

Numerical modelling of series-parallel cooling systems in power plant

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Abstract. The paper presents a mathematical model allowing one to study series-parallel hydraulic systems like, e.g., the cooling system of a power boiler's auxiliary devices or a closed cooling system including condensers and cooling towers. The analytical approach is based on a set of non-linear algebraic equations solved using numerical techniques. As a result of the iterative process, a set of volumetric flow rates of water through all the branches of the investigated hydraulic system is obtained. The calculations indicate the influence of changes in the pipeline's geometrical parameters on the total cooling water flow rate in the analysed installation. Such an approach makes it possible to analyse different variants of the modernization of the studied systems, as well as allowing for the indication of its critical elements. Basing on these results, an investor can choose the optimal variant of the reconstruction of the installation from the economic point of view. As examples of such a calculation, two hydraulic installations are described. One is a boiler auxiliary cooling installation including two screw ash coolers. The other is a closed cooling system consisting of cooling towers and condensers.

1 Introduction

In the current economic situation, the flexible work of a power unit requires a quick reaction to the power demand of the electrical grid. The market situation often forces the boiler to run with maximum thermal output, which also entails the work of hydraulic systems with maximum efficiency. Despite the fact that the daily consumption of fresh water by a professional power plant can reach the level of several thousand cubic meters, there are situations when the amount of water circulated in cooling systems is insufficient. This is usually observed in transient stages or periods when, e.g., the power unit works with variable thermal output or the ambient temperature is very high, making efficient heat exchange in cooling towers difficult. Such situations also often arise from the fact that the efficiency of coal-fired power plants gradually decreases with age [1].

In response, the Environmental Protection Agency (EPA) alerts that power plants are one of the largest consumers of natural water reservoirs in the USA, recommending, in September 2013, new standards for control of carbon dioxide (CO₂) emissions from new

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power units burning fossil fuels and regulations concerning the protection of water resources. It is also worth mentioning that the deterioration of the overall power unit efficiency results in the greater consumption of fossil fuels and consequently higher CO₂ emission (as well as that of other pollutants) [2]. Keeping in mind that the improvement of heat exchange processes could have a direct influence on the overall efficiency of a power unit, Ryabchikov *et al.* in [3] suggested a number of specific actions, among which the retrofit of cooling water installations is one of the most important.

An investigation of hydraulic installations indicates that most of them are designed to work under nominal parameters which do not take into account the cooling performance reserves [4, 5]. Later, during the long period of their utilization, the gradual deterioration of their operating parameters is observed, resulting from, e.g., changes in the operating characteristics of force pumps or the contamination of pipelines and heat exchanger surfaces by mineral deposits. Other important factors affecting the operating parameters of hydraulic systems are connected with various minor losses caused by installed valves and other fittings. In this situation, either the power unit will not reach its maximum nominal thermal output under the standard operational conditions or the hydraulic system must be modernized. Nichols *et al.* in [6] estimated that the modernization of cooling system performance in American power plants could lead to improving their potential efficiency by about 0.2-1.0 %. Similar estimations made for APEC countries by Boncimino *et al.* in [7] indicated that the improvement of feed water heaters and condensers could update the overall power unit efficiency by a value of approximately 0.8 %.

The modernization of an existing cooling installation is usually an expensive solution. Therefore, such a decision should be preceded by a careful analysis of various variants of the prospective reconstruction. Because hydraulic installations are usually complicated systems consisting of many series-parallel branches, their proper investigation should be based both on the measurements of the actual operating parameters and the development of a mathematical model of flow rates in individual pipelines. The analytical approach establishes a set of non-linear algebraic equations which can be solved with numerical techniques. As a result of the iterative process, a set of volumetric flow rates of water through all branches of the investigated hydraulic system is obtained. Calculations indicate the influence of changes in the pipeline geometrical parameters on the total cooling water flow rate in the analysed installation. Such an approach makes it possible to analyse different variants of the modernization of the studied system as well as enabling the indication of its critical elements, together with the directions of the flow in all branches.

2 Mathematical model of a series-parallel cooling system

The fundamental issue in modelling hydraulic installations is connected with the calculation of volumetric flow rates in each branch of the system. The scheme of the simplest series-parallel hydraulic system is presented in Fig. 1.

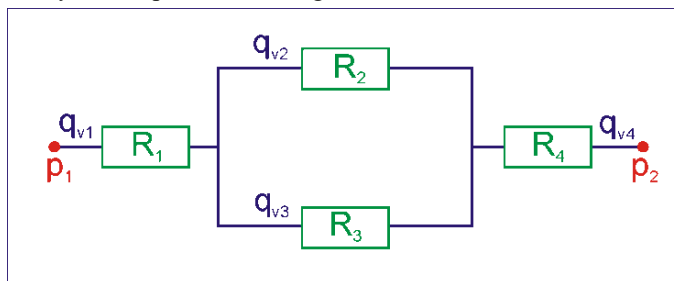


Fig. 1. Scheme of the simplest series-parallel hydraulic system (q_v – volumetric flow rate, R - flow resistance, p – absolute pressure)

Due to the fact that water installations are usually complicated systems consisting of many series-parallel branches, the mathematical modelling of a hydraulic system leads to a set of non-linear algebraic equations. In the model, the following three types of equations can appear (the examples of equations given below conform to the scheme presented in Fig. 1):

- an equation which describes volumetric flow rate conservation law in an individual joint occurring in the system (e.g., $q_{v1} = q_{v2} + q_{v3}$ and $q_{v4} = q_{v2} + q_{v3}$),
- an equation which describes pressure losses in an individual closed loop of the hydraulic system (e.g. $R_2 \cdot q_{v2}^2 - R_3 \cdot q_{v3}^2 = 0$). In the mesh, a “+” or “-” sign is taken when the flow in the considered branch is clockwise or counter-clockwise, respectively),
- an equation which describes a pressure drop Δp along a selected path connecting the inlet and the outlet of the system (e.g. $\Delta p_{12} = (p_1 - p_2) = R_1 \cdot q_{v1}^2 + R_3 \cdot q_{v3}^2 + R_4 \cdot q_{v4}^2$). In this case, knowledge of the inlet and outlet pressure (p_1 and p_2 , respectively) is required.

For example, the scheme presented in Fig. 1 requires four algebraic equations (listed above), which allow one to calculate 4 unknown volumetric flow rates: q_{v1} , q_{v2} , q_{v3} and q_{v4} .

The mathematical model also requires information about flow resistances R_i , which have to be assigned to each branch of the system (schematically marked by green boxes in Fig. 1). The flow resistance of an individual branch of the installation (denoted by index “ i ”) is calculated as the sum of local minor and friction losses:

$$R_i = \frac{8 \rho}{\pi^2 d_i^4} \left(\sum_k \xi_{i,k} + \lambda_i \frac{l_i}{d_i} \right) \tag{1}$$

where: $\xi_{i,k}$ – coefficients of minor losses and λ_i – coefficient of friction losses which occur on “ i ” branch; d_i and l_i – diameter and length of the “ i ” branch, respectively [8]. The values of ξ coefficients are usually selected basing on literature data or measurements. The values of λ coefficients are calculated from Blasius’ formula in the range of turbulent flow [8, 9]. It is worth mentioning that the formula for λ includes information about q_v , which is, at the beginning, an unknown value. Therefore, in the process of solving, the values of λ_i (and hence flow resistances R_i) must be updated before each iterative step. For this reason, the set of non-linear algebraic equations has to be solved using iterative techniques, e.g. Newton’s method [10], which enables determining the values of volumetric flow rates q_{vi} in the process of successive approximations with any predetermined precision ϵ .

3 Hydraulic installation cooperating with bottom ash coolers

3.1 Description of the installation and mathematical model

A cooling water installation cooperating with two bottom ash coolers of a circulating fluidized bed (CFB) boiler (called SAC-L and SAC-R, respectively for left and right bottom ash cooler) is discussed as the first example of a series-parallel hydraulic system. Figure 2 provides a schematic image of the most important elements of a SAC-R ash cooler – two screw coolers designated by the acronyms SC-L and SC-R on the left scheme. The right part of Figure 2 presents a schematic side view of the bottom ash cooler with an indication of the hot slag inlet (1), the outlet of the cooled slag (6), the inlet and outlet of the cooling water flowing through the shaft and blades of the SC-L screw (respectively (2) and (3)), the inlet and outlet of the cooling water which flows between the walls (casing) of the ash cooler (respectively (4) and (5)). The mass flow rate of the hot slag is directly delivered from the combustion chamber (1), after which it is split between two screw coolers (SC-L and SC-R), and then transported to point (6). During its motion along the

screw, the slag is gradually cooled by the cooling water circulating in the blades, the shaft and between the walls (casing).

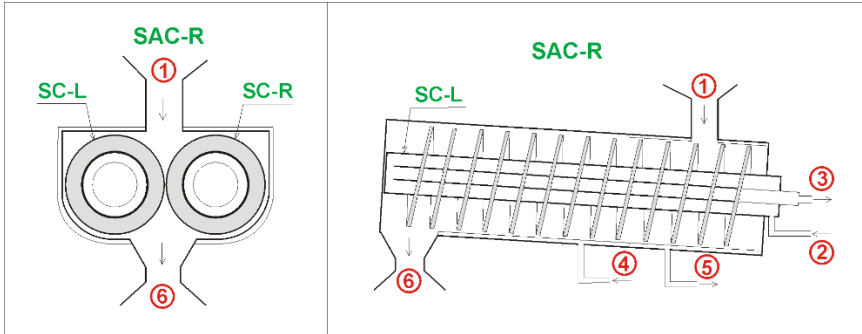


Fig. 2. Scheme of the bottom ash cooler (SAC-R) with internal cooling water distribution: two screw coolers (SC-L and SC-R), the inlet and outlet of the slag (1 and 6 respectively), the inlet and outlet of the cooling water (2,4 and 3, 5 respectively).

The series-parallel system of cooling water propagation to two bottom ash coolers of a CFB boiler is shown in Fig. 3.

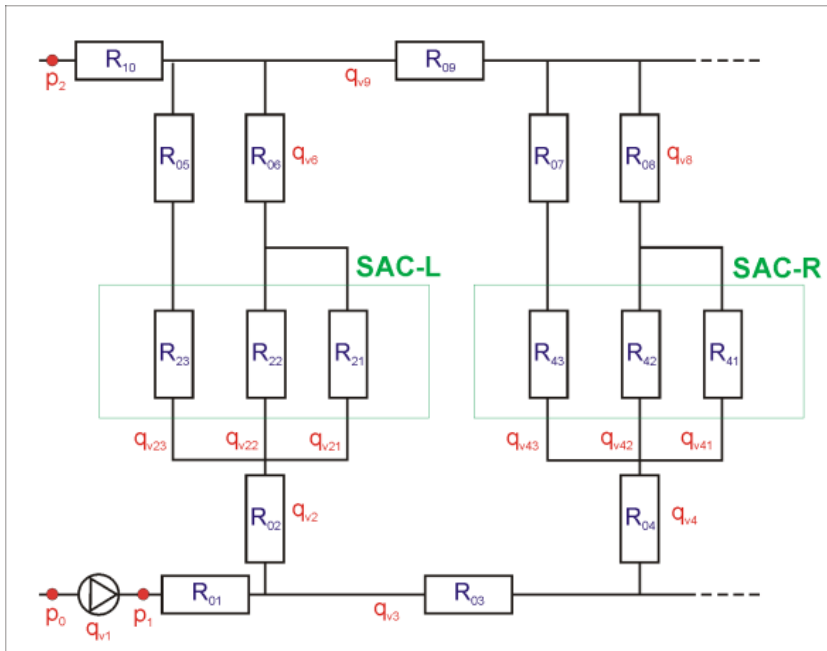


Fig. 3. An example of a series-parallel system of cooling water propagation to the two bottom ash coolers of a CFB boiler.

Each bottom ash cooler (SAC-L and SAC-R) consists of two screw coolers (SC-L and SC-R), supplied by cooling water volumetric flow rates denoted as q_{v21} and q_{v22} , q_{v41} and q_{v42} , respectively. The volumetric flow rates which cool down the walls of coolers are q_{v23} and q_{v43} , respectively. Cooling water is taken from the common collector and pumped into the system through a pump, which has a static pressure p_1 at the inlet to the installation. The static pressure at the exit from the system is denoted by p_2 . The pressure difference $\Delta p_{12} = (p_1 - p_2)$ forces the cooling water to circulate in all the branches of the system. The pump discharge is calculated based on the pressure difference $\Delta p_p = (p_1 - p_0)$. Cooling water

is delivered to (and removed from) each ash cooler by pipes with a diameter of $\phi 160$ (parts R_{01} , R_{03} , R_{09} and R_{10}), and next by pipes $\phi 120$ in diameter (parts R_{02} and $R_{04} - R_{08}$). The pipes which directly supply cooling water to the screws and walls of the ash coolers have a diameter of $\phi 60$ (parts $R_{21}-R_{23}$ and $R_{41} -R_{43}$).

Basing on the scheme presented in Fig. 3, it is possible to note eleven non-linear algebraic equations, including:

- 5 equations describing the water distribution in individual joints,
- 5 equations describing the pressure losses in the closed hydraulic meshes,
- 1 equation describing the pressure drop in the selected branch between the inlet and the outlet of the system (p_1 and p_2 , respectively).

The system of the abovementioned algebraic equations was solved iteratively by applying Newton’s method [10] with precision $\varepsilon = 10^{-3}$.

3.2 Numerical results and discussion

The aim of the numerical modelling was to indicate those elements of the analysed hydraulic installation whose improvement could significantly increase the overall volumetric flow rates of cooling water flowing through the bottom ash coolers (q_{v2} and q_{v4} , respectively). It is worth mentioning that for long-pipe installation, the major source of pressure drop are friction losses connected with the pipe parameters, such as their diameter, length and absolute roughness. Due to the fact that traces of pipelines are already fixed at the level of the boiler’s construction, the only parameters which could be changed in the existing installation, basing on the results of the mathematical model, are connected with the tube diameters. The starting point for numerical modelling was the existing hydraulic installation with the initial volumetric flow rates through the bottom ash coolers equalling $q_{v2} = 38.26 \text{ m}^3/\text{h}$ and $q_{v4} = 42.52 \text{ m}^3/\text{h}$. The pump discharge, forcing the flow in the cooling water system, amounted to $\Delta p_p = 0.325 \text{ MPa}$. The numerical simulation focused on the influence of the changes in the pipe diameters in the selected branches on the values of the volumetric flow rates q_{v2} and q_{v4} .

In the first variant, the diameters of the pipes in branches $R_{21}-R_{23}$ and $R_{41} -R_{43}$ were enlarged from the initial value of $\phi 60$ to $\phi 80$, $\phi 100$ and $\phi 120$, respectively. The results of the modelling are presented in Table 1.

Table 1. Results of the numerical modelling basing on the increase of the pipe diameters in branches $R_{21}-R_{23}$ and $R_{41} -R_{43}$.

pipe diameter ϕ , mm	volumetric flow rate q_{v2} , m^3/h	volumetric flow rate q_{v4} , m^3/h	pump discharge Δp_p , MPa
60	38.26	42.52	0.325
80	39.76	44.08	0.301
100	40.78	45.41	0.272
120	41.02	45.65	0.267

The increase of the diameters results in the increase of the volumetric flow rates q_{v2} and q_{v4} . It is worth noting that the reduction of the flow resistances in the system also results in the decrease of pump discharge Δp_p required to force the flow. An interesting observation is connected with the fact that the linear increase of the diameter does not reflect in the linear increase of q_{v2} and q_{v4} . Enlargement to $\phi 80$ results in the increase of the volumetric flow rate by $\Delta q_{v2} = 1.50 \text{ m}^3/\text{h}$ and $\Delta q_{v4} = 1.56 \text{ m}^3/\text{h}$ when further changes generate the increase at the level of:

- $\phi 80 \rightarrow \phi 100$: $\Delta q_{v2} = 1.02 \text{ m}^3/\text{h}$ and $\Delta q_{v4} = 1.33 \text{ m}^3/\text{h}$,
- $\phi 100 \rightarrow \phi 120$: $\Delta q_{v2} = 0.24 \text{ m}^3/\text{h}$ and $\Delta q_{v4} = 0.24 \text{ m}^3/\text{h}$.

The last change of diameter from $\phi 100$ to $\phi 120$ caused the increase of q_v by only $0.24 \text{ m}^3/\text{h}$ through both bottom ash coolers. This means that further enlargement of the diameter of these pipes does not have a significant influence on the volumetric flow rates q_{v2} and q_{v4} .

In the second variant, the diameters of the pipes in branches R_{01} , R_{03} , R_{09} and R_{10} were enlarged from the initial value $\phi 160$ to $\phi 200$ and $\phi 250$. The results of the modelling are presented in Table 2.

Table 2. Results of the numerical modelling basing on the increase of the pipe diameters in branches R_{01} , R_{03} , R_{09} and R_{10} .

pipe diameter ϕ , mm	volumetric flow rate q_{v2} , m^3/h	volumetric flow rate q_{v4} , m^3/h	pump discharge Δp_p , MPa
160	38.26	42.52	0.325
200	43.00	48.05	0.202
250	43.60	48.71	0.162

In this variant, the enlargement of the pipe diameters results in the increase of the volumetric flow rates:

- $\phi 160 \rightarrow \phi 200$: $\Delta q_{v2} = 4.74 \text{ m}^3/\text{h}$ and $\Delta q_{v4} = 5.53 \text{ m}^3/\text{h}$,
- $\phi 200 \rightarrow \phi 250$: $\Delta q_{v2} = 0.60 \text{ m}^3/\text{h}$ and $\Delta q_{v4} = 0.66 \text{ m}^3/\text{h}$.

This is a much larger increase than for the first variant, which means that the reduction of the flow resistances on these branches could increase the flow rates of cooling water through the bottom ash coolers more efficiently (by up to 14.5 %).

4 Hydraulic installation with condensers and cooling towers

4.1 Description of the installation and mathematical model

Another example of a series-parallel hydraulic system would be a closed cooling water system consisting of condensers and cooling towers. This kind of hydraulic installation usually appears in professional power plants located far from large natural sources of water (like seas, rivers or lakes). The daily water demand of power units often reaches a level of several thousand cubic meters, which signifies that knowledge about water distribution in hydraulic systems could help not only in better water management but also in improving heat exchange processes occurring in condensers and cooling towers.

The scheme of a closed cooling installation is presented in Fig. 4. Warm water from condensers is pumped to the cooling system by twelve pipes denoted as external volumetric flow rates ($q_{vz1} - q_{vz12}$). Through the pipelines, water is delivered to five cooling towers marked with black circles. Inside each cooling tower, water is additionally divided between a core and ring sections. Each branch of the system is described by the individual flow resistance R_i and volumetric flow rate q_{vi} (not marked in the scheme). The values of R_i were calculated basing on the technical documentation and literature data. Due to the fact that the pipelines are made from concrete, the value of the friction losses coefficient λ_i in (1) are calculated from the Altšul formula using the value of absolute roughness $k = 10^{-4} \text{ m}$ [8].

The aim of the mathematical model was to determine the values of internal volumetric flow rates q_{vi} . To do this, a set of 41 non-linear algebraic equations was created:

- 28 equations describe water distribution in individual joints,
- 4 equations describe pressure losses in the closed hydraulic meshes,
- 9 equations describe the pressure drop along traces connected to two outlets of the system.

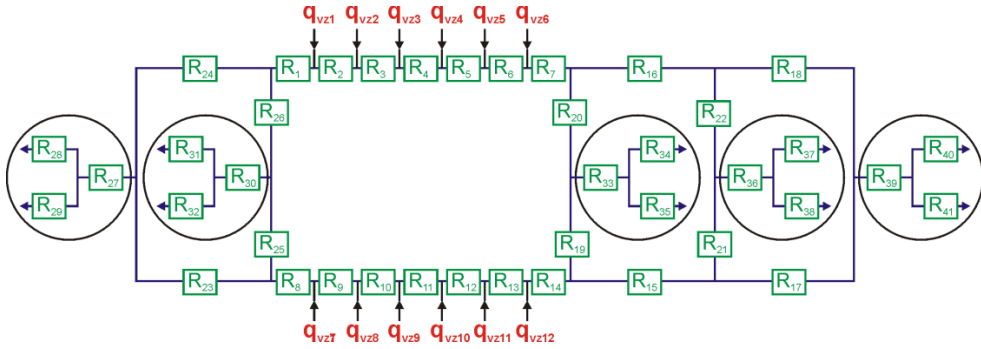


Fig. 4. Scheme of a closed cooling water system with external volumetric flow rates (from condensers) denoted by q_{vz} and flow resistances R_i (and connected internal volumetric flow rates q_{vi}). The five cooling towers are marked with black circles.

It is worth mentioning that for an open hydraulic installation it is not necessary to know the pressure at the inlet of the external volumetric flow rates q_{vz} because equations describing the pressure drop can be drawn between the outlets of the system (which have atmospheric pressure). A novelty with respect to the mathematical model described in subsection 3.1 is the need to determine the direction of the flow in all branches of the network. This is due to the fact that there are several inlets and outlets for cooling water. Therefore, in equations describing the pressure drop, the terms q_{vi}^2 were replaced by the expressions $(q_{vi} \cdot |q_{vi}|)$, which not only enable determining the value of q_{vi} but also the direction of the flow (through the positive or negative sign of q_{vi} values). The initial values of the volumetric flow rates and its flow directions are, in the further iterative process, adjusted to satisfy the equations of the mathematical model. The system of the abovementioned algebraic equations was solved by applying Newton’s method with precision $\varepsilon = 10^{-3}$. As a result of the calculations, volumetric flow rates to individual cooling towers were obtained: q_{v27} , q_{v30} , q_{v33} , q_{v36} and q_{v39} .

4.2 Numerical results and discussion

An example of measurements and numerical results from the mathematical model of the closed cooling water system is presented in Table 3.

Table 3. Comparison of the measurements and numerical results from the mathematical model of the closed cooling water system. The total external volumetric flow rate $q_{vz} = 151,100 \text{ m}^3/\text{h}$.

cooling tower	volumetric flow rate measurement, m^3/h	volumetric flow rate numerical result, m^3/h	relative error, %
1 (q_{v27})	27107	26986	0.45
2 (q_{v30})	31338	29087	7.74
3 (q_{v33})	25219	24599	2.52
4 (q_{v36})	35977	36612	-1.73
5 (q_{v39})	31459	33816	-6.97

The presented data indicates that the largest discrepancies between the measurements and numerical simulation are for the second and fifth cooling tower. The maximum value of relative error is at the level of 7.74 %. This is very high agreement taking into account the complexity of the series-parallel system reflected in the number of non-linear algebraic

equations. The numerical solution shows that the external volumetric flow rates q_{vz4} - q_{vz6} and q_{vz10} - q_{vz12} were directed only to cooling towers nos. 3-5 (q_{v33} , q_{v36} , q_{v39}) in contrast to the volumetric flow rates q_{vz1} , q_{vz2} and q_{vz7} , q_{vz8} , which only cooperate with cooling towers nos. 1 and 2 (q_{v27} and q_{v30}). The volumetric flow rates q_{vz3} and q_{vz9} are divided between the left and right group of cooling towers (1-2 and 3-5 respectively). Knowledge about flow resistances R_i in each branch enables numerically analysing the changes in the structure of the flows through the hydraulic system when e.g. one of the power units is turned off (which is simulated by the individual value of $q_{zi} = 0$).

5 Closing remarks

A mathematical model of a series-parallel hydraulic system enables the numerical study of the influence of individual components of the installation on the resulting volumetric flow rates. A set of non-linear equations is always individually established for the analysed hydraulic system. Such an approach makes it possible to study the impact of its various modifications on the generated volumetric flow rates. This information provides the investor with an opportunity to decide which of the analysed options gives the best economic effects as the cost of the improvements is often compared to the expected return in increased efficiency as a primary determinant of whether to go forward with a modernization.

References

1. R.J. Campbell et al., *Prospects for coal in electric power and industry*, Tech. rep., Congressional Research Service Report, R42950 (2013)
2. C. Henderson, P. Baruya, *CO₂ reductions from CCTs and CO₂ capture*, CCC/200, London, UK, IEA Clean Coal Centre (July 2012)
3. A. Ryabchikov et al., *Power Technology and Engineering*, **44 3** (2012)
4. M.A. Malek, *Heating boiler operator's manual: maintenance, operation and repair*, McGraw-Hill companies, New York (2007)
5. K.E. Heselton, *Boiler operator's handbook*, Lilburn, GA: Fairmont Press, New York, (2005)
6. C. Nichols et al., *Reducing CO₂ emissions by improving the efficiency of the existing coal-fired power plant fleet*, US Department of Energy National Energy Technology Laboratory, DOE/NETL-2008/1329 (2008)
7. G. Boncimino et al., *Costs and effectiveness of upgrading and refurbishing older coal-fired power plants in developing APEC economies*, Asia-Pacific Economic Cooperation, Energy Working Group Project EWG 04/2003T, (June 2005), available: http://www.egcfe.ewg.apec.org/projects/UpgradePP_Report_2005.pdf
8. F.M. White, *Fluid mechanics*, McGraw-Hill companies, New York (2010)
9. Y. Nakayama, R.F. Boucher, *Introduction to fluid mechanics*, Butterworth-Heinemann (2000)
10. J.H. Mathews, K.D. Fink, *Numerical methods using Matlab*, Prentice Hall (1999)
11. R.B. Baker, H.A. Shryock, *A comprehensive approach to the analysis of cooling tower performance*, SPX Marley, Thermal Science (2013)