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Cooling water flow influence on power plant unit performance for various condenser configurations setup

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Abstract This paper presents the influence of cooling water regulation on power plant net efficiency. It was examined whether, for the non-nominal low-pressure turbine load, it is justified to reduce the cooling water pump load, and how it would affect the unit net efficiency. Calculations for two types of power units were carried out: with condensing and extraction-condensing turbine. The tested condensing power plant consists of three surface condensers. The calculation included four condensers' connections set up on the cooling water side to check how the cooling water system pressure drop affects the net unit performance. The result has confirmed that implementing serial connection decreases net efficiency when cooling water flow regulation is used, but the mixed connection should be applied when pump load is not controlled. It was proved that the cooling water flow control gives a profit for both units. Net efficiency for combined heat and power plant can be improved by 0.1–0.5 pp, the gain is remarkable below 60% of the low-pressure turbine part load. Flow control implementation in the unit with condensing turbine water control gives a similar profit just below 80% of the turbine load. Next, an influence of the additional limitations of a cooling water system (minimal total pump head, cooling tower) affecting the feasibility of implementing the water control has been considered. Applying a multi-cell forced draft cooling tower does not have a significant impact on results, but when a natural draft cooling tower is used, the flow control range is strongly reduced.

Keywords: Cooling water; Power plant efficiency; Cooling water flow control; CHP plant efficiency

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Nomenclature

c	–	velocity, m/s
d_i	–	pipe inside diameter, m
Δp	–	pressure difference, m
Δh_p	–	friction losses in pipes, m
Δh_e	–	friction losses in fittings and equipment, m
g	–	gravitational acceleration, m/s^2
H_d	–	dynamic head, m
H_g	–	geometric pumping head, m
H_s	–	static head, m
H_t	–	total head, m
i	–	enthalpy, kJ/kg
k	–	roughness factor, m
l	–	length, m
$\dot{m}$	–	mass flow rate, kg/s
n	–	pump motor rotation, rpm
N_{CWP}	–	total power of cooling water pump, kW
N_{e_CWP}	–	effective net power of the cooling water pump, kW
N_{el}	–	electrical power generated by turbo generator, kW
N_{DWH}	–	heat generated in the district water heater, kW
p	–	pressure, MPa
PMP_{no}	–	number of running cooling water pumps
Re	–	Reynolds number
$\dot{Q}_B$	–	heat transferred from boiler, kW
$\dot{V}$	–	volumetric flow rate, m^3 /h
T	–	temperature, °C
T_{CWin}	–	inlet cooling water temperature, °C
$T_{\mathrm{DWH_{in}}}$	–	inlet district water temperature, °C
$T_{\mathrm{DWH_{out}}}$	–	outlet district water temperature, °C

Greek symbols

α	–	pump blade angle, °
η	–	efficiency
$\eta_{1-\mathrm{CWP}}$	–	unit efficiency considering cooling water pump power
λ	–	friction factor
ξ	–	friction factor in fittings and equipment
ρ	–	density, kg/m^3

Subscripts

g	–	cooling water to the condenser
gt	–	total cooling water (including water to coolers)
m_CPW	–	motor of cooling water pump
nom	–	calculation for nominal flow
opt	–	calculation for optimal flow
$Par.Conf$	–	parallel configuration

Par/Ser.Conf	–	parallel-to-serial configuration
Ser.Conf	–	serial configuration
Ser/Par.Conf	–	serial-to-parallel configuration

Abbreviations

B	–	boiler
C	–	condenser
CHP	–	combined heat and power
CP	–	condensing pump
CW	–	cooling water
CWP	–	cooling water pump
DWH	–	district water heater
FWP	–	feed water pump
FWT	–	feed water tank
HPH	–	high pressure heater
LP	–	low pressure
LPH	–	low pressure heater

1 Introduction

The opportunities to improve the efficiency of a power plant are sought in the project phase, but also in the whole operation cycle. At the design stage, the greatest emphasis is placed on the net system efficiency for nominal parameters [1]. This is a significant contractual parameter and its increase is one of the main goals in a project phase. However, companies producing electricity and heat, increasingly try to reduce additionally the operating costs for the full operating range. Efficiency improvement potential can be seen for lower turbine loads, for which the system has often not been optimized. Performing a comprehensive analysis and calculation of the system can help in introducing additional control systems, which will increase the net unit efficiency [2].

This paper presents the calculations aimed at optimizing the net power plant efficiency by using the cooling water flow regulation. There has been a question posed whether, for low unit loads, the reduction of the cooling water flow, and thus the cooling water pump power, increase the net power plant efficiency.

Power plants and combined heat and power plants (CHP) built in the 1980s and 1990s were very rarely equipped with cooling water pumps, where the load regulation would be possible. During the modernization of the 200-MW units, some motors have been replaced for ones with inverters. The papers [3,4] confirmed that the implementation of cooling water pump load control improved the net power plant efficiency in both tested examples. In [5] author described research results concerning the cooling water flow control under the variable load of 200 MW steam turbine based on the minimum entropy and maximum power generation. In [6] the same author expands the research by including a new unit model and pump characteristics. This analysis confirms that below the 60% turbine load, the cooling water flow should be reduced. In the next papers [7–9], the authors depicted the case of implementing cooling water control with seawater. The papers indicate the profits that can be obtained and the justification for the implementation of such installation. But what is important, all aforementioned works refer to condensing turbines and none of the cooling water flows limitations were considered.

In the following analysis, the first combined heat and power (CHP) plants are studied. Extraction-condensing turbines are characterized by low pressure (LP) turbine part load for high values of extraction steam flow, which gave grounds warranted a consideration that it has a potential for optimization [10]. Primarily, the heat balance model was prepared. Based on real data, the needed parameters have been selected to assure proper heat balance accuracy covering the full work range in both, condensing and CHP modes. As a next step, the cooling water system and pump characteristics were implemented. Additionally, the impact of another cooling water system equipment was considered – the limitations come from cooling tower or coolers, which could completely change result interpretation.

Afterward, the described calculation was performed for condensing turbine with three low-pressure turbine parts. In the beginning, the hydraulic resistance of various connection setups was verified. In the preceding part of the research [11], it was checked whether the changes of condenser connection on the cooling water side have a significant impact on the achieved unit parameters. It was calculated that the highest gross unit efficiency was obtained for the serial configuration, successively for mixed configuration, and the least advantageous for thermodynamic reasons was the parallel connection. At the beginning of research on the reasonableness of the implementation of cooling water control, the previously tested connections were compared, considering the influence of the cooling water system on the net power plant efficiency. Including hydraulic resistance of the tested condenser connections (parallel, series, mixed), the calculation has indicated whether the previously obtained optimal configuration changed. In the following stage, a similar step, as for the CHP plant, has been undertaken to get the characteristics of the optimal cooling water flow for all setups. To compute an optimized cooling water flow, the power plant heat balance, cooling water system resistance, and the cooling water pump power for the whole operation range, for both units have been calculated.

2 Tested thermal cycle scheme

The tests were performed for the two thermal cycles. In Fig. 1 the combined heat and power plant scheme (CHP plant A) was shown. Its nominal parameters are:

$N_{el} = 50$ MW, $N_{\mathrm{DWH}} = 86$ MW ($N_{el} = 50$ MW in condensing mode),
$p_{10} = 12$ MPa, $T_{10} = 545°$C, $\dot{m}_{10} = 60.8$ kg/s, $\dot{m}_g = 1854$ kg/s.

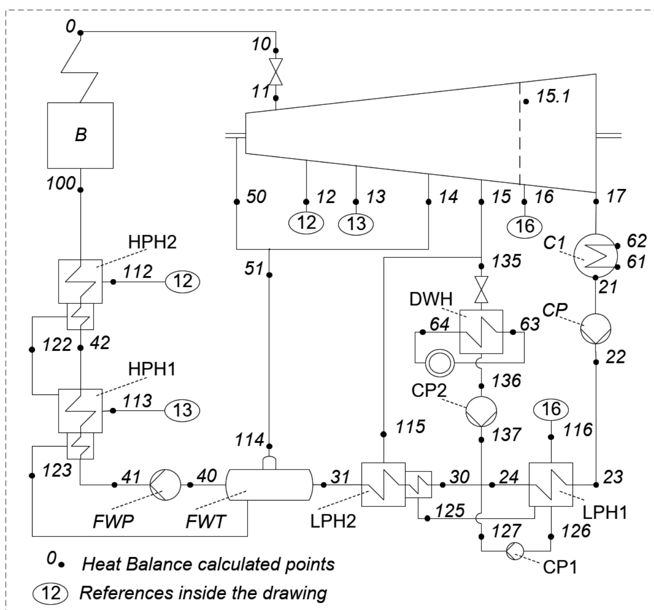


Figure 1: Tested thermal cycle scheme – CHP plant (A).

Figure 2 presents a cooling water system for this CHP plant. The tested cooling water system consists of a cooling water tank and 2 × 50% cooling water pumps with inverters. Apart from the main condenser, cooling water pumps also supply coolers (the consumption of coolers is about 9% of the total amount of cooling water for nominal parameters). The heat of the cooling water is transferred in a 4-cell forced draft cooling tower. It is feasible to work with one cooling water pump and a half condenser surface area. An important technological limitation of the cooling water system is keeping the minimum total pump head due to the coolers.

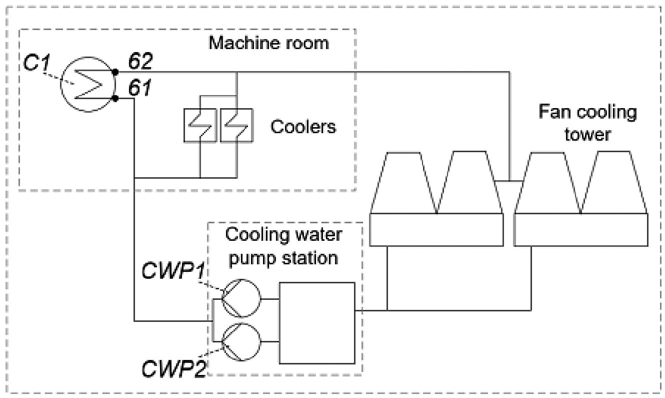


Figure 2: Cooling water system – CHP plant (A).

In Fig. 3 a second tested power plant with a condensing turbine (power plant B) was shown. The condenser configuration setups are described in Fig. 4. The nominal power plant parameters are:
N_{el}=910 MW, p_{10}=27.5 MPa, T_{10}=600°C, $\dot{m}_{10}$=658 kg/s, $\dot{m}_g$=22500 kg/s.

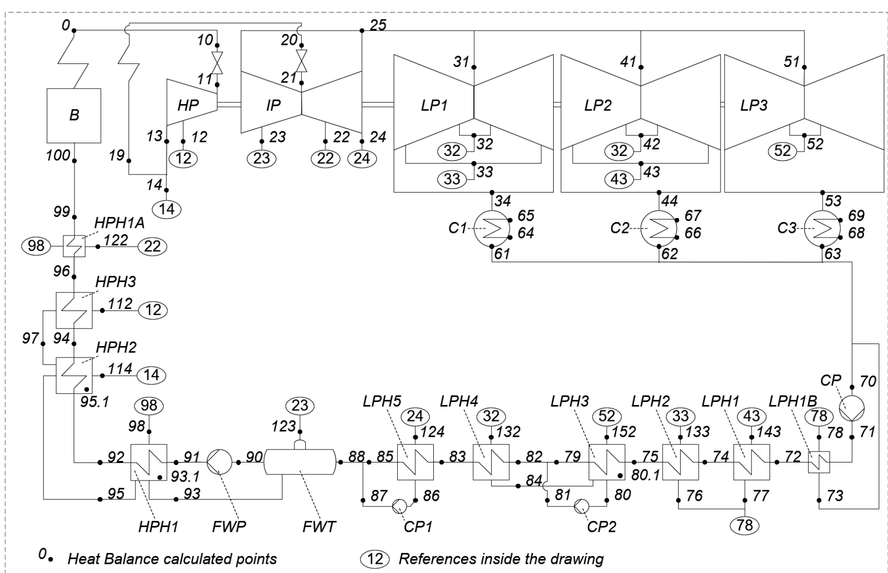
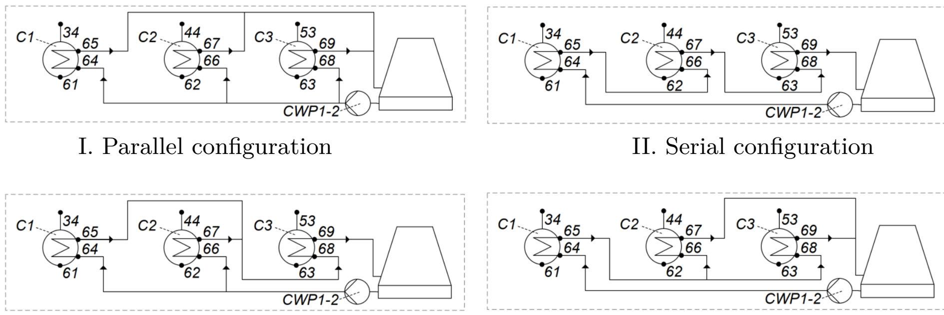


Figure 3: Tested thermal cycle scheme – power plant (B).



I. Parallel configuration

II. Serial configuration

III. Parallel-to-serial configuration

IV. Serial-to-parallel configuration

Figure 4: Tested condenser connections – power plant (B).

Figure 5 presents the corresponding cooling water system which consists of a cooling water tank and 2 × 50% cooling water pumps with adjustable blade angles. Apart from the main condenser, cooling water pumps also supply coolers (the consumption of coolers is about 4% of the total amount of cooling water for nominal parameters). The heat of the cooling water is transferred in the cooling tower. It is possible to work with one cooling water pump and half of the condensers' surface areas.

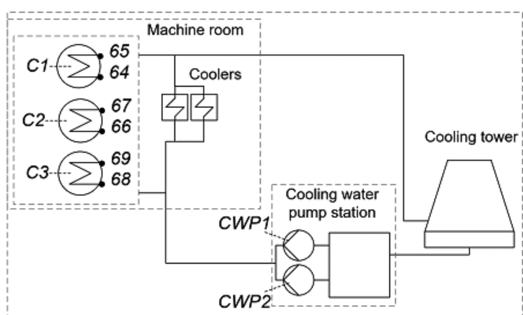


Figure 5: Cooling water system – power plant (B).

3 Calculation procedure

3.1 Heat balance calculation

The heat balance calculation for power plant (B) has been thoroughly depicted in [11]. A similar calculation has been prepared for CHP plant (A). The thermal cycle was described by the energy and mass balance equations.

The coefficients of the system of equations were appointed by the enthalpy value at the determined points. Enthalpy was calculated from the thermodynamic dependence in accordance with IAPWS IF-97 [12]. Exhaust steam pressure was calculated based on an algorithm with HEI (Heat Exchange Institute) heat transfer coefficient [13]. Using iterative calculations, the pressure, temperature, enthalpy, and mass flow rate were computed. All the data used for model validation comes from a real unit.

3.2 Cooling water pump characteristics

Cooling water pump characteristics are presented in Figs. 6–9. They are based on the data coming from the technical documentation of the pump manufacturer. Description P1 refers to the characteristics of the operation of one pump, P2 when two pumps are running. The area limited by $\dot{V}_{\min}$, $\dot{V}_{\max}$ lines is the available pump's operating range.

The nominal parameters for CHP plant (A) are:

$H_t = 24$ m, $\dot{V} = 3800$ m^3/h, $n = 744$ rpm, $\eta_{\text{CPW}} = 84\%$;
control type = inverter.

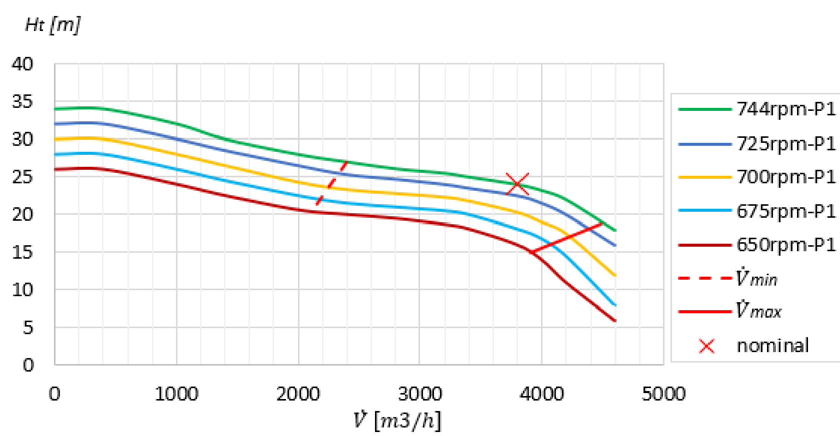


Figure 6: Total pump head – CHP plant (A).

The nominal parameters for power plant (B) are:

$H_t = 27.6$ m, $\dot{V} = 42500$ m^3/h, $n = 744$ rpm, $\eta_{\text{CPW}} = 88.5\%$; $\alpha = 4.5°$;
control type = adjustable blade angle.

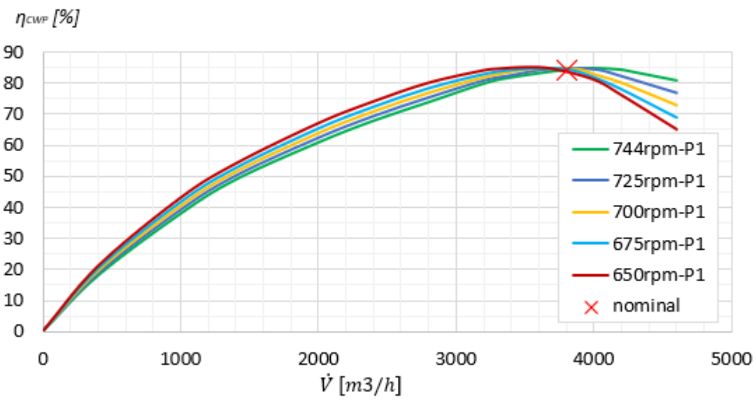


Figure 7: Pump efficiency – CHP plant (A).

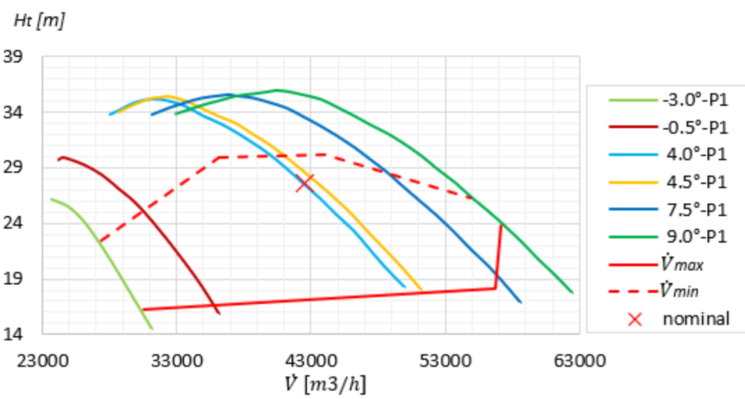


Figure 8: Total pump head – power plant (B).

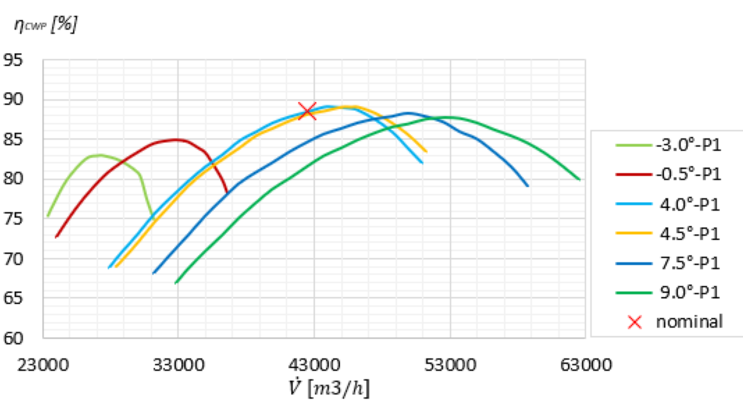


Figure 9: Pump efficiency – power plant (B).

3.3 Cooling water system characteristics

To read from the pump characteristics the demand pump power, the total pump head H_t must be calculated [14,15]. It is defined as

$$H_t = H_s + H_d \, . \tag{1}$$

The static head of the pump is calculated as

$$H_s = H_g + \frac{\Delta p}{\rho_g g} \, , \tag{2}$$

where H_g is the geometric pumping head, Δp is the pressure difference between the outlet and inlet of the installation, ρ_g is the cooling water density, and g represents gravitational acceleration. Since the inlet and outlet of the installation are at atmospheric pressure, the static head of the pump is calculated as a height difference between the surface of the liquid in the cooling water tank and the end of installation in a cooling water tower.

The dynamic head of the pump is expressed as

$$H_d = \frac{c_{g2}^2 - c_{g1}^2}{2g} + \Delta h_p + \Delta h_e \, , \tag{3}$$

where c_{g1} and c_{g2} are the water velocity at the installation inlet and outlet, respectively, Δh_p denotes the friction losses in pipes, whereas Δh_e stands for the friction losses in fittings and equipment. As the inlet water velocity c_{g1} equals zero, it is omitted.

After simplifications, the total head takes the form

$$H_t = H_g + \frac{c_{g2}^2}{2g} \Delta h_p + \Delta h_e \, . \tag{4}$$

Friction losses in pipes are calculated as

$$\Delta h_p = \lambda \frac{l}{d_i} \frac{c_g^2}{2g} \, , \tag{5}$$

where λ is the friction factor, l is the pipe length, d_i represents the pipe inner diameter. In accordance with the recommendations of the standard [15], the friction factor is determined based on the Colebrook-White formula

$$\lambda = \left[-2 \log \left(\frac{2.51}{\mathrm{Re}\sqrt{\lambda}} + \frac{k}{3.72 d_i} \right) \right]^{-2} , \tag{6}$$

where k is the roughness factor and Re is the Reynolds number.

Friction losses in fittings and equipment are calculated as

$$\Delta h_e = \xi \frac{c^2}{2g}\,, \tag{7}$$

where ξ is the friction factor in fittings and equipment.

Roughness factor k depends on the material and type of the pipe, surface condition, and operating conditions. Friction factor in fittings and equipment ξ depends on the element type and geometric dimensions. The proper values of both factors are read from [15].

Knowing the lengths and diameters of the pipes, fittings, and equipment used on cooling water installation, the system characteristics were calculated (Figs. 10, 11). Characteristic, prepared when whole condenser surface area and two pumps operate, was described by *CW.sys (whole cond.)* function. When one pump and a half condenser surface area was used, function was named *CW.sys (half cond.)*. In Fig. 10 cooling water system characteristics for CHP plant (A) is presented. For validation two reference points were marked: *p1-705* rpm which indicates a real working point when the whole condenser is used, and *p2-712* rpm represents the case when half of the condenser and one pump are running. Figure 11 demonstrates the cooling water system characteristics of power plant (B) when the mixed connection is implemented. A comparison of cooling water characteristics for various condenser configurations (mixed, serial, and parallel) is shown in Fig. 12.

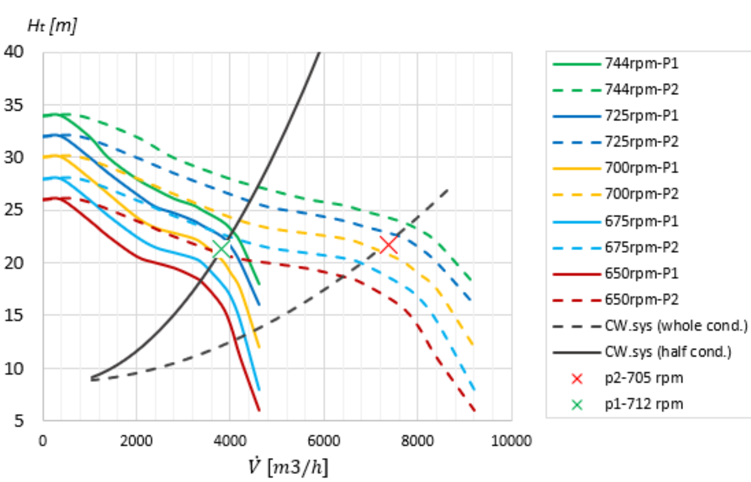


Figure 10: Cooling water system characteristics – CHP plant (A).

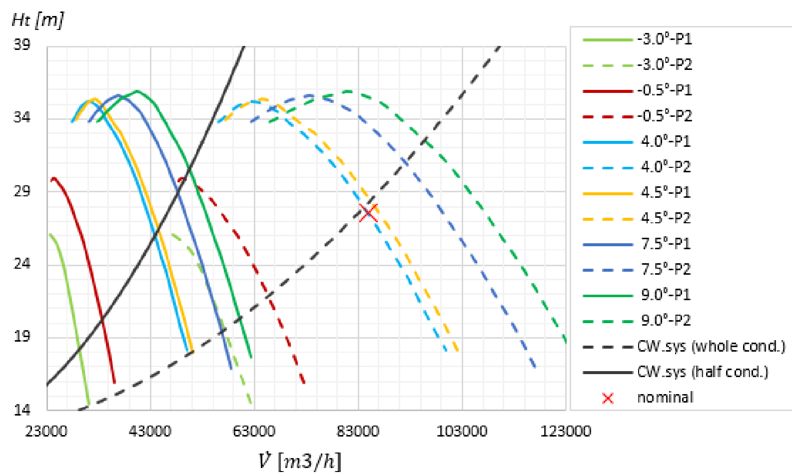


Figure 11: Cooling water system characteristics – power plant (B).

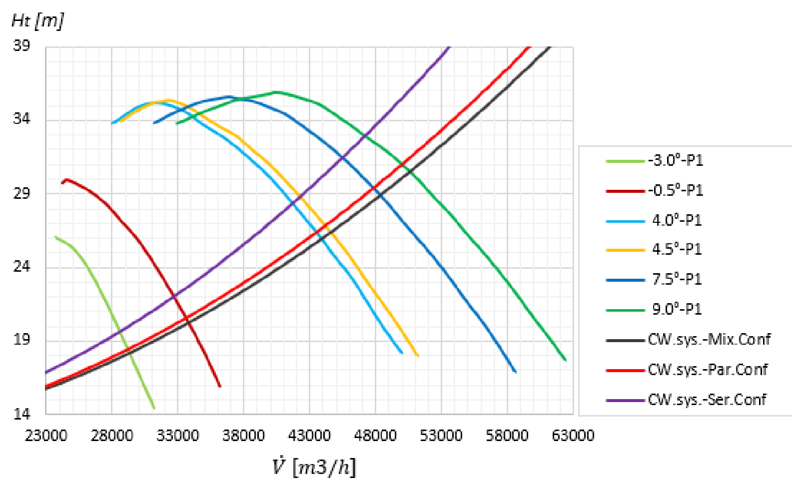


Figure 12: Cooling water system characteristics, comparison of condenser connection setups – power plant (B).

CW.sys (whole cond.) corresponds to the operating mode with two pumps and whole condenser surface area. *CW.sys (half cond.)* presents characteristic when one pump and half condenser surface was used.

Legend description refers to the cooling water system characteristics when mixed configuration (*CW.sys – Mix.Conf*, parallel configuration (*CW.sys – Par.Conf*, serial configuration (*CW.sys – Ser.Conf*) are implemented.

3.4 Cooling water pump power

The effective net power of the cooling water pump is calculated from

$$N_{e\_\mathrm{CWP}} = H_t \dot{m}_{gt} g, \tag{8}$$

where $\dot{m}_{gt}$ is the total cooling water flow rate.

The total power of the cooling water pump is then expressed as

$$N_{\mathrm{CWP}} = \frac{N_{e\_\mathrm{CWP}}}{\eta_{\mathrm{CWP}} \eta_{m\_\mathrm{CWP}}}, \tag{9}$$

where η_{CWP} is the cooling pump efficiency and η_{m_CWP} is the motor efficiency.

3.5 Comparative index

The following indicator was calculated for the comparison and evaluation of results

$$\eta_{1-\mathrm{CWP}} = \frac{N_{el} - N_{\mathrm{CWP}}}{\dot{Q}_B}, \tag{10}$$

where

$$\dot{Q}_B = \dot{m}_0 (i_0 - i_{100}) \tag{11}$$

for CHP plant (A), and

$$\dot{Q}_B = \dot{m}_0 (i_0 - i_{100}) + \dot{m}_{20} (i_{20} - i_{19}) \tag{12}$$

for power plant (B).

3.6 Description of the calculation procedure

To calculate the optimal cooling water flow and the possible net efficiency increase, the next steps were executed:

I. New work point was selected (N_{el}, N^*_{DWH}, p_0, T_0, $T_1 T^*_{\mathrm{DWH_{in}}} T^*_{\mathrm{DWH_{out}}}$) from the assumed operating range (* – only for CHP plant (A)).

II. New value of cooling water flow rate $\dot{m}_{gt}$ from the assumed operating range was appointed.

III. New heat balance was computed.

IV. The cooling water system parameters were determined:

- Required total head for actual water flow (Figs. 10 and 11).
- Based on the total head value and cooling water flow, reading the values of rotation speed/blade angle (Figs. 6 and 8), and next the pump efficiency (Figs. 7 and 9).
- Pump power calculation using Eq. (9).

V. The quality indicators were calculated (based on Eq. (10)).

VI. If cooling water reduction is possible return to step II. If the minimum flow is achieved – return to step I.

Apart from the basic calculations, computing was done considering the various constraints of the cooling system. It was verified, how the results would change when including the cooling water limitations:

- Usually, the main cooling water pumps also supply the water to coolers. In the tested CHP plant, it forced the minimal total head (21.5 m) and this limited the cooling water real flow regulation range.
- Natural and forced draft cooling towers have also cooling water flow limitations. In both cases, they are designed for nominal volumetric flow (±10%). When a forced draft cooling tower with more cells is used, limitations concern each cell by itself, so there is also a possibility to control the number of working cells (CHP plant A). But for a natural draft cooling tower (power plant B) it would be more complicated.

4 Calculation results

4.1 Combined heat and power plant A

The ranges of the variability of the tested operating points for calculation were assumed as follows:

$N_{el} = 20\,000 - 60\,000$ MW (step 5000 kW)
$T_{CW} = 14 - 25°$C (step 3°C),
$\dot{m}_{gt} = 800 - 2300$ kg/s (step 15 kg/s),
$N_{\mathrm{DWH}} = 7\,000 - 100\,000$ kW (step 5000 kW),
$T_{\mathrm{DWH_{in}}} = 45 - 55°$C (step 3°C),
$T_{\mathrm{DWH_{out}}} = T_{\mathrm{DWH_{in}}} + 30°$.

When minimal pump head was expected its value was $H_{t\_\min} = 21.5$ m (limitation 1).

When water flow limitation due to forced draft cooling tower was considered (4 cells designed, each for nominal flow of $\dot{V} = 1900$ m^3/h ($\pm 10\%$)), the flow had to belong to one of the ranges:

$\dot{V} \in (1710 - 2090) \cup (3420 - 4180) \cup (5130 - 6270) \cup (6840 - 8360)$ m^3/h (limitation 2).

Calculation results were provided on the charts presenting the cooling water mass flow, the total pump head, the operating pump number (Figs. 13, 17, and 21), and the power plant efficiency increase (defined by Eq. (13), Figs. 14, 18, and 22) as a function of LP turbine part load

$$d\eta_{1-\mathrm{CWP}} = \eta_{1-\mathrm{CWP(opt)}} - \eta_{1-\mathrm{CWP(nom)}}, \tag{13}$$

where subscripts *nom* and *opt* refer to calculations for nominal flow and optimal flow, respectively.

Due to the often-repeated question, on how to interpret that an increase in efficiency translates into real profit, the additional graphs showing this profit were prepared. Namely:

- Figures 15, 19, and 23 present a possible additional production of electrical power, assuming the same amount of generated heat that is needed for nominal water flow, but considering the optimal net efficiency

$$dN_{el} = \dot{Q}_{B(\mathrm{nom})}\eta_{1-\mathrm{CWP(opt)}} - (N_{\mathrm{el(nom)}} - N_{\mathrm{CWP(nom)}}). \tag{14}$$

Substituting Eqs. (10) and (13), the formula can be simplified to

$$dN_{el} = \dot{Q}_{B(\mathrm{nom})} d\eta_{1-\mathrm{CWP}}. \tag{15}$$

- Figures 16, 20, and 24 present the possible daily profit (for 24 h constant operating point), assuming the electricity energy production price of 65 EUR/MWh [16,17]. This calculation shows only the order of magnitude of savings and the electricity production cost was not analyzed.

Below three groups of characteristics are allocated. Figures 13–16 show results for a basic calculation, Figs. 17–20 illustrate the case when the minimal head is expected (limitation 1), and Figs. 21–24 present results when water flow is limited due to forced draft cooling tower conditions (limitation 2).

Figures 13–16 show optimization results for basic calculation. Characteristics confirm that the implementation of cooling water regulation increases the net CHP plant efficiency for not nominal load. Obtained results indicate that especially below 50% of LP turbine part load changing the speed of the pumps gives meaningful benefits – an increase in efficiency by 0.1–0.5 pp. When the total head is not limited, in the whole turbine load range both pumps are running.

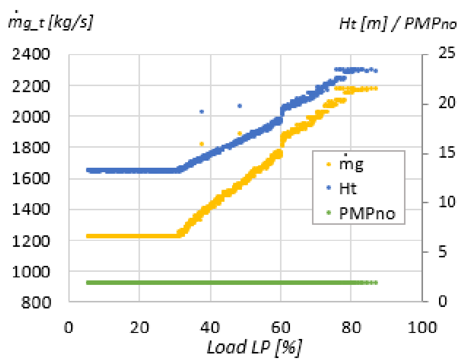


Figure 13: Cooling water system parameters – basic calculation.

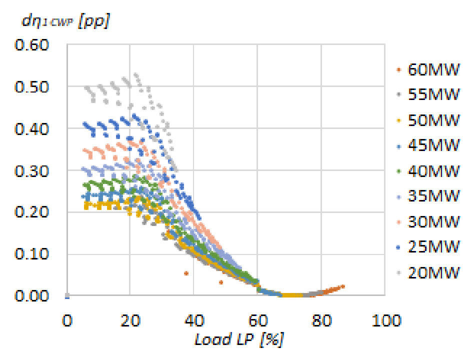


Figure 14: Net efficiency increase – basic calculation.

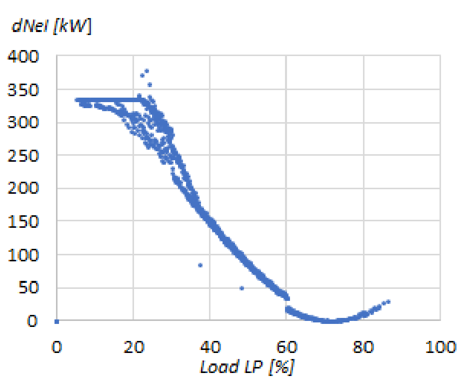


Figure 15: Optimization profit – basic calculation.

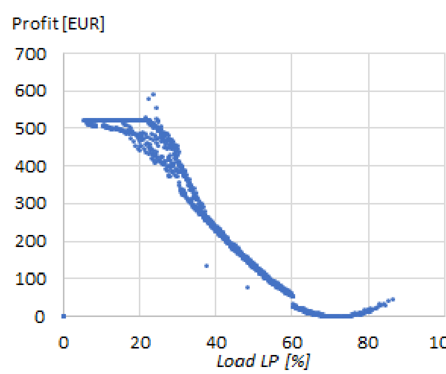


Figure 16: Optimization daily profit – basic calculation.

Figures 17–20 present optimization results when the minimal head is expected (limitation 1). In this case, the cooling water flow regulation is limited to the selection of the number of working pumps, because the pumps' setpoint must be always set near the nominal point. Considering this limitation, the net efficiency is improved below 34% of LP turbine part load and in this range, it reaches lower values of 0.1–0.4 pp.

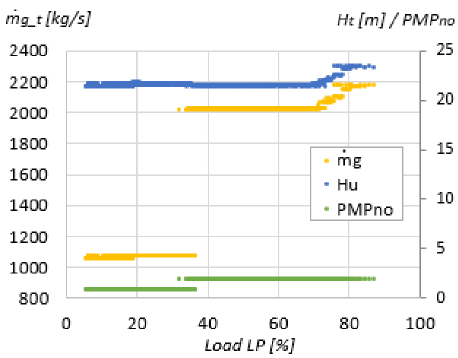


Figure 17: Cooling water system parameters – minimal pump head limitation actives.

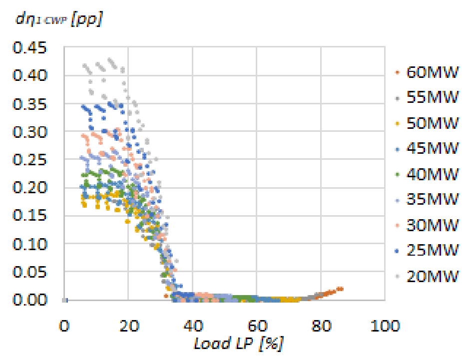


Figure 18: Net efficiency increase – minimal pump head limitation actives.

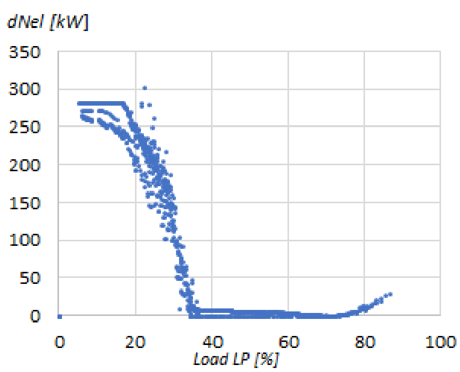


Figure 19: Optimization profit – minimal pump head limitation actives.

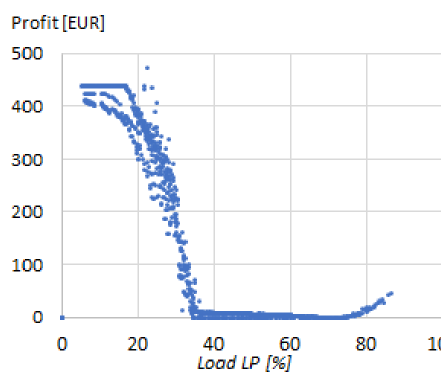


Figure 20: Optimization daily profit – minimal pump head limitation actives.

Figures 21–24 present optimization results for the case when water flow is limited due to forced draft cooling tower conditions (limitation 2). In the described example, there are four independent cells, which can be turned on and off by an operator because of the expected cooling water flow. Based on the results, it can be stated that using a forced draft cooling tower with more cells does not limit the ability to regulate the cooling water, which would significantly reduce the net efficiency improvement, given by the implementation of cooling water flow regulation.

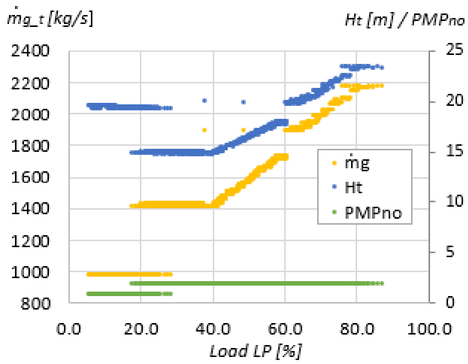


Figure 21: Cooling water system parameters – cooling tower flow limitation actives.

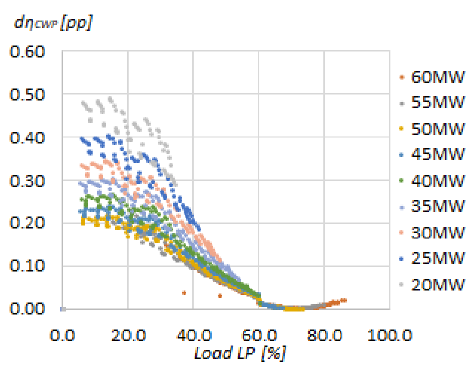


Figure 22: Net efficiency increase – cooling tower flow limitation actives.

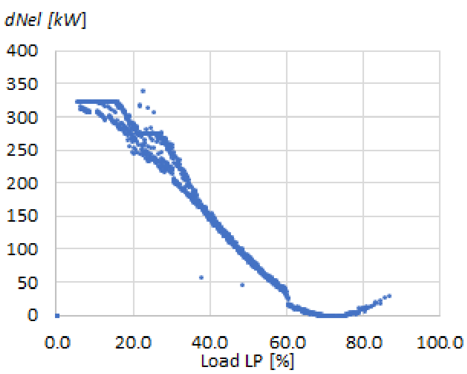


Figure 23: Optimization profit – cooling tower flow limitation actives.

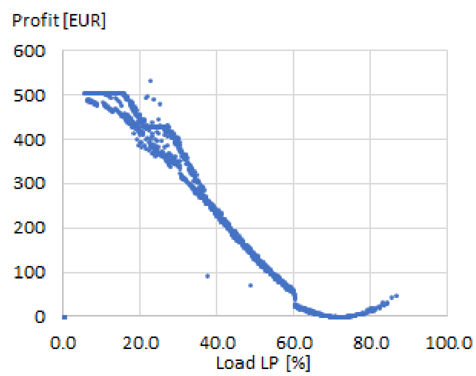


Figure 24: Optimization daily profit – cooling tower flow limitation actives.

4.2 Power plant B

The ranges of the variability of the tested operating points for calculation were assumed as follows:

$N_{el} = 100\ 000 - 900\ 000$ MW (step 25000 kW),

$T_{\mathrm{CW_{in}}} = 16 - 24°\mathrm{C}$ (step 8 °C),

$\dot{m}_{gt} = 8500 - 23500$ kg/s (step 500 kg/s).

When the water flow rate is limited due to natural draft cooling tower the volumetric flow must belong to range $\dot{V} \in (76\ 500 - 93\ 500)$ m^3/h (limitation 1).

Four groups of characteristics are demonstrated further in the paper. First, the influence of surface condenser connection on the net efficiency is shown, and next, the opportunities to increase the net efficiency for all tested condenser setups.

Figures 25–28 present calculation results for a nominal cooling water flow rate for three condenser setups, serial, parallel-to-serial, and serial-to-parallel, compared with the result obtained for parallel connection. The charts show the cooling water flow rate (Fig. 25), the net efficiency increase (Fig. 26), which is defined by Eqs. (16)–(21), the possible additional production of electrical power (Fig. 27), assuming the same amount of gen-

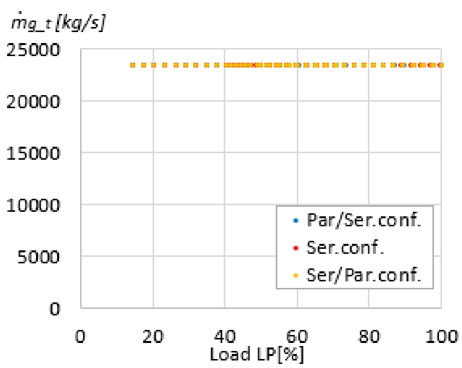


Figure 25: Cooling water flow rate – comparison of condenser connection for nominal flow.

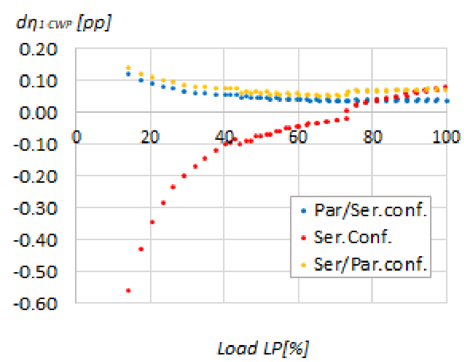


Figure 26: Net efficiency increase – comparison of condenser connection for nominal flow.

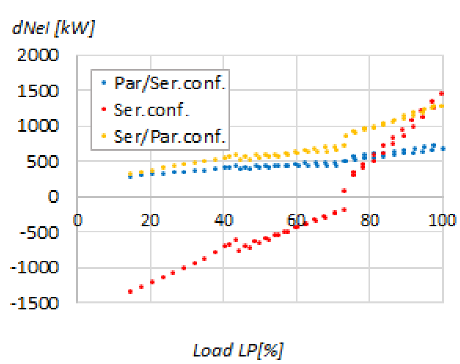


Figure 27: Optimization profit – comparison of condenser connection for nominal flow.

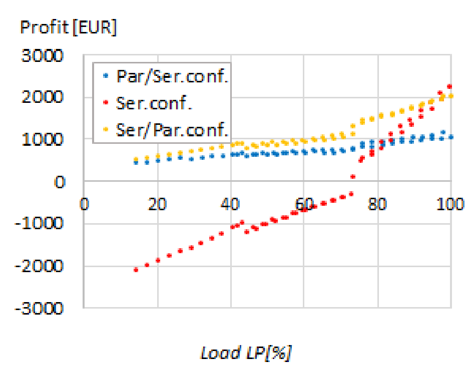


Figure 28: Optimization daily profit – comparison of condenser connection for nominal flow.

erated heat that is needed for parallel configuration but considering the net efficiency being one of the three more complex configurations (according to Eqs. (19)–(21)), and the possible daily profit (Fig. 28):

$$d\eta_{1-\mathrm{CWP}(Ser.Conf)} = \eta_{1-\mathrm{CWP}(Ser.Conf)} - \eta_{1-\mathrm{CWP}(Par.Conf)}, \tag{16}$$

$$d\eta_{1-\mathrm{CWP}(Ser/Par.Conf)} = \eta_{1-\mathrm{CWP}(Ser/Par.Conf)} - \eta_{1-\mathrm{CWP}(Par.Conf)}, \tag{17}$$

$$d\eta_{1-\mathrm{CWP}(Par/Ser.Conf)} = \eta_{1-\mathrm{CWP}(Par/Ser.Conf)} - \eta_{1-\mathrm{CWP}(Par.Conf)}, \tag{18}$$

$$dN_{el(Ser.Conf)} = \dot{Q}_{Par.Conf}\, d\eta_{1-\mathrm{CWP}(Ser.Conf)}, \tag{19}$$

$$dN_{el(Ser.Conf)} = \dot{Q}_{Par.Conf}\, d\eta_{1-\mathrm{CWP}(Ser/Par.Conf)}, \tag{20}$$

$$dN_{el(Par/Ser.Conf)} = \dot{Q}_{Par.Conf}\, d\eta_{1-\mathrm{CWP}(Par/Ser.Conf)}. \tag{21}$$

Here, index *Par.Conf* refers to a calculation for parallel configuration. Correspondingly, indices *Ser.Conf*, *Ser/Par.Conf*, and *Par/Ser.Conf* apply to parameter values determined for serial configuration, serial-to-parallel configuration, and parallel-to-serial configuration, respectively.

Figures 25–30 present an influence of a surface condenser setup on a net power plant efficiency. New characteristics change the evaluation of the previous calculations concerning the influence of condenser connection setup on the gross efficiency. Having the condenser resistance and its impact on cooling water pump power, the serial connection gives the best results only for a nominal load. Below the 72% turbine load, it becomes the worst even from the standard – parallel connection. When cooling water flow is fixed, the highest net efficiency gives mixed connections.

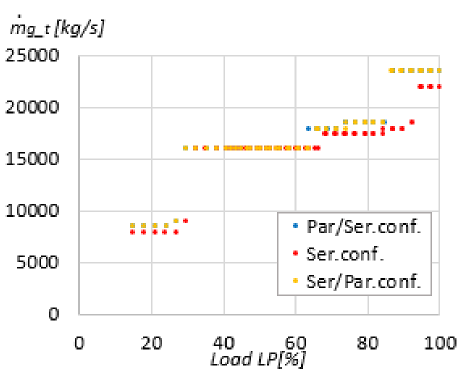


Figure 29: Cooling water flow rate – comparison of condenser connection for optimal flow.

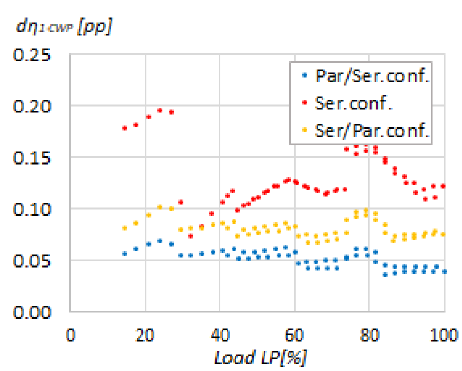


Figure 30: Net efficiency increase – comparison of condenser connection for optimal flow.

Its increase in comparison with a parallel connection is 0.05–0.07 pp. Although the technical minimum of the tested unit is 40% for the theoretical considerations, calculations were performed for the LP turbine part loads down to 20%.

A similar set of comparisons, but for the data obtained for optimal cooling water flow is shown in Figs. 29–32. These figures the comparison of a condenser setup influence on the net power plant efficiency for three connection configurations, when cooling water flows control is implemented. In this case, the highest increase in the net efficiency (0.1–0.2 pp) gives the serial configuration. Also, a mixed configuration gives better results than a parallel connection and an opportunity for an additional profit of 0.05–0.1 pp in the whole operating range.

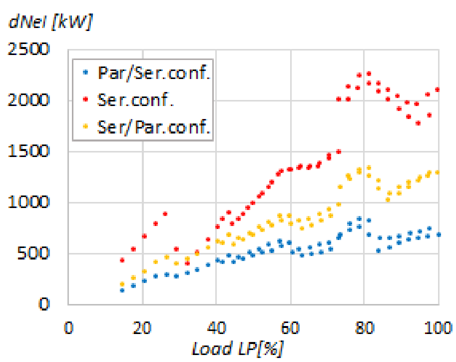


Figure 31: Optimization profit – comparison of condenser connection for optimal flow.

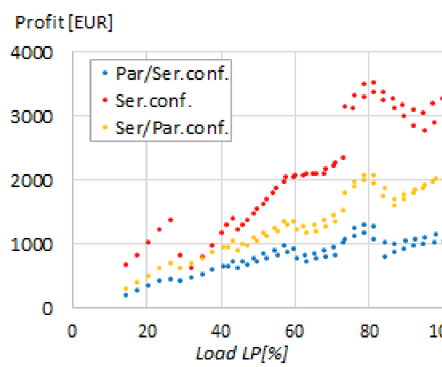


Figure 32: Optimization daily profit – comparison of condenser connection for optimal flow.

In Figs. 33–36 the opportunities to increase the net efficiency for all tested condenser setups are presented. These figures summarize the optimization results for basic calculation, i.e. without any limitation in the flow range control, for four connection types. Figure 33 shows the cooling water mass flow rate, whereas Fig. 34 illustrates the power plant efficiency increase (determined based on Eq. (13)), for all tested configurations as a function of LP turbine part load. The following Fig. 35 presents the possible additional production of electrical power, assuming the same amount of generated heat that is needed for a nominal water flow rate but considering an optimal net efficiency (Eq. (15)), and in Fig. 36 the possible daily profit is shown.

The characteristics obtained for this (basic) case confirm that the implementation of cooling water regulation increases the net power plant efficiency. The biggest net efficiency increase is remarked for a serial configuration, but it is a result of a big resistance for nominal cooling water flow rate. All other configurations have similar optimization possibilities. Below 80% power plant load, the influence of a cooling water pump load reduction is visible and the net efficiency gain is 0.1 pp. In the technical minimum (40%) it decreases by 0.5 pp. Although the increase may seem negligibly small, translating this value to a real profit gives values worth attention.

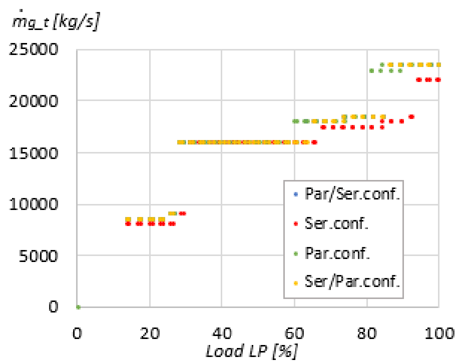


Figure 33: Cooling water flow rate – basic calculation for four tested configurations.

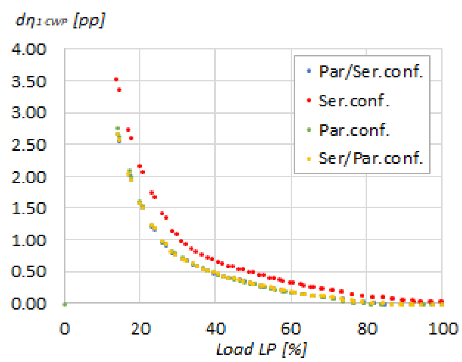


Figure 34: Net efficiency increase – basic calculation for four tested configurations.

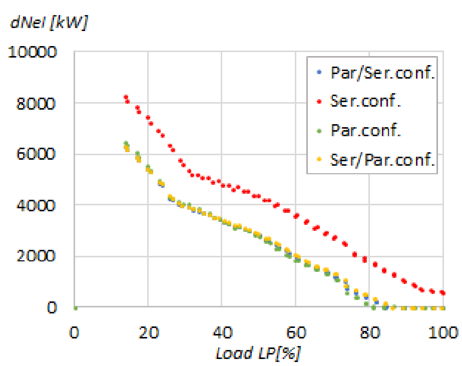


Figure 35: Optimization profit – basic calculation for four tested configurations.

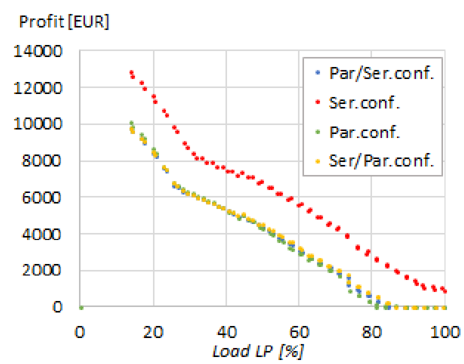


Figure 36: Optimization daily profit – basic calculation for four tested configurations.

Correspondingly, Figs. 37–40 present a set of comparisons for the studied case when the water flow rate is limited owing to the natural draft cooling tower condition. The obtained optimization results confirm that the profits from implementing cooling water flow control exist, but they are much smaller than in the case when the whole operating range is considered. Profit is visible below 72% turbine load and for the technical minimum, it reaches 0.18 pp for serial connection and 0.12 pp for other ones. It is justified to verify if the implementation of a cooling water distribution system in a draft cooling tower is possible and could expand the cooling water control range.

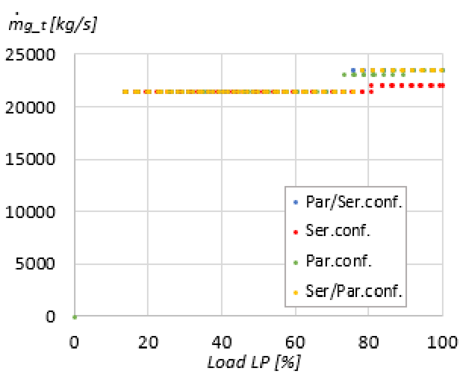


Figure 37: Cooling water flow rate – cooling tower flow limitation actives for four tested configurations.

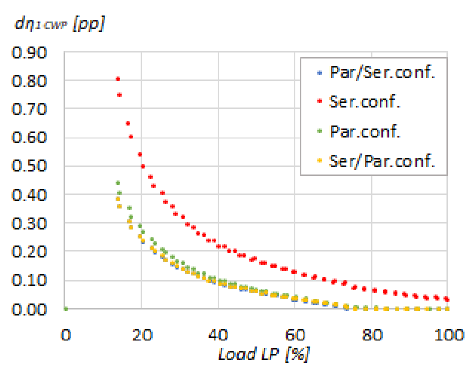


Figure 38: Net efficiency increase – cooling tower flow limitation actives for four tested configurations.

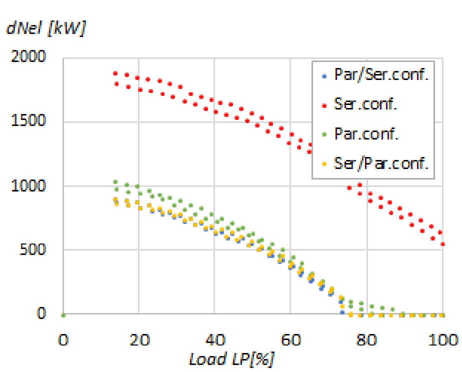


Figure 39: Optimization profit – cooling tower flow limitation actives for four tested configurations.

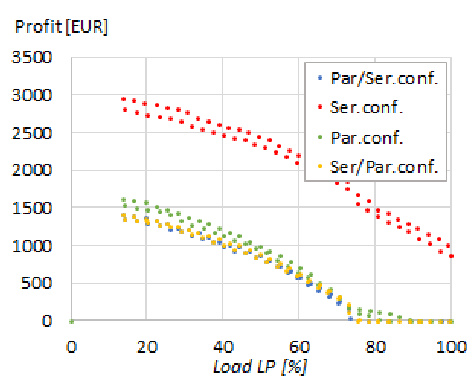


Figure 40: Optimization daily profit – cooling tower flow limitation actives for four tested configurations.

5 Result discussion

The purpose of the presented calculations is to check whether it is possible to improve the net efficiency of the power plant, by interfering in the cooling water system. First, the CHP plant was tested. The obtained results confirm that the control of a cooling water flow can be profitable.

The biggest efficiency increase is observed for low electrical load and high load of the heat exchanger, where it achieves 0.5 pp. Although the gain is remarkable just below 60% LP turbine part load, under this value the cooling water control gives a profit. The determined net efficiency increase recalculated for a daily profit gives 65–500 EUR/day. It should however be explained that taking into account real profits depends on the way it is defined. On the one hand, it can be defined as savings, which means, that the production cost of 1 MWh must be considered. On the other hand, these savings can mean more electrical power sold to the grid, so in this case, the market price should be included. Defining this kind of calculation is very difficult, especially today, when the market price of energy changes and has a significant, even daily, fluctuation. In the calculation carried out, the market price of 65 EUR/MWh was assumed. However, looking at the actual energy prices, which achieved the monthly average levels of 100 EUR/MWh and 120 EUR/MWh over the past two months, respectively [17, 18], it means that calculated profits can be greater.

In the performed calculations, the cooling water system limitations were included. When the implementation of cooling water flow control is considered, all cooling water system components must be verified. First, an external cooler condition shall be checked. Very often units are supplied by a main cooling water pump and the total heads of these units cause limitations in cooling water pump control. An example of such is the tested CHP plant. In this condition, when LP turbine often operates in a part load range of 30–50%, it is worth calculating the probability of a new small pump installation, which guarantees a minimal head for coolers and makes the cooling water pump independent [4]. When the turbine load is often outside this range, the control can be reduced to the selection of a number of working pumps, and the profits would be achieved below 30% LP turbine part load.

When a closed cooling circuit is used, the specification of the cooling tower must be verified. It is then very important to check what flow range is accepted and if there is a possibility to implement an additional water distribution automation system to avoid unfavorable or dangerous cooling tower work conditions. In presented results for CHP plant, considering permissible flow did not change possible profit significantly. The operating range of four independent cell cooling water systems is wide enough to get a visible profit.

In the first step of testing the power plant with a condensing turbine, the influence of condenser connection setups on the cooling water side on the net unit efficiency has been checked. In the previous paper [11], the condenser connection setups effect on gross efficiency was studied. Considering only the thermodynamic parameters, the serial condenser connection gives the best results. In the present study, the impact of cooling water system resistance was analyzed. Based on obtained results, a serial condenser connection should not be implemented when a cooling water pump has no automatic control system. Tube's resistance in this connection generates loss which surpasses the thermodynamic profit. In this case, a mixed connection should be applied. The results derived from the cooling water flow control approach are different and show that serial connection gives greater net unit efficiency. The mixed connection is also better than the parallel one.

The provided here calculations are important at the design stage when the connection type of a condenser is chosen. Although, when the existing power plant is considered, it is necessary to verify, if the optimization is possible. According to this study outcomes, all condenser connections have a potential for optimization. The net efficiency increase is feasible below 80% turbine load. The operating range for calculation was consciously extended. For condensing turbine, the LP turbine part load approximately equals turbine load. Increasingly, the transformation of the power plant to the CHP plant is done. In such a case, the LP turbine part load can be smaller than a technological minimum, and simultaneously, a control system for the cooling water flow should be introduced. However, a big technological problem for the large units would be the limitations posed by the natural draft cooling towers. The possibility of introducing automatic regulation on the cooling tower must be checked by a technologist to avoid a situation when the cooling tower has not enough load. If applicable, an application of an automatic cooling water distribution system in a natural draft cooling tower could change possibilities of control in a similar way that for a fan draft cooling tower with more cells.

6 Conclusion

The calculation result has confirmed that the cooling water systems of both tested thermal power plants have the optimalization potential.

- Implementation of cooling water flow control in the combined heat and power plant gives a profit below 60% LP turbine part load. In line with the expectations, the biggest profit can be achieved for the minimal exhaust steam flow and reaches 0.5 pp.
- Verifying power plant cooling system, the conclusions were drawn, that the biggest efficiency is reached for the serial to parallel condenser configuration set up, when the water flow regulation is not implemented.
- When the water flow control is active, better results gives a serial connection, where the water flow should be decreased just below the nominal turbine load.
- For all tested configuration the water flow should be reduced below 80% turbine load to improve unit net efficiency.
- Although the values of the efficiency increase could be considered slight, it translates the calculation results into an additional generated power or a financial profit, hence makes these values noteworthy and gives real gain.
- When analyzing the possibility of optimizing of the cooling water system, the external restrictions should be always taken into consideration.

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