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# The Influence of Fouling Build-up in Condenser Tubes on Power Generated by a Condensing Turbine

Slawomir Alabrudzinski*, Mariusz Markowski, Marian Trafczynski, Krzysztof Urbaniec

Warsaw University of Technology - Plock Campus, Department of Process Equipment, Jachowicza 2/4, 09-402 Plock
slawomir.alabrudzinski@wp.pl

After years of operation of a power-block condenser, even in systems equipped with common tube-side cleaning systems, the fouling build-up rate may increase and affect condensing turbine overall performance. In this paper, a CFD based estimation method was employed to study the effect of the thermal resistance of fouling on the power output of a condensing turbine unit. For old condenser tubes, a significant decrease in the power generated by the condensing turbine was observed. Compared to new tubes, the old ones have to be cleaned more frequently (up to 6 times) to maintain a power drop not exceeding 1 %. Correlation between fouling resistance and condenser inlet pressure was investigated numerically by adopting the continuum (porous zone) approach to condenser modelling. The estimation is based on experimentally determined literature data on the fouling build-up in a real power-block condenser. For very old condensers, fouling resistance after one year of operation (with tube-side fluid being seawater) may reach 0.0007 $m^2K/W$. It was shown that such values of the fouling resistance result in turbine power drop up to 4.1 %, whereas with new condenser tubes, reasonable values of the heat transfer coefficient can be maintained for 6 months and after one year of operation, turbine power drop does not exceed 1.5 %.

## 1. Introduction

In the steam circuit of a thermal power plant, steam generated in a high-pressure steam boiler flows through a turbine and is subsequently condensed in a water-cooled surface condenser. As the condensation pressure, and more specifically, condenser inlet pressure influences steam pressure at the turbine outlet, it has a direct effect on the power generated by the turbine. The condenser inlet pressure depends on various design parameters which determine the overall efficiency of the apparatus (Chuang and Sue, 2005). The overall performance of the apparatus is also affected by operational parameters, which may be divided into subgroups: medium time-scale, for example the cooling water temperature fluctuations due to climate change were taken into account by Attia (2015), and long time-scale effects (including the increase of thermal resistance of tube fouling and partial pressure of air present in the steam). These parameters significantly affect the condensing turbine performance and cause economic losses for the plant operator (Putman and Harpster, 2002). In order to prevent excessive fouling build-up, many power-plant condensers are periodically cleaned using conventional cleaning procedures (e.g. brush or acid) or specialized cleaning systems (e.g., sponge balls or blowing plugs) and therefore, have to be monitored in order to determine optimum cleaning cycles. However, in systems with tube-side fluid being seawater, due to strong corrosive nature of salt water, the rate of fouling build-up may significantly increase with time.

The long time-scale change in fouling build-up rate has been confirmed experimentally by Rabas and Elliott (1993). Kukulka et al. (2004) have introduced a specialized monitoring device to alarm the power plant operator when a critical point of biofilm build-up is reached. Many fouling build-up prediction models have been developed (Nebot et al., 2007) and used to optimize the cleaning schedule of the equipment. Markowski and Urbaniec (2005) have analyzed the phenomena of fouling build-up in HEN and indicated that thermal resistance of deposits may increase for non-equal periods between cleaning interventions.

For over 20 y, CFD based approach to simulate steam condensation in surface condensers has been developed. Brodowicz et al. (1995) have introduced the 2D numerical model of condenser operation with

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fouling, which was used later for optimizing the performance of existing power-block condensers. Zhang and Bokil (1997) have developed a quasi-3D numerical model to simulate shell-side two-phase fluid flow and heat transfer. Comprehensive 3D numerical model was proposed by Mirzabeygi and Zhang (2015) and used to analyze the fluid flow and heat transfer in small and large-scale condensers. Two and three-dimensional numerical modelling is successfully used to estimate and improve steam flow distribution and overall heat transfer coefficient (Zeng et al., 2012), and to increase the overall performance of the apparatus (Rusowicz et al., 2016). Two crucial phenomena affecting condenser performance (air leakage and build-up of fouling) are taken into account in all of these models.
In the literature, various models have been used to investigate how fouling build-up influences power generated by the steam turbine (Chuang and Sue, 2005). In the present work, a CFD based approach was used to answer this question and to study long-term changes in steam turbine performance during the period between cleaning interventions.

## 2. Numerical model

### 2.1 Geometry and meshing

The complex geometry of heat exchangers equipped with a large number of bundled tubes is commonly modelled by adopting the continuum (or porous zone) approach. In the present research, the overall condenser geometry including its improved tube arrangement was assumed after Zeng et al. (2012); however, the 2D domain was meshed (Figure 1) with 453,488 cells and 228,036 nodes. Grid independence test was carried out by studying the influence of mesh size on the resulting condenser inlet pressure (Table 1).

*Table 1: Results of the grid independence test*

| Cells | Nodes | Inlet pressure [Pa] |
|---|---|---|
| 113,372 | 57,324 | 5,349 |
| 453,488 | 228,036 | 5,375 |
| 1,813,952 | 909,576 | 5,375 |

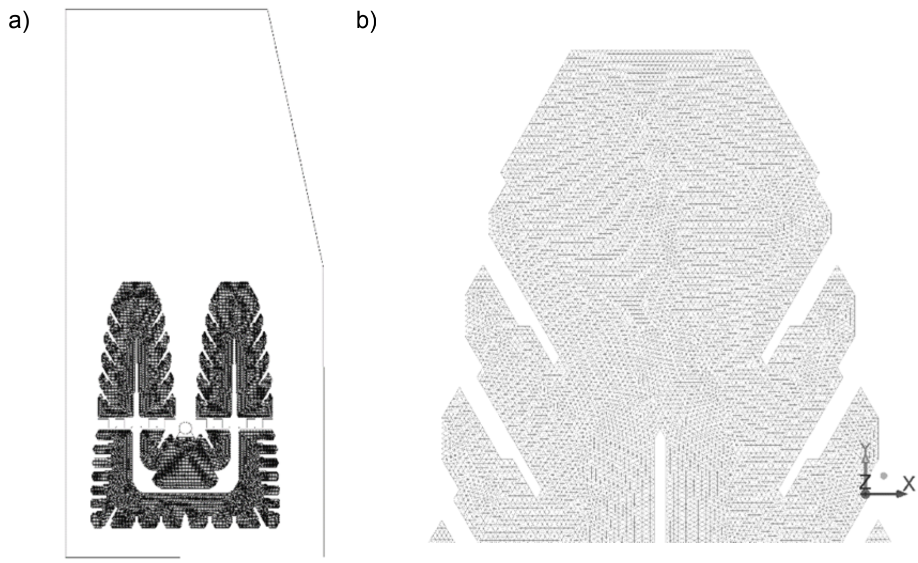


*Figure 1: Finite volumes mesh: a) tube bundle geometry, b) the mesh generated for a part of the tube-bundle area*

### 2.2 Numerical model

The following simplifying assumptions were adopted:
a) Tube thermal resistance and thickness are negligibly small,
b) No inert gases are present,
c) Condenser operates in a steady state,
d) Steam at condenser inlet is dry saturated,
e) Inlet profile of steam velocity is uniform.

The equation of mass conservation:

$$\nabla \cdot (\beta \rho_v \vec{v}_v) = \dot{m} \tag{1}$$

$$\dot{m} = -\frac{h \cdot A \cdot (T_v - T_w)}{LV} \tag{2}$$

The equation of momentum conservation:

$$\nabla \cdot (\beta \rho_v \vec{v}_v \vec{v}_v) = -\beta \nabla p + \nabla \cdot \bar{\bar{\tau}}_v + \vec{F}_r - \dot{m} \cdot \vec{v}_v \tag{3}$$

$$\bar{\bar{\tau}}_v = \beta \mu_v \left( \nabla \vec{v}_v + \nabla \vec{v}_v^T \right) + \beta \left( \lambda_v - \frac{2}{3} \mu_v \right) \nabla \cdot \vec{v}_v \bar{\bar{I}}, \tag{4}$$

where $\mu_v$ and $\lambda_v$ are the shear and bulk viscosities of steam.
Flow resistance is given as:

$$\vec{F}_r = -Re^{0.821} \cdot \theta \cdot \frac{\mu_v}{d_o^2} \cdot \vec{v}_v, \tag{5}$$

where $\theta$ is an anisotropic correction factor for including condensate-induced flow resistance.

The local heat transfer coefficient across condenser tubes was calculated taking account of the thermal resistances of shell side $R_v$, fouling $R_f$ and cooling water side $R_w$:

$$\frac{1}{h} = R = R_w + R_f + R_v \tag{6}$$

The modified Nusselt shell-side heat transfer coefficient (Brodowicz et al., 1995) was calculated as:

$$h_v = \frac{1}{R_v} = 0.725 \cdot \left[ \frac{\lambda_v^3 \cdot \rho_v^2 \cdot L \cdot g}{\mu \cdot d_o \cdot (T_v - T_t)} \right]^{0.25} \cdot \phi, \tag{7}$$

where $\phi$ is the correction coefficient to account for the effects of condensate layer thermal resistance.
The value of the water-side heat transfer coefficient was determined from the formula previously applied by Zhang and Bokil (1997) and more recently by Rusowicz et al. (2016):

$$h_w = \frac{1}{R_w} = 0.023 \cdot \frac{\lambda_w}{d_i} \cdot Re_w^{0.8}\, Pr^{0.4} \tag{8}$$

The above model was implemented in ANSYS Fluent 14.5 using its UDF (User Defined Functions) feature. Numerical simulations were carried out using coupled solver for pressure–velocity coupling, QUICK discretization scheme for all the equations, convergence criterion of $10^{-5}$ and k-ε turbulence model.

## 3. Results and discussion

A number of numerical simulations were carried out assuming turbine operating conditions listed in Table 2. For each simulation, different values of fouling thermal resistance were used in accordance with the results of fouling build-up measurements available in the literature (Rabas and Elliott, 1993).

*Table 2: Geometric and operating parameters of the condensing turbine unit*

| | |
|---|---|
| Tube-side coolant | seawater |
| Average cooling water temperature (°C) | 30 |
| Cooling water velocity (m $s^{-1}$) | 1.793 |
| Tube bundle zone volume ($m^3$) | 216.86 |
| Condenser total volume ($m^3$) | 976.01 |
| Condenser total heat duty (MW) | 300 |

**3.1 Condenser inlet pressure**

The goal of the simulation exercise was to obtain desired condenser heat duty (300 MW) by modifying its inlet pressure. In all the solved cases, the resulting steam condensation rate was equal to 123.81 kg/s.

Examples of the velocity-field map and distribution of streamlines determined for $R_f$ = 1.14 ×·10$^{-4}$ m$^2$kW$^{-1}$ are shown in Figure 2. In Figure 3, the corresponding distribution of the water-side heat transfer coefficient is compared with that determined for zero fouling.

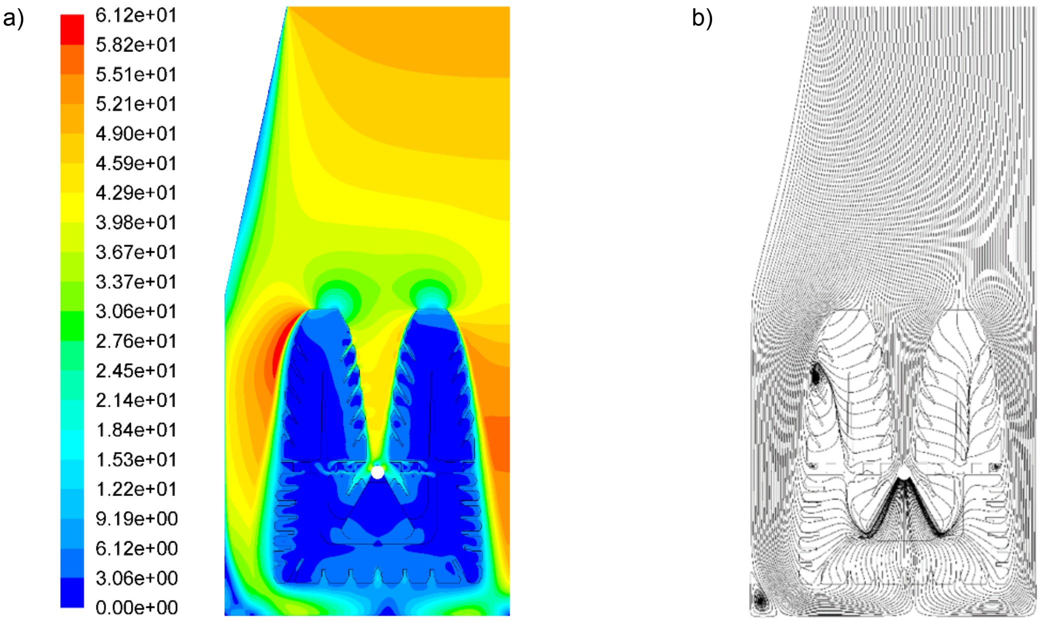


*Figure 2: Example of simulation results obtained for $R_f$ = 1.14·× 10$^{-4}$ m$^2$KW$^{-1}$:a) velocity-field map, b) streamlines*

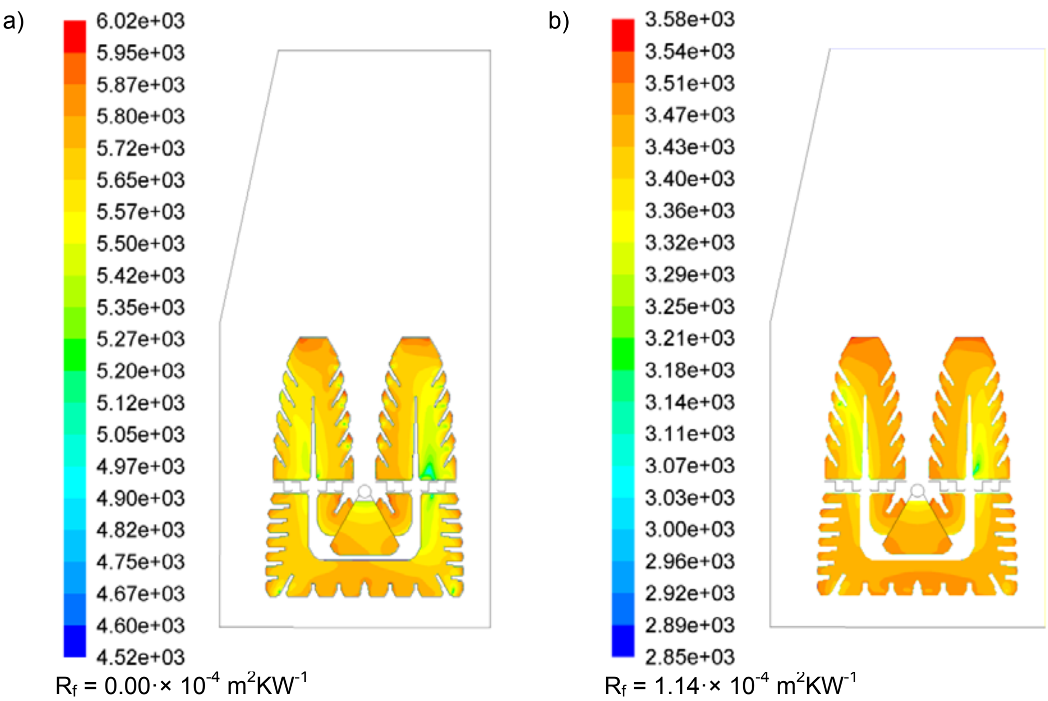


*Figure 3: Water-side heat transfer coefficient [Wm$^{-2}$K$^{-1}$]: a) no fouling, b) acceptable fouling resistance*

**3.2 Turbine power decrease with time of the operation period**

The power generated by the steam turbine was estimated using ASPEN HYSYS 8.6 turbine expander model. Constant values of the operating parameters listed in Table 3 were assumed. The values of turbine outlet pressure were set equal to the condenser inlet pressure $p(R_f)$ obtained from numerical simulations.

*Table 3: Steam turbine operating parameters*

| | |
|---|---|
| Inlet pressure (kPa) | 14,080 |
| Inlet temperature (°C) | 540 |
| Adiabatic efficiency (%) | 82 |

In Figure 4, the resulting values of condensing turbine output (power generated) are shown as functions of the time of operation (between cleaning interventions). Two plotted curves represent new clean tubes and 40 years old tubes used immediately after cleaning.
The overall changes in condensing turbine output, for new and 40 y old tubes are shown in Table 4.

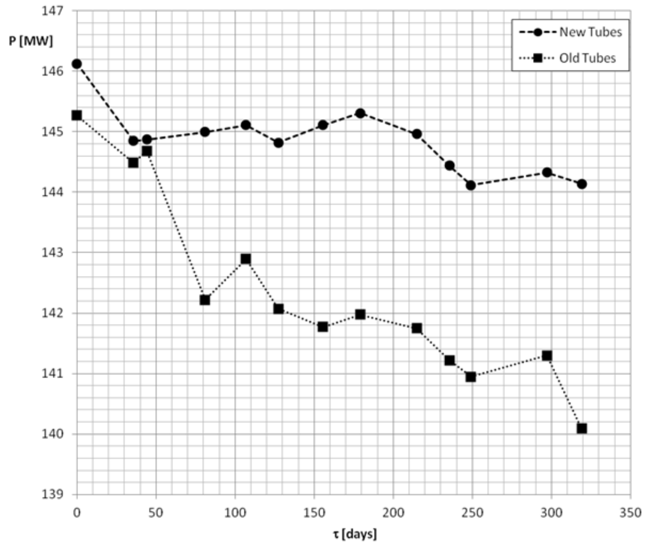


*Figure 4: The influence of fouling build-up time on power generated by the condensing turbine*

*Table 4: Changes in condensing turbine output due to fouling build-up*

| Operation time (d) | New vs. New (%) | Old vs. Old (%) | Old vs. New (%) |
|---|---|---|---|
| 35 | -0.9 | -0.5 | -1.1 |
| 80 | -0.8 | -2.1 | -2.7 |
| 179 | -0.6 | -2.3 | -2.8 |
| 319 | -1.4 | -3.6 | -4.1 |

The results of estimation of turbine power match with observations of Rabas and Elliott (1993), who reported that due to significantly greater fouling build-up rate, 40 y old tubes have to be cleaned with 28 days interval. As it was shown in Table 4, this specific cleaning interval corresponds to turbine power drop not exceeding 1 %. Applying the same criterion to new tubes, the cleaning interval of 6 months would be recommended.
It can be added that in extreme cases, frequent cleaning interventions and significant decrease in overall condensing turbine performance may result in permanent economic losses. Under such circumstances, retubing would be recommended for the plant management.

## 4. Conclusions

The effect of time-dependent increase in thermal resistance in the condenser on power generated by steam turbine is not negligible. Therefore, even if the apparatus is equipped with cleaning systems, monitoring of fouling build-up rate is necessary especially for older power-block condensers.

The proposed estimation method of condensing turbine power is presented as a tool for power plant operator. Economically justified cleaning schedule can be determined basing on a specific value of maximum economically accepted power drop and periodically measured fouling build-up rate. Alternatively, a model-based prediction of the build-up of fouling can be used.

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**Nomenclature**

$A$ - heat transfer area ($m^2$)
$T$ - temperature (°C)
$d_o$ - tube diameter (m)
$h$ - heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
$L$ - latent heat of condensation ($J\ kg^{-1}$)
$\dot{m}$ - condensation rate per unit volume ($kg\ m^{-3}\ s^{-1}$)
P - power generated by steam turbine (MW)
$p$ - absolute condenser inlet and turbine outlet pressure (Pa)
$R$ - thermal resistance ($m^2\ K\ W^{-1}$)
$V$ - volume occupied by tube bundle ($m^3$)
$\beta$ - local volume porosity
$\lambda$ - thermal conductivity ($W\ m^{-1}\ K^{-1}$)
$\mu$ - dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
$\rho$ - density ($kg\ m^{-3}$)
$\vec{v}$ - velocity ($m\ s^{-1}$)

Subscripts: v - vapour-side, t – tube, f – fouling, w - cooling water-side

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