resistances put together. In working installations it is impermissible to let noncondensable gases mass fraction grow to this value. In an absorption refrigerator noncondensable gases mass fraction has other consequences too. This phenomenon inhibits steam production in the generator and steam absorption in the LiBr solution. The pressure in the evaporator rises. The vaporization temperature rises too. This could be caused by not obtaining the required temperature in the evaporator. Noncondensable gases mass fraction increases the corrosion process too, because of its compounds. Rust changes the chemical composition of water and accentuates the corrosion process. When noncondensable gases mass fraction is elevated in a plant, it can destroy an absorption refrigerator in the space of just a few months. Accordingly, most commercial absorption refrigerators chillers have a degasifier to decrease noncondensable gases mass fraction.

Fig. 6 illustrates the relationship between deposit thermal resistance and thermal resistance. It shows that when the fouling heat transfer coefficient decreases, total thermal resistance will go up. If the pipes are not cleaned, fouling deposit thermal resistance will soon constitute a major part of total thermal resistance. The main water compounds that increases scale formation are:

- calcium carbonate;
- calcium phosphate;
- magnesium salts;
- silica.

It is not only water compounds have an influence on deposit formation. Temperature, alkalinity, biological growth, influence of other materials impact fouling deposit formation.

4. Conclusions

The thermal resistances are similar in both types of condenser. If water velocity rises, the total thermal resistance falls. It is worthwhile bearing in mind the key role played by non-condensable gases mass fraction on the total heat transfer coefficient and on whole system performance, especially in absorption refrigerators. It is important to prepare water for use if the water is drawn from the sea or a lake. Poorly

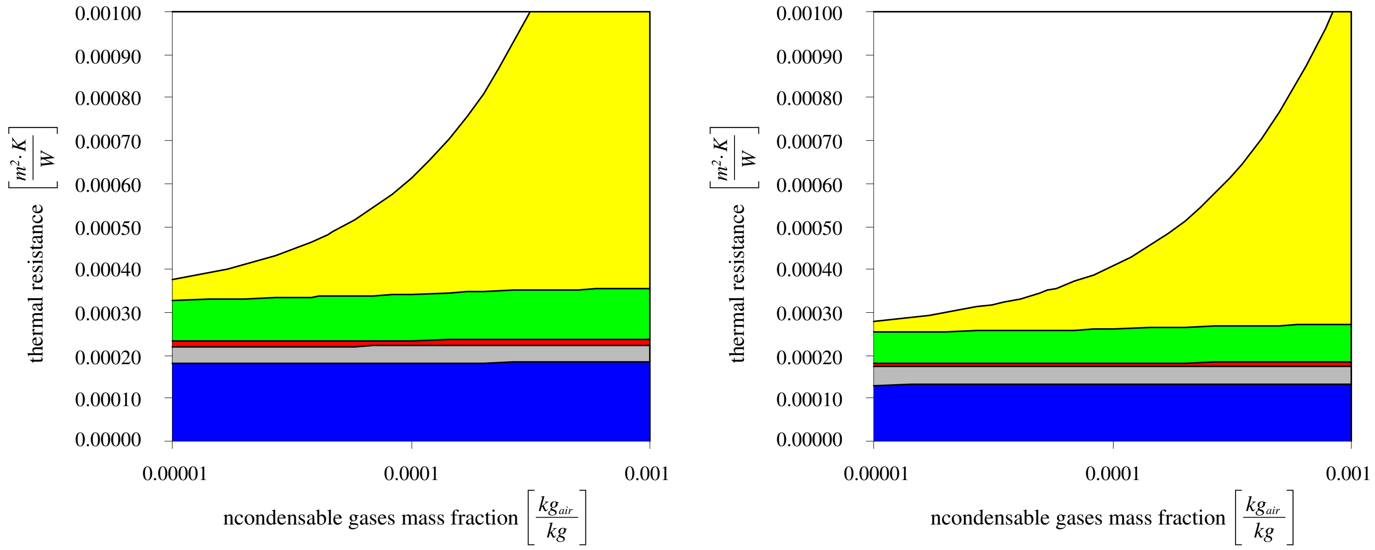


Figure 5: Relationship between noncondensable gases mass fraction and thermal resistance. Absorption refrigerator condenser – left side. Power plant condenser – right side

R_w R_d R_p R_a R_v

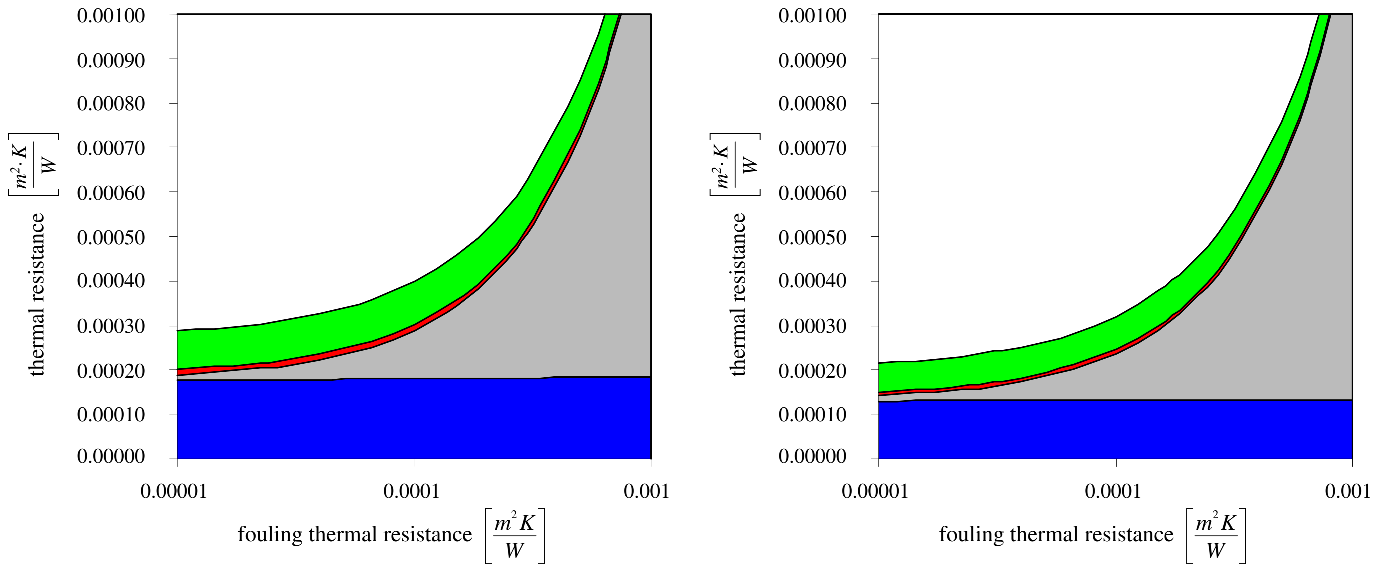


Figure 6: Relationship between fouling deposit thermal resistance and thermal resistance. Absorption refrigerator condenser – left side. Power plant condenser – right side

R_w R_d R_p R_v

prepared water potentiates the problem of fouling deposit formation to such a degree that deposit thermal resistance can become a main part of total thermal resistance alarmingly quickly.

This paper shows that both types of condensers under consideration work in very similar conditions and that the previously developed numerical model can be used equally for absorption refrigeration condensers.

References

[1] R. E. Putman, Steam surface condensers: Basic principles, performance monitoring and maintenance, ASME PRESS, NY.

[2] F. W. Dittus, L. M. K. Boelter, Heat transfer in automobile radiators of the tubular type, University of California at Berkeley, Publications in Engineering 2 (1930) 443–461.

[3] A. Rusowicz, Analysis of fouling for tubes in a power plant condenser (in Polish), XII Sympozjum Wymiany Ciepła i Masy, AGH Kraków (2004) 753–761.

[4] L. D. Berman, S. N. Fuks, Mass transfer in condensers with horizontal tube when the steam contains air, Teploenergetika Vol. 5, No.8 (1958) 66–74.

[5] W. Nusselt, Die oberflashenkondensation des wasserdampfes.

[6] D. Q. Kern, Mathematical development of loading in horizontal condensers, AIChE Journal 4 (1958) 157–160.

[7] J. M. McNaught, Two-phase forced convection heat-transfer during condensation on horizontal tube bundles, Proc. 7th International Heat Transfer Conference, Munich Vol. 5 (1982) 125–131.

[8] K. Brodowicz, A. Czaplicki, Badanie charakterystyk cieplno przepływowych kondensatora 360 MW (in Polish), Politechnika Warszawska, Instytut Techniki Cieplnej, Warszawa.

[9] M. M. Prieto, I. M. Suarez, E. Montanes, Analysis of the thermal performance of a church window steam condenser for different operational conditions using three models, Applied Thermal Engineering No.23 (2003) 163–178.

[10] I. S. Ramón, M. P. González, Numerical study of the performance of a church window tube bundle condenser, Int. J. Therm. Sci. 40 (2001) 195–204.

[11] K. Brodowicz, K. M. Ostrowski, A. Rusowicz, D. Wierzbicki, Influence of numerical viscosity on correctness of power condenser calculation (in Polish), Biuletyn Instytutu Techniki Cieplnej Politechniki Warszawskiej Vol. 80.

Nomenclature

A–heat transfer area, m^2

c_p–specific heat of condensate, J/kg/K

d_i–tube inside diameter, m

d_o–tube outside diameter, m

D_a–diffusion coefficient of air in steam, m^2/s

g–gravitational acceleration, m/s^2

h_a–noncondensable gases heat transfer coefficient, W/m^2/K

h_d–fouling heat transfer coefficient, W/m^2/K

h_{grav}–gravity controlled heat transfer coefficient, W/m^2/K

h_L–liquid film heat transfer coefficient, W/m^2/K

h_v–steam side heat transfer coefficient, W/m^2/K

$\tilde{h}_v$–mean heat transfer coefficient for tube bundle, W/m^2/K

h_{sh}–vapor shear controlled heat transfer coefficient, W/m^2/K

h_w–water side heat transfer coefficient, W/m^2/K

k_p–thermal conductivity of pipe material, W/m/K

k_c–thermal conductivity of condensate, W/m/K

k_w–thermal conductivity of water, W/m/K

L–tube length, m

N–Nth tube row in the bundle

p–total pressure, Pa

p_s–steam partial pressure, Pa

$\dot{Q}$–heat transfer rate, W

r–latent heat of condensation, J/kg

r'–effective latent heat of condensation, $r\left(1 + 0.68 \cdot \left(c_p \cdot (t_s - t_t)/r\right)\right)$

R–total thermal resistance, m^2·K/W

R_a–noncondensable gases thermal resistance, $1/h_a$

R_d–fouling thermal resistance, $1/h_d$

R_p–wall resistance, m^2·K/W

R_v–condensate thermal resistance, $1/h_v$

R_w–water side thermal resistance, $1/h_w$

Re–Re Reynolds number, $\rho_s \cdot v_s \cdot d_o/\mu_s$

R_{es}–two phase Reynolds number, $\rho_c \cdot v_s \cdot d_o/\mu_c$

x–vapor quality - vapor mass flow rate/total -mass flow rate

X_{tt}–Lockhart-Martinelli parameter, $((1-x)/x)^{0.9} \cdot (\rho_s/\rho_c)^{0.5} \cdot (\mu_c/\mu_s)^{0.1}$

γ–empirical exponent, 0.13 for triangular tube layouts, 0.22 for square tube layouts

Γ–condensation rate from one tube onto tube below, $(N \cdot \pi \cdot d_o \cdot h_v \cdot (t_s - t_t))/r$

ρ_c–condensate density, kg/m^3

ρ_s–steam density, kg/m^3

μ_s–dynamic viscosity of steam, $N \cdot s$/m^2

υ_s–kinematic viscosity of steam, m^2/s