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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, the 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:

$$\dot{Q} = \dot{m}_s(i_s - i_c) \quad (1)$$

$$\dot{Q} = \dot{m}_g Cp_g (T_2 - T_1)\ \frac{1}{\eta_c} \quad (2)$$

Nomenclature: $\dot{Q}$ – heat transfer rate, $\dot{m}_s$– exhausted steam flow rate, i_s– exhausted steam enthalpy, i_c– condensate enthalpy, $\dot{m}_g$– cooling water flow rate, Cp_g – water specific heat, T_1 – inlet cooling water temperature, T_2 – outlet cooling water temperature, η_c– condenser efficiency

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

Condenser heat load equation is:

$$\dot{Q} = UA_s LMTD \quad (3)$$

$$LMTD = \frac{T_2 - T_1}{ln\frac{T_c - T_1}{T_s - T_2}} \quad (4)$$

Nomenclature: $\dot{Q}$ – condenser heat load, U – heat transfer coefficient, A_s– surface tube area, $LMTD$ – logarithmic mean temperature difference, T_c– condensate temperature, T_s– steam temperature, T_1– inlet cooling water temperature, T_2– outlet cooling water temperature.

1.3 Heat transfer coefficient - Characteristic numbers [2].

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

$$U = \frac{1}{R_m + R_t \frac{D_{out}}{D_i} + R_s + R_f} 10^{-3} \quad (5)$$

Tube-Wall resistance was computed as follows:

$$R_m = D_{out}\, ln \frac{D_{out}}{D_i} \frac{1}{2K_m} \quad (5.1)$$

Shellside resistance was computed based on dimensionless equation for heat transfer when the steam condenses on the outside horizontal pipe's surface.

$$R_s = (\frac{Nu\, K_s}{D_{out}})^{-1} \quad (5.2)$$

$$Nu = 0.725\, {C_v}^{0.25} \quad (5.3)$$

$$C_v = \frac{{D_{out}}^3 {\delta_c}^2 g\, {di_{vap}}^2}{K_t \mu_c dT_{ct}} \quad (5.4)$$

dT_{ct}- 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: $\delta_c, K_t,\ \mu_c$ are determined for surface and saturation average temperature $T_f = \frac{T_s + (T_s + dT_{ct})}{2}$

Tubeside resistance was computed based on dimensionless equation for forced convection for turbulent flow inside a circular pipe:

$$R_t = (\frac{Nu\, K_t}{D_i})^{-1} \quad (5.5)$$

$$Nu = 0.021\, Re^{0.8} {Pr_g}^{0.43} \left(\frac{Pr_g}{Pr_t}\right)^{0.25} \quad (5.6)$$

$$Re = \frac{V_g\, D_i \delta_g}{\mu_g} \quad (5.7)$$

$$V_g = \frac{W_g}{\delta_g} \frac{4}{N\, \pi\, {D_i}^2} \quad (5.8)$$

$$Pr = \frac{\mu\, Cp}{K} \quad (5.9)$$

Pr_g / Pr_t – calculated for water temperature T_g and wall temperature $T_t = T_s - dT_{ct}$

Nomenclature: U– heat transfer coefficient, D_i– tube inside diameter, D_{out}– tube outside diameter, K_m– tubewall resistance, K_t– tubeside thermal conductivity, K_s– shellside thermal conductivity, R_m– tubewall resistance, R_t– tubeside resistance, R_s– shellside resistance, R_f– fouling resistance, Nu– Nuselt number, Pr– Prandl number, Re– Reynolds number, T_g– average cooling water temperature, V_g– cooling water velocity, Cp_g– water specific heat, di_{vap}– enthalpy of exhausted steam vaporization, dT_{ct}– The difference in condensate and wall temperatures, μ_c – condensate viscosity, μ_g – cooling water viscosity, δ_g– cooling water density, δ_c– condensate 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

$$U = U_1 F_w F_m F_c \quad (6)$$

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_1values 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:

$$U = \frac{1}{R_m + R_t \frac{D_{out}}{D_i} + R_s + R_f} 10^{-3} \quad (7)$$

Tube-Wall resistance was computed as follows:

$$R_m = D_{out}\, ln \frac{D_{out}}{D_i} \frac{1}{2K_m} \quad (7.1)$$

Tubeside resistance was computed as follows:

$$R_t = (\frac{Nu\, K_t}{D_i})^{-1} \quad (7.2)$$

$$Nu = 0.0158\, Re^{0.835} Pr^{0.426} \quad (7.3)$$

$$Re = \frac{V_g\, D_i \delta_g}{\mu_g} \quad (7.4)$$

$$V_g = \frac{W_g}{\delta_g} \frac{4}{N\, \pi\, {D_i}^2} \quad (7.5)$$

$$Pr = \frac{\mu\, CP}{K} \quad (7.6)$$

Shellside resistance for the first iteration was computed as follows:

$$R_s = \frac{1}{U\, 10^3} - R_m - R_t \frac{D_{out}}{D_i} - R_f \quad (7.7)$$

Shellside resistance for the next iteration was computed as follows:

$$R_s = R_{s0} (\frac{\dot{Q}_0}{\dot{Q}})^{\frac{1}{3}} (\frac{\mu_0}{\mu})^{\frac{1}{3}} \frac{K_{s0}}{K_s} (\frac{\delta_0}{\delta})^{\frac{2}{3}} \quad (7.8)$$

Nomenclature determined as in point 1.3.

Index 0 means the value from the previous iteration. Physical properties of condensate:δ, K, μ are determined for condensate film $T_f = T_s - 0.2LMTD$.

Condenser technical data: Steel 1.4401 - $K_m = 15\ \frac{W}{m\, K}$ was assumed to be used, tube dimension Ø24x0.7mm, condenser efficiency $\eta_c = 0.99$, cleanliness factor k $F_c = 0.95$; 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

$$\eta_1 = \frac{i_{s\_LP} - i_s}{i_{s\_LP} - i_{s0}} \quad (8)$$

Nomenclature: i_{s_LP}– inlet LP turbine steam enthalpy, i_s– exhaust steam enthalpy, i_{s0}– exhaust steam enthalpy when isentropic flow, p_s– exhaust steam pressure,

η_1 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_s– 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, $\dot{m}_g$– 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, $\dot{m}_s$– exhausted steam flow rate, dT_c– condensate subcooling, η_1– turbine efficiency, η_c– condenser efficiency, condenser pass number

Calculation procedure

CN, HEI method	ASME method
- Initialization parameters $\dot{Q}, U$	- Initialization parameters p_s
Calculation	Calculation
- T_2 based on (2)	- $T_s = f(T\_sat(p_s))$, $i_{s\_LP} = f(p_{s\_LP}, T_{s\_LP})$
- $LMTD$ based on (3)	- i_s based on (8)
- T_s based on (4)	- $x_s = f(p_s, i_s)$
- $p_s = f(p\_sat(T_s))$, $i_{s\_LP} = f(p_{s\_LP}, T_{s\_LP})$	- $\dot{Q}$ based on (1)
- i_s based on (8)	- T_2 based on (2)
- $x_s = f(p_s, i_s)$	- $LMTD$ based on (4)
- $\dot{Q}$ based on (1)	- U based on (3)/ (7)
- U based on (5) / (6)	- $LMTD$ based on (3)
Next iteration	- T_s based on (4)
	- $p_s = f(p\_sat(T_s))$
	Next iteration

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: 65MW and 460MW. This paper presents results for

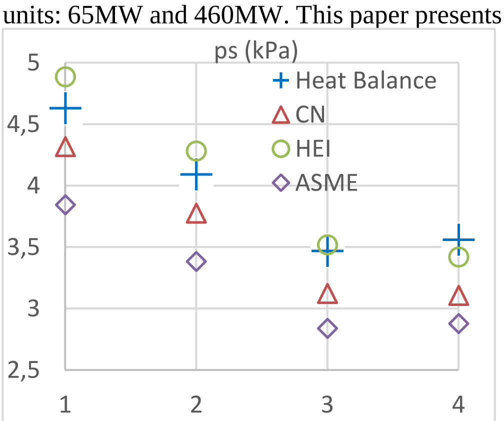

Fig. 1. Exhausted steam pressure

Table 1. Input data for calculations

series		1	2	3	4
load		100%	90%	75%	60%
$\dot{m}_g$	$\frac{t}{h}$	48060	48060	48060	48060
T_1	℃	18.3	16.8	15.2	16.7
$\dot{m}_s$	$\frac{kg}{h}$	753240	697910	587350	491770
T_{s_LP}	℃	278.7	273.6	280	272.6
p_{s_LP}	kPa	579	526	441	358
η_1		0.82	0.82	0.82	0.82

Nomenclature: $\dot{m}_g$– cooling water flow rate; T_1 – inlet cooling water temperature; $\dot{m}_s$ – exhausted steam flow rate; T_{s_LP}– inlet LP turbine steam temperature; p_{s_LP} – inlet LP turbine steam pressure, η_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
CN					
p_s	kPa	4.31	3.77	3.12	3.10
$\dot{Q}$	kW	466641	432688	368136	310295
U	$\frac{kW}{m^2K}$	3.45	3.41	3.38	3.45
T_2	℃	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	$\frac{kW}{m^2K}$	2.59	2.52	2.45	2.52
T_2	℃	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	$\frac{kW}{m^2K}$	5.19	5.11	5.02	5.06
T_2	℃	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

the bigger one. Figure 1 presents calculated exhausted steam pressure compared with reference value. The most similar results to the expected value gave HEI method although results of characteristic numbers method are also correct. The square root error of exhaust 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 on 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 presented in first part of this thesis.

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.

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

$$q = 3600 \frac{\dot{Q}_d}{N_{el}} \left[\frac{kJ}{kWh}\right] \quad (9)$$

$$\eta = \frac{N_{el}}{\dot{Q}_d} \quad (10)$$

Where $\dot{Q}_d = m_0(i_0 - i_{100}) + m_{19}(i_{20} - i_{19})$
Comparing the proposed configurations, the following assumptions were made: equal steam distribution to the

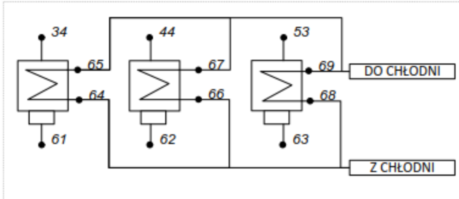

Fig. 2. Parallel configuration

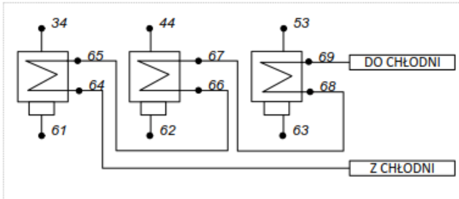

Fig. 3. Serial configuration

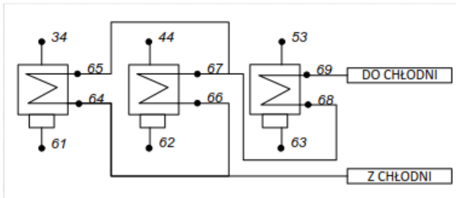

Fig. 4. Parallel-to-serial configuration

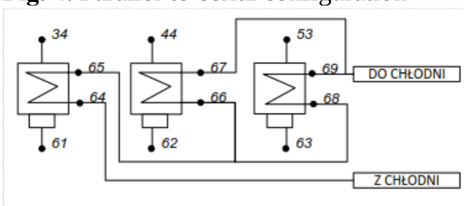

Fig.5. Serial-to-parallel configuration

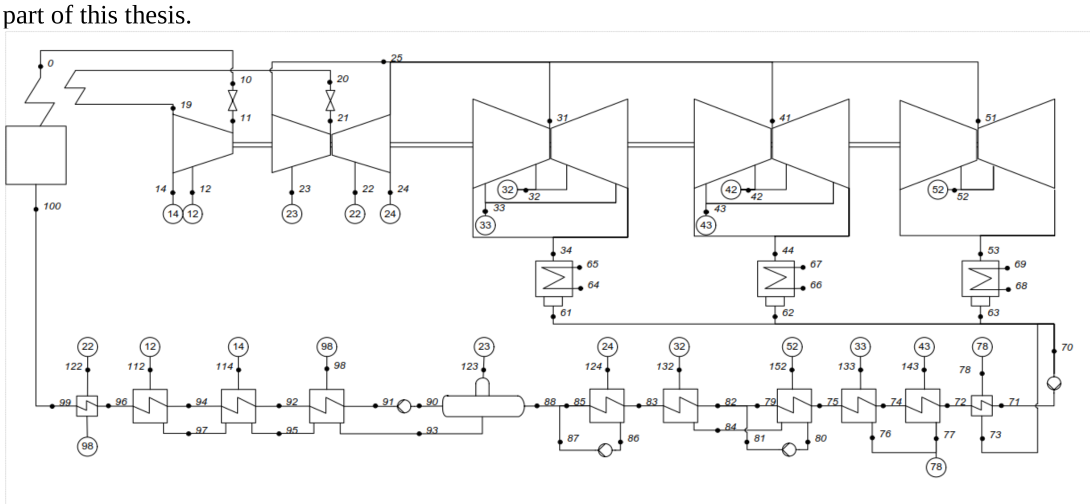

Fig.6. Tested thermal cycle scheme

LP turbine part, the total surface tube in each configuration is similar, the amount of cooling water is the same and the number of tubes has been chosen so that the cooling water velocity does not exceed 2.6 $\frac{m}{s}$.

It was assumed: one pass surface condenser except the parallel configuration where two pass surface condenser was selected; use of steel 1.4401 - $K_m = 15\ \frac{W}{m\,K}$., tube dimension Ø 22 x 0.5 mm, condenser efficiency $\eta_c = 0.99$, cleanliness factor $F_c = 0.95$;

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	~48700
t_0	℃	600	$\dot{m}_g$	$\frac{t}{h}$	81000
t_{20}	℃	610/600*	η_1	-	0.89
T_1	℃	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		19640	25000	16840	25260
N_2		19640	25000	16840	16840
N_3		19640	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, $\dot{m}_g$– cooling water flow rate, T_1– inlet cooling water temperature, A_s– surface tube area N – quantity of tubes η_1 – LP turbine's isentropic efficiency

2.2 Result discusion

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.

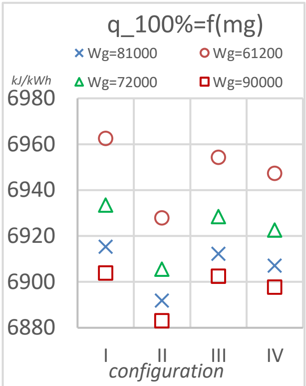

Fig. 7.1. Gross unit heat rate tested configuration when $\dot{m}_g$ change for nominal load

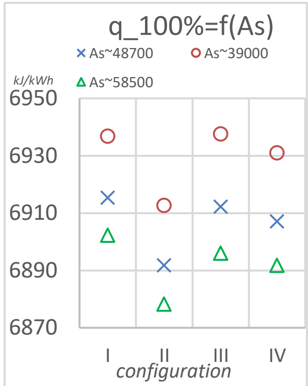

Fig. 7.2. Gross unit heat rate tested configuration when A_s $\dot{m}_g$ change for nominal load

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 $72000\frac{t}{h}$ cooling water flow for serial configuration was obtained for $90000\frac{t}{h}$ 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	p	m	p	m	p	m	p
i	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	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	0.00366	431568	0.00398
100	2416392	32.41	2393172	32.41	2413044	32.41	2406960	32.41
Operation indicators								
q	6915		6892		6912		6907	
η	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	p	m	p
i	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	MPa	$\frac{kg}{h}$	MPa
0	910116	10.82	903996	10.85	909288	10.82	907560	10.83
34	222300	0.00260	221796	0.00228	222264	0.00255	222120	0.00220
44	205992	0.00254	204624	0.00244	205848	0.00249	205452	0.00271
53	184248	0.00245	183024	0.00259	184068	0.00252	183744	0.00262
100	910116	12.31	903996	12.34	909288	12.31	907560	12.32
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 connection 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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