THE DYNAMIC BEHAVIOUR OF ONCE-THROUGH COOLING WATER SYSTEMS UNDER FOULING PHENOMENA

J. N. M. Souza[1], A. R. C. Souza[2], L. Melo[3] and *A. L. H. Costa[2]
[1] ENKROTT, Rua Thilo Krassman, 7, Bloco B, Armazém 7, Abrunheira, 2710-141 Sintra, Portugal, jaime.souza@enkrott.
[2] Chemistry Institute, Rio de Janeiro State University, Rua São Francisco Xavier, 524, Maracanã, CEP 20550-900, Rio de Janeiro, RJ, Brazil.
[3] LEPABE, Department of Chemical Engineering, Faculty of Engineering, University of Porto, Rua Dr. Roberto Frias, s/n, 4200-465 Porto, Portugal, lmelo@fe.up.pt.

ABSTRACT

Fouling in thermoelectric power plant cooling systems may increase the backpressure of the steam turbine and cause deleterious effect in the thermal performance of the power plant. A complete analysis of the condenser must handle the exchanger as an integrated part of an entire cooling system, that is highly affected by the temperature and the cooling water quality parameters. The proposed model adopts a graph based network approach and includes the pump water supply, the associated pipes, and the steam condenser itself. It is able to describe the behavior of the system during a determined operation horizon where the effects of the temperature and water quality are associated to a fouling rate model due to the precipitation of calcium carbonate. The main results describe the performance decay of a hypothetical once-through cooling water system - including a horizontal condenser - due to fouling.

INTRODUCTION

The performance of a thermoelectric power plant is directly related to its cooling water system. The main task is to condense the exhaust steam from the corresponding thermodynamic cycle. Nevertheless, under certain scenarios, the formation of fouling can increase the thermal resistance and the backpressure of the steam turbine. This is particularly relevant in once-through cooling systems, where water is pumped from an outside source, flows through the heat exchangers and then is discarded to the environment.

Revealing the relevance of such systems, the literature contains several papers that investigated the modelling and simulation of steam condensers such as the model for steam condensers including the effect of in-leakage air [1]. Despite the good accuracy of the available models, the literature does not consider impact related to the quality parameters (e.g. suspended solids, pH, conductivity, hardness, etc) and conditions of the water in the condenser during an entire operation horizon, particularly involving fouling problems.

Due to the complexity of this system, an appropriate analysis of the condenser must handle it as an integrated part of an entire cooling system, which is highly affected by the cooling water behavior. For that, it must include the modelling of all network elements simultaneously and also the fouling effects [2].

The proposed model is able to describe the behavior of a once-through steam condenser system during a fixed operation horizon using a graph-based network approach to include in the analysis the

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

water supply pump, the associated pipes, and the steam condenser itself.

The effects of the temperature and water quality are associated to an inorganic fouling rate model, can describe the impact of the fouling layer on reduction the overall heat transfer coefficient, but also on increasing the flow resistance associated to the reduction of the free flow area.

INVESTIGATED SYSTEM

This paper focuses on once-through cooling water system that feeds the tube side of a large scale film condenser that receives the low pressure steam of a steam turbine as illustrated in Figure 1.

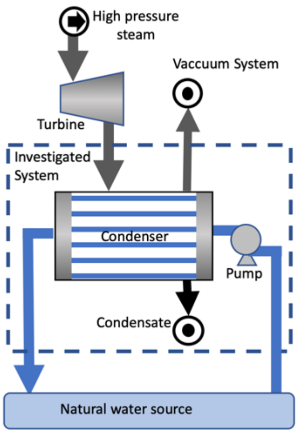

Fig. 1. Investigated system is a set of interconnected elements: (i) a natural water source point, (ii) a pump, (iii) a distribution pipe that delivers water to the tube side of a heat exchanger, (iv) a single pass low pressure saturated steam film condenser with horizontal tube bundle, (v) a collecting pipe, and (vi) a sink point where the water is discharged

In order to guarantee sufficient heat exchange area to condense large amounts of low pressure steam, the steam power plants condensers are designed with large dimensions and high number of tubes. The cooling water flows inside the tubes surrounded by condensate in the shell, where it is collected through the lower compartment of the condenser (hot well).

The vertical position of each tube row in the bundle is identified according to variable z (from top to bottom, first row corresponds to $z = 1$), that is relevant for the calculation of the external convection coefficient on each tube (Equation 2).

NETWORK MODEL

The process modeled in this paper has different pieces of equipment that are interconnected and intimately related. The heat exchanger itself is modeled as a set of tubes that have mass, momentum and heat balances calculated for time t and normalized axial position x coordinates varying from 0 to 1. In this context, the best approach to organize the model is to define a network structure using a digraph [3] composed of $|E|$ edges and $|V|$ vertice, where:

- Vertices are identified by $v_i \in V$, where i is the corresponding index, representing the network elements as: pumping supply vertex (subset SV, representing the natural water source), fixed pressure demand vertex (subset DV, representing the return point to the river or the lake), connecting vertex (subset CV, representing the point of interconnection of different elements);
- Edges are identified by $e_j \in E$, where j is the corresponding index, which represents the material streams and classified as pipes (subset PE, responsible for the transport of water from the source to the condenser and then to return the heated water to the water sink) and heat exchanger tubes (subset TE, that remove heat from the steam).

Therefore, the investigated system is described according to the digraph G in Figure 2 with a pipe edge pe_1 representing the distribution header that connects a pumping supply vertex v_1 to a connecting vertex v_2 and a pipe edge pe_2 that leaves the connecting vertex v_3 to the fixed pressure demand vertex v_4. The condenser tube side is represented as a set of heat exchanger tube edges $\{te_1,...,te_n\}$ between v_2 and v_3 vertice.

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

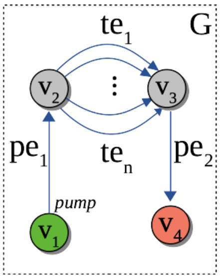

Fig. 2. Network digraph

The state variables of the model are identified according to their corresponding digraph element:

- The vertex state variables are pressure, temperature and external mass flow rate (inlet/outlet of the system).

- The edge state variables are the spatio-temporal distribution of pressure, cooling water temperature, mass flow and the diameter associated to the free flow area.

The specifications of the model were restricted to vertex variables, where the number of required specifications is equal to |E|.

The vertex model is composed of a mass balance and an energy balance, where the effect of the pressure in the energy balance is neglected.

The edge model is composed of a mass balance, a momentum balance and a heat balance. In this approach, a unique edge model adopted in this paper is employed to represent the behavior of pipes and condenser tubes, unifying the edge representation by the same mathematical structure. In other words, the same model can predict the carbonate deposit formation not only in the condenser tubes but also in the pipe sections.

In order to consider the head loss originated by pipe accidents, the model includes concentrated head losses for edges inlet and outlet. In this context, this approach can be adopted to account for pipe curves, valves, and heat exchanger nozzles and headers.

Each edge has two pressure boundary conditions - that relate the pressure of edge, the pressure of the vertice and the concentrated head loss.

The mass source term was neglected and the momentum source term describes the momentum variation of the fluid related to the viscous shear stress and the gravitational acceleration and is modeled assuming a Darcy friction factor calculated by the Churchill correlation [4].

The heat source term for pipe edges was neglected, and for heat exchanger tubes was calculated according to the model presented in the next Section.

The closure equations describing the physical properties of cooling water, condensate and steam were obtaining by the IAPWS Industrial Formulation 1997 correlations.

Heat Source Term Model

For j-th edge representing a heat exchange tube, the heat source term represents the heat transfer rate from the steam to the cooling water stream and considers internal convection, conduction across the fouling layer and the pipe wall and external convection due to steam condensation. The heat source term is defined as:

$$\Phi_{T_j} = \frac{T_{V_j} - T_j}{R_{I_j} + R_{W_j} + R_{E_j} + \frac{R''_{f_j}}{\pi D}} \tag{1}$$

Where T_V is the steam temperature, T is the cooling water temperature, D is the tube internal diameter, R''_f is the fouling resistance and R_I, R_W and R_E are the unitary length thermal resistance for the internal convection, the wall conduction and the external convection.

For the sake of simplicity, we are assuming that tube side flow regime is turbulent, the shell side flow regime is laminar and the internal convection coefficient is obtained by the correlation [5] modified by [6]. The external convection coefficient considers the effect of the tube position z is calculated by [7]:

$$h_{E_j} = \left(z_j^{5/6} - (z_j - 1)^{5/6}\right) \overline{h}_{E_j} \qquad (2)$$

where $\overline{h}_E$ is the heat transfer coefficient for external laminar film condensation of an horizontal isolated tube and is calculated in $W/(m^2K)$ based on the correlation [8] adjusted by [9]:

$$\overline{h}_{E_j}(t,x) = 0.729 \left(\frac{g\,\rho_{L_j}\left(\rho_{L_j}-\rho_{V_j}\right) k_{L_j}^3\, \Delta H_{VAP}}{\mu_{L_j}\,(T_{V_j}-T_{WO_j})\, D_{WO_j}}\right)^{1/4} \qquad (3)$$

where ρ_L, k_L and μ_L are the density, the conductivity and the dynamic viscosity of the external condensate film in kg/m^3, $W/(m\cdot K)$ and $Pa\cdot s$, respectively; T_{WO} and D_{WO} are temperature and diameter at the tube outer wall in K and m, respectively; ρ_V is the density of the saturated steam; ΔH_{VAP} is the vaporization enthalpy in J/kg; and g is the gravitational acceleration in m/s^2.

Fouling Model

Considering a deposition rate based on Kern and Seaton model [10], it is decomposed into formation φ_d and removal φ_r rates:

$$\frac{dm_f}{dt} = \varphi_d - \varphi_r \qquad (4)$$

where m_f is the mass of deposit per unit of area (kg/m^2).

The fouling resistance can be obtained from the mass of the deposit in a circular tube with internal diameter D:

$$R_f^{"} = log\left(\frac{D}{D-2m_f/\rho_f}\right)\frac{D}{2k_f} \qquad (5)$$

where $R_f^{"}$ is in m^2K/W, ρ_f is the deposit density in kg/m^3 and k_f is the thermal conductivity of the deposit in $W/(m\cdot K)$.

The cooling water fouling model adopted in this work focuses on calcium carbonate inorganic deposits. An experimental investigation of Wu and Cremaschi [11] indicated that the fraction of calcium carbonate in the fouling inorganic deposits in a cooling water system was more than 85%.

The deposition rate (φ_d) in $kg/(m^2\cdot s)$ is described by [12] based on Hasson-Quan [13]:

$$\varphi_d = kd[CO_3^{2-}]\,\frac{1-\frac{ksp}{[Ca^{2+}][CO_3^{2-}]}}{1+\frac{kd}{kr[CO_3^{2-}]}+\frac{[CO_3^{2-}]}{[Ca^{2+}]}} \qquad (6)$$

where *ksp* is the solubility product of $CaCO_3$ at reference temperature and pressure (25°C and 1 atm, respectively), $[Ca^{2+}]$ and $[CO_3^{2-}]$ are the ionic concentrations, *kr* is the precipitation rate (Equation 7) [13] and *kd* is the transport rate (Equation 8) [13].

$$\ln kr = 38{,}74 - \frac{20700}{Rg\,T_w} \qquad (7)$$

$$kd = 0{,}023\ u\ Re^{-0{,}17}\ Sc^{-0{,}67} \qquad (8)$$

where *Rg* is the ideal gas constant in $cal/(K\cdot mol)$; *T* is the cooling water temperature in K; *u* is the water velocity in m/s; *Re* is the Reynolds number; and *Sc* is the deposit Schmidt number obtained by Equation (9).

$$Sc = \frac{(\mu_w)^2}{3{,}07\cdot 10^{-15}\rho_w\,T_w} \qquad (9)$$

where μ_w is the water dynamic viscosity in Pa s, ρ_w is the water density in kg/m^3 and T_w is the water temperature in K.

The removal rate (φ_r) in $kg/(m^2\cdot s)$ can be obtained by [12]:

$$\varphi_r = \left(\frac{0{,}00212\,u^2}{{k_f}^{0{,}5}\,\psi}\right)\varphi_d \qquad (10)$$

where ψ is the mechanical resistance of the deposition, k_f is the thermal conductivity of the deposit in $W/(m\cdot K)$, and *u* is the fluid velocity in m/s.

SIMULATION METHODOLOGY

The mass, momentum and energy balances partial differential equations

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

(PDE) were solved using the center finite difference method available in DAETools [14], a free software released under the GNU General Public License.

RESULTS

The application of the proposed model is illustrated by the simulation of a hypothetical once-through cooling water system including a horizontal condenser with geometry and operational conditions similar to a typical steam power plant, as described in Section Investigated System (Figures 1 and 2 with parameters presented in Tables 1 and 2).

Table 1. Simulation parameters

Description	Nomenclature	Value
Initial steam pressure	$P_V(t = 0)$	10000 Pa
Head loss coefficient at pe_1 outlet		1.1
Head loss coefficient at pe_2 inlet		0.7
Head Loss coefficient at each tube inlet		0.45
Head Loss coefficient at each tube outlet		0.45
Number of mesh points		10
Number of tubes in the bundle		34x34
Pipe length		50 m
Pipe inner diameter	D	0.356 m
Pipe/tube inclination		0 rad
Pipe/tube roughness		0.000045 m
Shell cross section		Square
Steam volume in the shell		20 m^3
Total heat transfer area		261.51 m^2
Tube bundle layout		Square pitch
Tube length		4.572 m
Tube inner diameter	D_{WI}	0.0348 m
Tube outer diameter	D_{WO}	0.0381 m
Tube wall thermal conductivity	k_W	50 W/(m^2K)
Condenser steam input		10.95 kg/s
Water source temperature	T_1	286.95 K

Table 2. Parameters for the fouling model

Description	Value
Mechanical resistance (ψ)	0.01
Deposit thermal conductivity (k_f)	2.941 W/m K
Density of $CaCO_3$ deposit (ρ_f)	2710 kg/m³
Solubility product (ksp)	4.9E-9 mol^2/L^2
Molar fraction of CO_2 in air	3.14E-4
pH	9.55
Ca^+ concentration	1.2 mmol/L

The vertex v_1 is modeled as a pumping supply vertex with pump curve corresponding to a polynomial with the following coefficients equal to 276935 Pa, 0 Pa/(kg/s) and -0.1865 Pa/(kg/s)^2. The vertex v_4 is modeled as a fixed pressure demand vertex with pressure equal to 10^5 Pa.

For the sake of simplicity, the determination of ionic specimen concentrations in Equation 6 is not covered in this paper, but they can be obtained based on the pH, the molar fraction of carbon dioxide in air, the calcium cation concentration in Table 2 and dissociation equilibrium and Henry equilibrium equations [16].

In Figures 3-13, the dynamic simulation of the proposed model with and without fouling are compared. The initial condition considers the initial steam pressure defined in Table 1 and the solution of the steady state model forcing null value for the time derivative terms.

By starting our analysis with the steam quality, we can observe that there is an increase in the steam pressure over time (Figure 3). The fouling caused an increase in the backpressure of the steam turbine. Although this work does not include the turbine modelling, it is clearly recognized that a higher backpressure will reduce the turbine power.

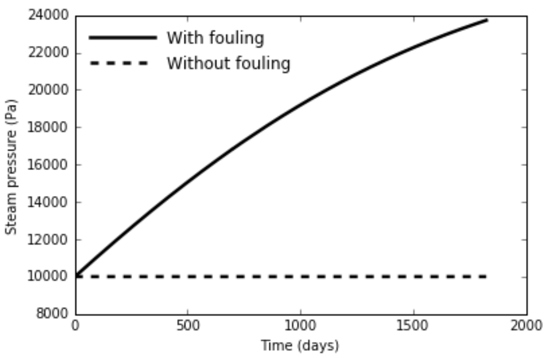

Fig. 3. Shell steam pressure

Nevertheless, we can observe that there is a secondary effect that is relevant to the heat exchange phenomena, that corresponds to the main reason to adopting a process system approach. If we assume that the It corresponds to an increase in the steam temperature caused by the

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

achievement of a different equilibrium point in the shell side (Figure 4).

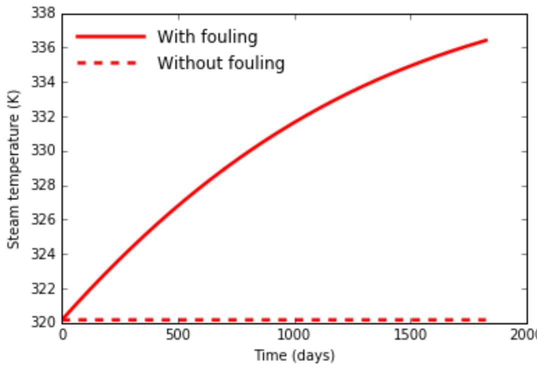

Fig. 4. Shell steam temperature

Another aspect that reinforces the necessity of considering the hydraulic system and not the isolated heat exchanger is the cooling water flowrate reduction over time (Figure 5).

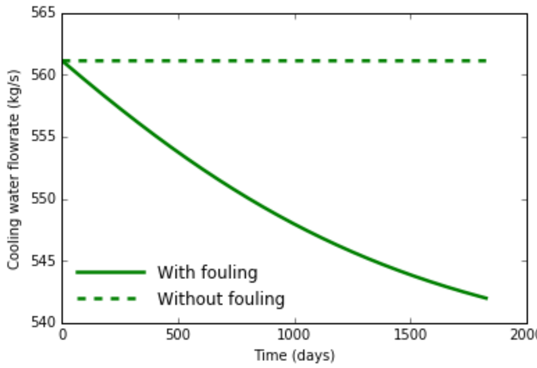

Fig. 5. Cooling water flowrate

The increase in the fouling deposit implies a reduction of the free flow area. The temporary larger flow velocity (Figure 6) - with corresponding increase in head loss - causes the system to shift to a new operational point in terms of system head loss and pump head curve.

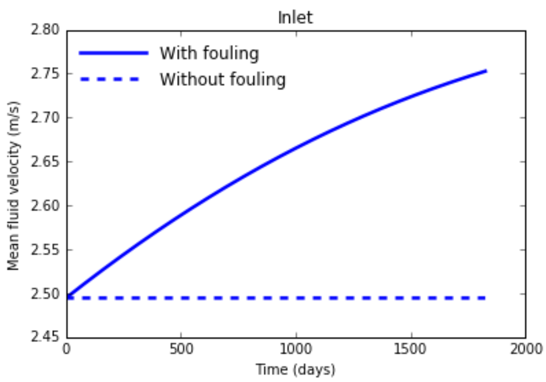

Fig. 6. Fluid velocity (mean value of the different z positions)

Now that we discussed the secondary effects, we can focus the analysis on the cooling water temperature. It is important to remember that out model uses distributed parameters along the tube axial position *x*, and also considers the tube vertical position inside the tube bundle (*z*=1 is the top position and *z*=34 is the lower position for this system).

In Figure 7, the temperature of cooling water at the outlet of a tube in the top position (*z*=1) reduces with time. Between days 720 and 770, the numerical solver produced numerical oscillations that are not expected in the modeled phenomena, but are explained by the complexity of the mathematical model and its integration over a very long time horizon.

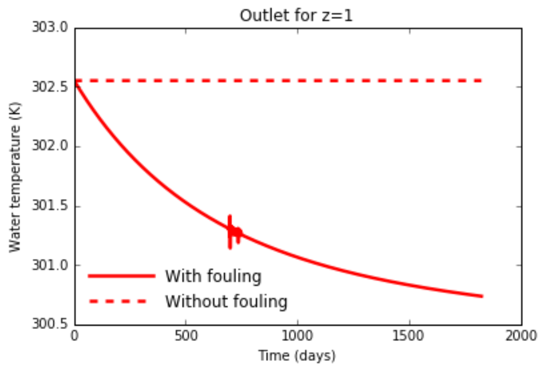

Fig. 7. Water temperature at tube outlet (z=1)

Nevertheless, Figure 8 shows that the temperature at the outlet of a tube in the lower position (*z*=34) increases with time.

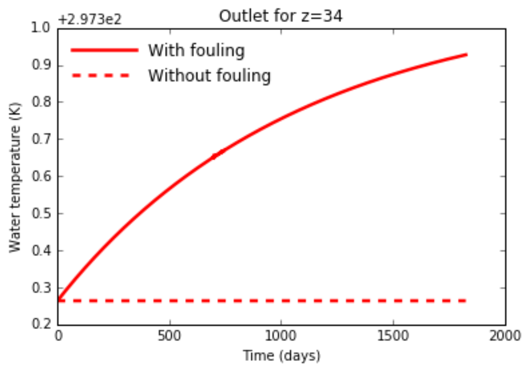

Fig. 8. Water temperature at tube outlet (z=34)

The same analysis can be observed for the profiles of cooling water temperature along the axial position as in Figure 9.

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

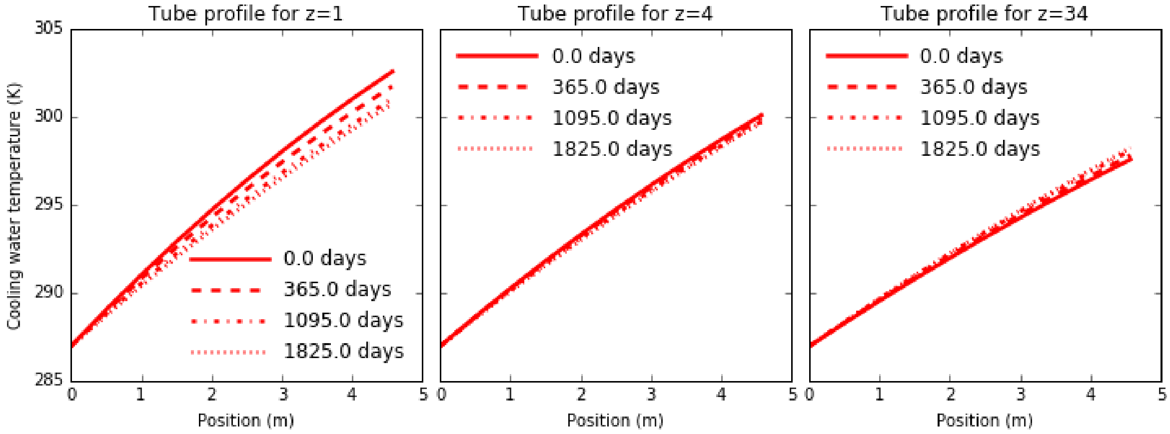

Fig. 9. Cooling water temperature profile

The fouling resistance increases with time and after 5 years of operation becomes about $0.21 \cdot 10^{-3}$ m^2K/W for the tube at z=1 (Figure 10) and $0.19 \cdot 10^{-3}$ m^2K/W for tube at z=34 (Figure 11). The profiles in Figure 12 indicates that the tube inlet presents a fouling resistance that is about 20% lower than the tube outlet.

Finally, we can observe the overall heat load of the condenser in Figure 13, where the total heat load of the system decreases due to the pressure dependence of the heat of vaporization.

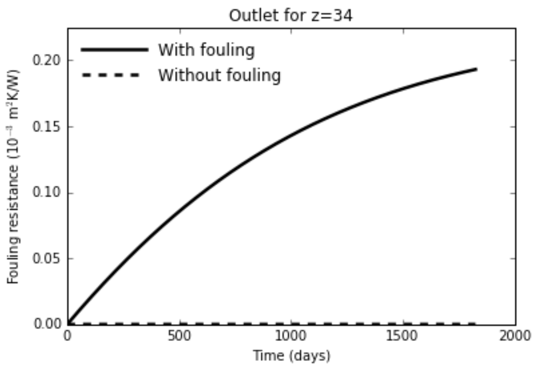

Fig. 11. Fouling resistance at tube outlet (z=34)

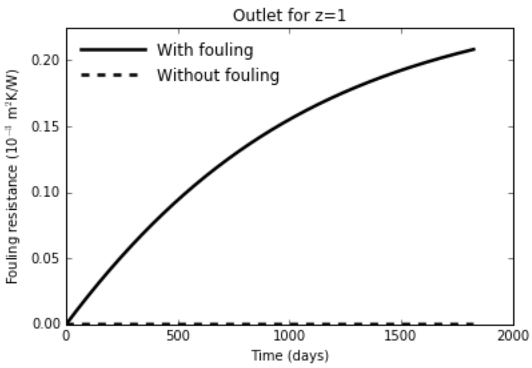

Fig. 10. Fouling resistance at tube outlet (z=1)

ISBN: 978-0-9984188-1-0; Published online www.heatexchanger-fouling.com

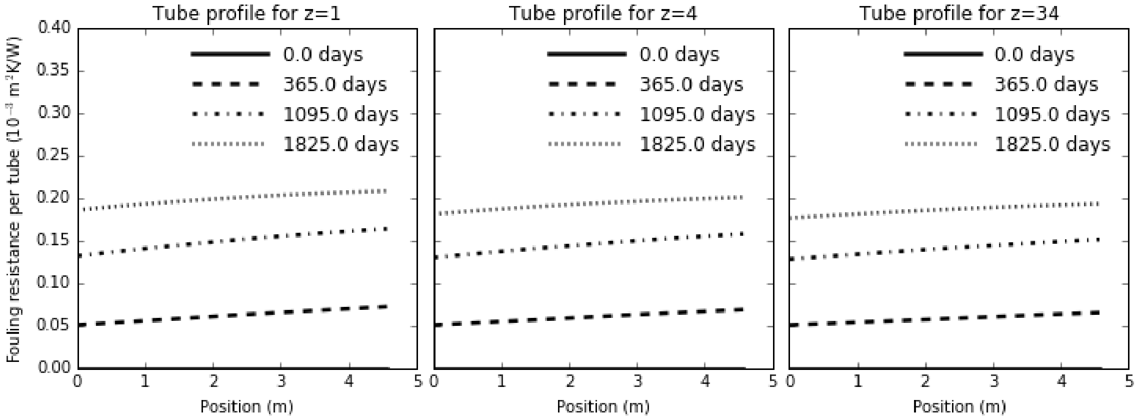

Fig. 12. Fouling resistance profiles

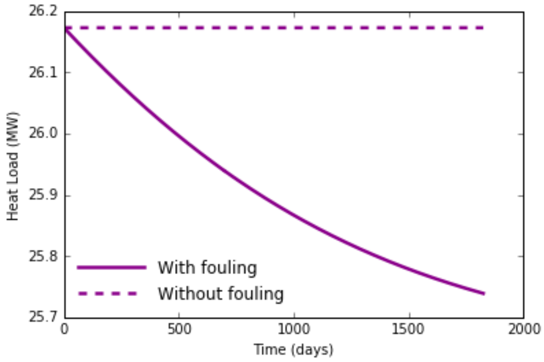

Fig. 13. Total heat load

CONCLUSION

The proposed model was able to describe the system of interest, where different complex phenomena occur and compete.

The classic adoption of rules of thumb with stationary fouling factor is a common engineering practice and is valid for different applications. Nevertheless, the adoption of a process systems approach, represented by a graph based model, can give better insights (especially when the interaction between the elements implies in competitive feedbacks).

An accurate model of the power plant cooling systems under fouling phenomena can give better insights and a more detailed description of the operational behavior of the equipment [15].

NOMENCLATURE

List all symbols used within the manuscript, their definitions, and their SI units.

ρ	Density, kg/m^3
μ	Dynamic viscosity., Pa s
φ_d	Deposit formation rate, kg/s
φ_r	Deposit removal rate, kg/s
Φ_T	Heat source term, W/m
[X]	Concentration of ion X, mol/L
cv	Connecting vertex element
CV	Connecting vertice set
dv	Demand vertex element
DV	Demand vertice set
e	Edge element
E	Edge set
G	Graph
H	Enthalpy, J/kg
h	Heat transfer coefficient, W/(m^{2}K)
kd	Transport coefficient
kr	Precipitation rate coefficient
ksp	Solubility product of $CaCO_3$, mol^2/L^2
m	Mass of deposit per area, kg/m2
pe	Pipe edge element
PE	Pipe edge set
R	Thermal resistance per length, mK/W
R"	Fouling resistance, m^{2}K/W
Re	Reynolds number
Sc	Schmidt number
sv	Supply vertex element
SV	Supply vertice set
T	Temperature, K
te	Heat exchange tube element
TE	Heat exchange tube set
u	Velocity, m/s
v	Vertice element
V	Vertex set
VAP	Vaporization
z	Tube position

Subscript

E	External
f	Fouling or deposit
I	Internal
i	Vertex index

j	Edge index
L	Condensate
V	Vapor
w	Cooling water
W	Wall

ACKNOWLEDGMENT

This work was financially supported by the POCI-01-0145-FEDER-006939 project at Laboratory for Process Engineering, Environment, Biotechnology and Energy – UID/EQU/00511/2013, funded by European Regional Development Fund (ERDF) through COMPETE2020 – Programa Operacional Competitividade e Internacionalização (POCI), and by national funds (PIDDAC) through FCT – Fundação para a Ciência e a Tecnologia/MCTES and by the project "LEPABE-2-ECO-INNOVATION" -- NORTE-01-0145-FEDER-000005, funded by Norte Portugal Regional Operational Programme (NORTE 2020), under PORTUGAL 2020 Partnership Agreement, through the European Regional Development Fund (ERDF). A L. H. Costa thanks UERJ – the Rio de Janeiro State University, and CNPq – the National Council for Scientific and Technological Development, for the research productivity fellowship (Process 311225/2016-0) and by the financial support through the Prociência Program.

REFERENCES

[1] R. P. Roy, M. Ratisher, V. K. Gokhale, A Computational Model of a Power Plant Steam Condenser, Journal of Energy Resources Technology, 123 (2001) 81.

[2] A. R. Souza, A. L. Costa, Modelling and Simulation of Cooling Water Systems Subjected to Fouling, Chemical Engineering Research and Design, 141 (2019) 15-31.

[3] R. S. H. Mah, Chemical Process Structures and Information Flows, Butterworths, 1990.

[4] S. W. Churchill, Friction-factor Equation Spans All Fluid-Flow Regimes 84 (1977) 91-92.

[5] B. Petukhov, Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties, Advances in Heat Transfer 6 (1970) 503-564.

[6] V. Gnielinski, New Equations for Heat and Mass Transfer in the Turbulent Flow in Pipes and Channels, (Jahrestre_en der Verfahrensingenieure, Berlin, West Germany, Oct. 2-4, 1973.) Forschung im Ingenieurwesen, vol. 41, no. 1, 1975, p. 8-16. In German. 75 (1975) 8-16.

[7] D. Q. Kern, Mathematical Development of Tube Loading in Horizontal Condensers, AIChE Journal 4 (1958) 157-160.

[8] W. Nusselt, Die Ober a Chenkondensation des Wasserdampfes, 1916.

[9] V. Dhir, J. Lienhard, Laminar Film Condensation on Plane and Axisymmetric Bodies in Nonuniform Gravity, Journal of Heat Transfer 93 (1971) 97.

[10] D. Q. Kern, R. E. A. Seaton, Theoretical Analysis of Thermal Surface Fouling. *British Chemical Engineering*, v. 4, n. 5, p. 258-262.

[11] X. Wu, L. Cremaschi, Thermal and Chemical Analysis of Fouling Phenomenon in Condensers for Cooling Tower Applications. *International Refrigeration and Air Conditioning Conference at Purdue*, n. 2198, p. 1-10, 2012.

[12] X. Wu, L. Cremaschi, Effect of Fouling on the Thermal Performance of Condensers and on the Water Consumption in Cooling Tower Systems. *Proceeding of International Conference on Heat Exchanger Fouling and Cleaning*, p. 134-141, 2013.

[13] Z. H. Quan, Y. C. Chen, C. F. Ma, Heat Mass Transfer Model of Fouling Process of Calcium Carbonate on Heat Transfer Surface. *Science in China*

Series E: Technological Sciences, v. 51, n. 7, p. 882-889, 2008.

[14] D. D. Nikolic, DAE Tools: Equation-Based Object-Oriented Modelling, Simulation and Optimisation Software, PeerJ Computer Science 2 (2016) e54.

[15] C. Borges de Carvalho, E. P. Carvalho, M. A. S. S. Ravagnani, Tuning Strategies for Overcoming Fouling Effects in Proportional Integral Derivative Controlled Heat Exchangers, *Industrial & Engineering Chemistry Research* 57 (2018) 10518-10527.

[16] J. M Smith, H. Van Ness; M. Abbott, Introduction to Chemical Engineering Thermodynamics, 6th edition, The Mcgraw-Hill Chemical Engineering Series, 2000.