Influence of three surface condensers connection setup on power plant unit performance

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Abstract. This paper presents a comparison of three surface condenser connection setups on the cooling water side. Serial, mixed and parallel connections were considered. The thermodynamic justification for the use of more complex configurations was verified. The analysis was conducted based on the calculated heat balances of verified power units for nominal and not nominal parameters for tested connections. The exhaust steam pressure was calculated using the technical data of the surface condenser and cooling water parameters. Three methods of calculating the heat transfer coefficient based on characteristic numbers, HEI method, and the ASME standard, were used. The most advantageous model was indicated and used in heat balance calculations. The assumptions and simplifications for the calculations are discussed. Examples of the calculation results are presented.

The direction of development and transformation in the Polish power industry are a response to the new European legal regulations, the aim of which is the environmental protection. As a result, diversification of electricity production was provoked, additionally coal-fired units were forced to apply new solutions. New units are getting greater, technically and technologically more advanced. Introduced improvement that increases unit efficiency by as much as 0.1 percent are important considering the actual requirements and unit efficiency of 45% net.

In the case of great unites, when few surface condensers are used, it is essential to verify, which connection setups on the cooling water side would give the highest unit efficiency.

When research, it is necessary to consider the entire steam-condensate cycle to analyze exhaust steam pressure changes effect on steam flow and thermodynamic parameters fluctuations. The correct calculation of the exhausted steam pressure value is one of key steps to ensure the calculation’s correctness. This paper presents analysis of three exhausted steam pressure calculation algorithms, comparison of their complexity and the set of the data needed for these calculations. The model of the surface condenser was based on three methods of calculating the heat transfer coefficient: dimensionless equation with characteristic numbers, HEI method and the ASME standard. The most advantageous model was indicated after verification with the data from the site.

Study of three surface condenser connection setups on the cooling water side aims to help to find the answer on the rationale behind using more complex configurations, to analyze their advantages and disadvantages, and to give advice on which system is the best from the thermodynamics perspective.

1 Exhaust steam pressure calculation

Condensation turbine exhaust steam pressure was calculated using the heat transfer equations.

Model assumed isobaric heat exchange, no condensate subcooling. Calculations were made for steady state. Thermodynamic calculations in accordance with IAPWS IF-97 [1].

1.1 Heat transfer equations:

Heat transfer equations are:

\[
\dot{Q} = m_s(i_s - i_c)
\]  
\[\dot{Q} = m_g C_g (T_2 - T_1) \eta_c \]  
where \(\dot{Q}\) is the heat transfer rate, \(m_s\) is the exhausted steam flow rate, \(i_s\) is the exhausted steam enthalpy, \(i_c\) is the condensate enthalpy, \(m_g\) is the cooling water flow rate, \(C_g\) is the water specific heat, \(T_1\) is the inlet cooling water temperature, \(T_2\) is the outlet cooling water temperature, \(\eta_c\) is the condenser efficiency.

1.2 Condenser heat load: [2],[4],[5].

Condenser heat load equation is:

\[
\dot{Q} = U A_s LMTD
\]  
\[LMTD = \frac{T_s - T_1}{\ln\frac{T_s - T_1}{T_2 - T_1}}\]

where \(\dot{Q}\) is the condenser heat load, \(U\) is the heat transfer coefficient, \(A_s\) is the surface tube area, \(LMTD\) is the logarithmic mean temperature difference, \(T_s\) is the condensate temperature, \(T_1\) is the steam temperature, \(T_2\) is the outlet cooling water temperature.
1.3 Heat transfer coefficient - Characteristic numbers [2].

Heat transfer coefficient, when characteristic number method used, equation is:

\[
U = \frac{1}{\frac{R_m + R_t + R_f}{D_i}} 10^{-3}
\]  
(5)

Tube-Wall resistance was computed as follows:

\[
R_m = D_{\text{out}} \ln \frac{D_{\text{out}}}{D_i} \frac{1}{2K_m}
\]  
(5.1)

Shellside resistance was computed based on dimensionless equation for heat transfer when the steam condenses on the outside horizontal pipe:

\[
R_s = \left( \frac{N_u K_s}{D_i} \right)^{-1}
\]  
(5.2)

\[
N_u = 0.725 C_v^{0.25}
\]  
(5.3)

\[
C_v = \frac{D_{\text{out}}^2 \delta_i^3 g d_{\text{cap}}^2}{K_{\text{Nuss}} T_{\text{cond}}}
\]  
(5.4)

\[
dT_{ct} = \frac{\Delta T}{\delta_c} \frac{1}{\delta_c}
\]  
(5.5)

- The difference in condensate and wall temperatures depends on the thickness of the condensate layer, therefore on the heat transfer coefficient. It is indicating that the most appropriate calculation method is the iterative method, but with satisfactory accuracy, the value can be calculated as \( dT_{ct} = \frac{LMTD}{2} \) [3].

Physical properties of condensate: \( \delta_c, K_c, \mu_c \) are determined for surface and saturation average temperature

\[
T_f = \frac{T_s + T_c + dT_{ct}}{2}
\]  
Tubewall resistance was computed based on dimensionless equation for forced convection for turbulent flow inside a circular pipe:

\[
R_t = \left( \frac{N_u K_t}{D_i} \right)^{-1}
\]  
(5.5)

\[
N_u = 0.021 Re^{0.8} Pr_g^{0.43}\left( \frac{P_r_2}{P_r_1} \right)^{0.25}
\]  
(5.6)

\[
Re = \frac{\omega g \delta_i}{\mu_g}
\]  
(5.7)

\[
V_g = \frac{\omega g \delta_i}{\mu_g N \pi D_i^2}
\]  
(5.8)

\[
Pr = \frac{\mu_C P}{K}
\]  
(5.9)

\[
PR_g / PR_t - \text{calculated for water temperature } T_g \text{ and wall temperature } T_t = T_s - dT_{ct}
\]

Nomenclature: \( U \)- heat transfer coefficient, \( D_i \)- tube inside diameter, \( D_{\text{out}} \)- tube outside diameter, \( K_m \)- tube thermal conductivity, \( K_c \)- shellside thermal conductivity, \( R_m \)- tubewall resistance, \( R_t \)- tube side resistance, \( R_f \)- fouling resistance, \( N_u \)- Nuselt number, \( Pr \)- Prandtl number, \( Re \)- Reynolds number, \( T_g \)- average cooling water temperature, \( V_g \)- cooling water velocity, \( C_P g \)- water specific heat, \( d_{\text{cap}} \)- enthalpy of exhausted steam vaporization, \( dT_{ct} \)- The difference in condensate and wall temperatures, \( \mu_c \)- condensate viscosity, \( \mu_g \)- cooling water viscosity, \( \delta_c \)- condensate density, \( \delta_g \)- cooling water density.

1.4 Heat transfer coefficient - HEI standard [4]

In this case, the calculation of heat transfer coefficient is based on design guidelines of Heat Exchange Institute (HEI). The proposed function uses the data from experimental research. The heat transfer coefficient was computed as follows

\[
U = U_1 F_w F_m F_c
\]  
(6)

Nomenclature: \( U_1 \)- uncorrected heat transfer coefficients, as a function of tube diameter and cooling water velocity, \( F_w \)- inlet water temperature correction factor, \( F_m \)- tube material and gauge correction factors, \( F_c \)- cleanliness factor.

\( U_1, F_w, F_m \) are read from HEI table. \( U_1 \) values are based on clean, 1.245 mm tube wall gauge, Admiralty metal tubes with 21.1°C cooling water temperature. Uncorrected heat transfer coefficient is describing as a function of tube diameter and water velocity. \( F_w \) introduces a water temperature correction and \( F_m \) introduces a tube material and gauge correction.

1.5 Heat transfer coefficient - ASME PTC 12.2 codes: [5]

Heat transfer coefficient, when ASME codes used, equation is:

\[
U = \frac{1}{\frac{R_m + R_t + R_f}{D_i}} 10^{-3}
\]  
(7)

Tube-Wall resistance was computed as follows:

\[
R_m = D_{\text{out}} \ln \frac{D_{\text{out}}}{D_i} \frac{1}{2K_m}
\]  
(7.1)

Shellside resistance was computed as follows:

\[
R_s = \left( \frac{N_u K_c}{D_i} \right)^{-1}
\]  
(7.2)

\[
N_u = 0.01158 Re^{0.835} Pr_{g}^{0.426}
\]  
(7.3)

\[
Re = \frac{V_g \omega g \delta_i}{\mu_g}
\]  
(7.4)

\[
V_g = \frac{\omega g \delta_i}{\mu_g N \pi D_i^2}
\]  
(7.5)

\[
P_r = \frac{\mu_C P}{K}
\]  
(7.6)

Shellside resistance for the first iteration was computed as follows:

\[
R_s = \frac{1}{T_{i+1}} - R_m - R_t \frac{D_{\text{out}}}{D_i} - R_f
\]  
(7.7)

Shellside resistance for the next iteration was computed as follows:

\[
R_s = R_{s0} \left( \frac{0.1}{D_i} \right)^{1.7} \frac{1}{P_r} \frac{1}{K_c} \frac{1}{K_m} \left( \frac{\delta_{g_0}}{\delta_{g}} \right)^{2} F_c
\]  
(7.8)

Nomenclature determined as in point 1.3.

Index 0 means the value from the previous iteration. Physical properties of condensate: \( \delta_c, K, \mu \) are determined for condensate film \( T_f = T_s - 0.2LMTD \).

Condenser technical data: Steel 1.4401 - \( K_m = 15 \frac{W}{mK} \) was assumed to be used, tube dimension 024x0.7mm, condenser efficiency \( \eta_c = 0.99 \), cleanliness factor \( F_c = 0.95 \); Calculation was done for two pass surface condenser with surface tube area \( A_s = 19177 m^2 \), quantity of tubes \( N = 31920 \).

1.6 Turbine’s isentropic efficiency

In calculation turbine’s isentropic efficiency was used

\[
\eta_1 = \frac{P_{1\text{ex}}-P_{0\text{ex}}}{L_{\text{inlet}}-L_{\text{ex}}}
\]  
(8)

Nomenclature: \( L_{\text{inlet}} \)- inlet LP turbine steam enthalpy, \( L_{\text{ex}} \)- exhaust steam enthalpy, \( L_{0\text{ex}} \)- exhaust steam enthalpy when isentropic flow, \( P_{1\text{ex}} \)- exhaust steam pressure,
η₁ was set as a constant because in external test, small impact of vary this value as a function of load, for main calculation, was verified.

1.7 Calculation procedure and example calculation results

Variables calculation was based on the iterative algorithm. Calculation procedure is the same for dimensionless equation with characteristic number (CN) and HEI methods but different for ASME standard. Input data: \( A_e \) – surface tube area, \( N \) – quantity of tubes, \( F_c \) – cleanliness factor, \( D_i \) – tube inside diameter, \( D_{out} \) – tube outside diameter, \( K_m \) – tubewall thermal conductivity, \( m_g \) – cooling water flow rate, \( p_g \) – cooling water pressure, \( T_1 \) – inlet cooling water temperature, \( p_{s,LP} \) – inlet LP turbine steam pressure, \( T_{s,LP} \) – inlet LP turbine steam temperature, \( m_{\text{exhausted}} \) – exhausted steam flow rate, \( dT_c \) – condensate subcooling, \( \eta_1 \) – turbine efficiency, \( \eta_c \) – condenser efficiency, condenser pass number

Table 1. Input data for calculations

| series | 1     | 2     | 3     | 4     |
|--------|-------|-------|-------|-------|
| load   | 100%  | 90%   | 75%   | 60%   |
| \( \dot{m}_g \) | kg/h  | 48060 | 48060 | 48060 | 48060 |
| \( T_1 \) | °C    | 18.3  | 16.8  | 15.2  | 16.7  |
| \( m_s \) | kg/h  | 753240| 697910| 587350| 491770|
| \( T_{s,LP} \) | °C    | 278.7 | 273.6 | 280   | 272.6 |
| \( p_{s,LP} \) | kPa   | 579   | 526   | 441   | 358   |
| \( \eta_1 \) |       | 0.82  | 0.82  | 0.82  | 0.82  |

Nomenclature: \( \dot{m}_g \) – cooling water flow rate; \( T_1 \) – inlet cooling water temperature; \( m_s \) – exhausted steam flow rate; \( T_{s,LP} \) – inlet LP turbine steam temperature; \( p_{s,LP} \) – inlet LP turbine steam pressure, \( \eta_1 \) – LP turbine’s isentropic efficiency

Table 2. Example calculation results

| series | 1     | 2     | 3     | 4     |
|--------|-------|-------|-------|-------|
| load   | 100%  | 90%   | 75%   | 60%   |
| \( p_{s,REF} \) | kPa   | 4.63  | 4.09  | 3.47  | 3.56  |
| \( p_s \) | kPa   | 4.31  | 3.77  | 3.12  | 3.10  |
| \( \dot{Q} \) | kW    | 466641| 432688| 368136| 310295|
| \( U \) | kW/m² | 3.45  | 3.41  | 3.38  | 3.45  |
| \( T_2 \) | °C    | 26.6  | 24.5  | 21.7  | 22.2  |
| HEI    |       |       |       |       |
| \( p_s \) | kPa   | 4.88  | 4.28  | 3.51  | 3.41  |
| \( \dot{Q} \) | kW    | 467298| 433316| 368667| 310664|
| \( U \) | kW/m² | 2.59  | 2.52  | 2.45  | 2.52  |
| \( T_2 \) | °C    | 26.6  | 24.5  | 21.7  | 22.2  |
| ASME   |       |       |       |       |
| \( p_s \) | kPa   | 3.84  | 3.38  | 2.83  | 2.87  |
| \( \dot{Q} \) | kW    | 466023| 432138| 367714| 310000|
| \( U \) | kW/m² | 5.19  | 5.11  | 5.02  | 5.06  |
| \( T_2 \) | °C    | 26.6  | 24.5  | 21.7  | 22.2  |

Nomenclature: \( p_{s,REF} \) – reference exhausted steam, \( p_s \) – calculated exhausted steam, \( \dot{Q} \) – condenser heat load, \( U \) – heat transfer coefficient, \( T_2 \) – outlet cooling water temperature

In table 1 input data were presented. In table 2 example calculation results compare with real exhausted steam pressure were shown.

1.8 Discussion of the results and exhaust steam pressure calculation method selection.

Verifying calculations have been made for data from real units: 65M W and 460M W. This paper presents results for

Fig. 1. Exhausted steam pressure
was used to assess the series of results. For calculation based on HEI method it is 0.176kPa, for characteristic numbers method 0.361kPa, for ASME method 0.704kPa. For second reference unite the best results gave characteristic numbers and HEI method. Least accurate results give ASME method. Although this method largely based on similar equations as characteristic numbers method, significant results difference follows from shellside resistance calculation. Reviewing the actual reference data, it is concluded that the results with the best accuracy were obtained using the HEI method. It is also the simplest method when considering the complexity of the calculations.

2 Comparison of three surface condenser connection setups on the cooling water side.

Four condenser connection configurations were tested: I- parallel (Fig.2), II- serial (Fig.3), III- parallel-to-serial (Fig.4) and IV- serial-to-parallel (Fig.5). For proposed thermal cycle (Fig.6) nominal load heat balance was calculated. Next heat balance for 70% and 40% of nominal load was computed. Considering steam flow change and thermodynamic parameters fluctuations, the influence of the tested connections on improving the unit efficiency was verified.

2.1 Calculation procedure [6]

The unit shown as a Figure 6 was described by energy and mass balances equations. The coefficients of the system of equations were appointed by the enthalpy value at the determined points. Enthalpy was calculated from the thermodynamics dependence in accordance with IAPWS IF-97. Exhausted steam pressure was calculating based on algorithm with HEI heat transfer coefficient. By iterating these three calculation steps, the pressure, temperature, enthalpy and mass flow were computed for the determined points at nominal load.

Calculations input data were: \( p_0 \) – live steam pressure, \( t_0 \) – live steam temperature, \( t_{20} \) – reheated steam temperature, \( N_{el} \) – electric power and value needed to exhausted steam pressure calculation.

To compare the results operation following indicators were calculated: gross unit heat rate and unit efficiency.

Using Stodola-Flügel turbine passage equation calculation for 70% and 40% of nominal load were done. Calculations input data were: \( t_0, t_{20}, N_{el} \) and value needed to exhausted steam pressure calculation.

\[
q = 3600 \frac{\dot{Q}_d}{N_{el}} \left[ \frac{kW}{h} \right] \\
\eta = \frac{N_{el}}{\dot{Q}_d} 
\]

Where \( \dot{Q}_d = m_0(\dot{i}_0 - \dot{i}_{100}) + m_{19}(\dot{i}_{20} - \dot{i}_{19}) \)

Comparing the proposed configurations, the following assumptions were made: equal steam distribution to the configurations.
compared after changing parameters: temperature or flow live and reheated steam, cleanliness factor of cooling water, heat exchange surface, temperature of and IV- serial-to-parallel was selected; use of steel 1.4401 -

In table 3.1 and 3.2 input data for the calculations were presented. Following, proposed configurations were compared after changing parameters: temperature or flow of cooling water, heat exchange surface, temperature of live and reheated steam, cleanliness factor

Tabela 3.1 Input data for the calculations (the same for all configuration)

|      |  |  |
|------|---|---|
| $p_0$ MPa | 28.5 | $A_s$ m$^2$ |
| $t_0$ °C | 600 | $m_g \frac{t}{k}$ 81000 |
| $t_{20}$ °C | 610/600° | $\eta_1$ - 0.89 |
| $T_1$ °C | 16.0 | |

Tabela 3.2. Input data for the calculations (different for each configuration)

| configuration | I** | II** | III** | IV** |
|---------------|-----|------|-------|------|
| $A_{s1}$ m$^2$ | 16252 | 16242 | 13935 | 20903 |
| $A_{s2}$ m$^2$ | 16252 | 16242 | 13935 | 13935 |
| $A_{s3}$ m$^2$ | 16252 | 16242 | 20903 | 13935 |
| $N_1$ | 9640 | 25000 | 16840 | 25260 |
| $N_2$ | 9640 | 25000 | 16840 | 16840 |
| $N_3$ | 9640 | 25000 | 25260 | 16840 |

*) Temperature for 40% load
**) Proposed configuration: I-parallel, II-serial, III-parallel-to-serial and IV-serial-to-parallel

Nomenclature: $p_0$ – live steam pressure, $t_0$ – live steam temperature, $t_{20}$ – reheated steam temperature, $m_g$ – cooling water flow rate, $T_1$ – inlet cooling water temperature, $A_s$ – surface tube area $N$ – quantity of tubes $\eta_1$ – LP turbine’s isentropic efficiency

2.2 Result discussion

The results of calculations for 100% and 40% loads are presented in tables 4.1-4.2. Figures 7.1-7.2 show the gross unit heat rate for the nominal load when the value of cooling water flow (7.1) and surface tube area (7.2) was changed. The results were compared with the results from the initial calculations (marked by x).

The best results in thermodynamic terms were obtained for a serial connection. In this case efficiency was improved by 0.15% compared to the parallel connection for nominal load and 0.1% for minimum load. Series-parallel connection was also somewhat more favorable, while other configurations are least beneficial.

Fig. 7.1 Gross unit heat rate tested configuration when $m_g$ change for nominal load

Fig. 7.2. Gross unit heat rate tested configuration when $A_s$ $m_g$ change for nominal load

Figures 7.1-7.2 show the impact of changing the relevant parameters on the gross unit heat rate $q$. On the one hand, the results show that regardless of the tested parameter, the serial system is the most advantageous. On the other hand, the charts show the savings this configuration gives. For example, a similar indicator $q$ for 72000°C cooling water flow for serial configuration was obtained for 90000°C using a parallel configuration (Fig.7.1). This gives a 20% reduction in the amount of cooling water. Figure 7.2 shows that the same indicator $q$ as for the base data series in parallel configuration can be obtained by reducing the surface tube area by 20% for the serial configuration - this can be interpreted as a decreasing surface tube area during operation. Similar conclusions were reached when analyzing subsequent results for the previously described parameters changes.

Table 4.1. Calculations results for nominal load

| configuration | I-parallel | II-serial | III-parallel-to-serial | IV-serial-to-parallel |
|--------------|------------|-----------|------------------------|----------------------|
| $m$ kg/k | $p$ MPa | $m$ kg/k | $p$ MPa | $m$ kg/k | $p$ MPa | $m$ kg/k | $p$ MPa |
| 0 | 2416392 | 28.50 | 2393172 | 28.50 | 2413044 | 28.50 | 2406960 | 28.50 |
| 34 | 496260 | 0.00384 | 492192 | 0.00291 | 495648 | 0.00368 | 494604 | 0.00270 |
| 44 | 482724 | 0.00376 | 478836 | 0.00341 | 482148 | 0.00362 | 481176 | 0.00424 |
| 53 | 432900 | 0.00350 | 429552 | 0.00386 | 432432 | 0.00386 | 431568 | 0.00398 |
| 100 | 2416392 | 32.41 | 2393172 | 32.41 | 2413044 | 32.41 | 2406960 | 32.41 |

Operation indicators

| $q$ | 6915 | 6992 | 6912 | 6907 |
| $\eta$ | 0.521 | 0.522 | 0.521 | 0.521 |
Table 4.2. Calculations results for 40% of nominal load

| Configuration | I-parallel | II-serial | III-parallel-to-serial | IV-serial-to-parallel |
|---------------|------------|-----------|-----------------------|----------------------|
| m             | p          | m         | p                     | m                    |
| i             | kg/h       | MPa       | kg/h                  | MPa                  |
| 0             | 910116     | 10.82     | 903996                | 10.85                |
| 34            | 222300     | 0.00260   | 221796                | 0.00228              |
| 44            | 205992     | 0.00254   | 204624                | 0.00244              |
| 53            | 184248     | 0.00245   | 183024                | 0.00259              |
| 100           | 910116     | 12.31     | 903996                | 12.34                |

Operation indicators

| q         | 7433 | 7416 | 7431 | 7427 |
| η         | 0.484 | 0.485 | 0.484 | 0.485 |

3 Conclusions

This paper presents calculation and verification of which connection setups of surface condensers on the cooling water side is the best from thermodynamics perspective. The study was not easy, because the phenomena occurring in the last stage of the turbine and in the condenser are complex and difficult to describe using mathematical formulas. Therefore, in the first part of work, the focus was on describing and choosing the best method for calculating the exhausted steam pressure of condensing turbine. Three calculation methods were compared, results were verified with data from the factual site. Considering the correctness of the results and the complexity of the calculations, the method based on HEI standard has been identified as the most advantageous method for calculating the turbine exhaust steam pressure. It needs to be highlighted that using this method to calculate heat transfer coefficient requires only the cooling water and condenser technical parameters. There is no need to enter the parameters of exhausted steam what simplifies the calculation.

In the next step, four condenser connection configurations were tested. In each case, the serial configuration was the most thermodynamically favourable. For nominal parameters, obtained improvement of unit efficiency was around 0.15%. The use of this configuration can improve unit efficiency or reduce design or operating costs by reducing surface tube area, cooling water quantity, superheated steam temperature.

However, for a serial connection, the problem of unequal operation of the LP turbine part attention should be paid to. The design of each parts of the turbine is the same, but when serial configuration is used, the exhausted steam pressure of each part is different, so they do not work at their optimal point. This is a significant problem when assuming the work of the unit mainly with nominal parameters. When serial configuration is used, there is also a large dependence of the steam parameters of next LP turbine parts, which is not present for a parallel system. Incorrect assumptions or design calculations may have a greater impact to the operation then in parallel configuration. When choosing a series system, attention should also be paid to the useful power of the condensate pump, which will certainly be higher due to the greater drop in water pressure due to the flow in the tubes.

Summarizing the researches, it has been proven that the most advantageous configuration for thermodynamic reasons is the serial configuration. Although this setup has several important disadvantages that can have a significant impact on the final result.

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