ANALYSIS OF OPERATION OF THE CONDENSER IN A 120 MW THERMAL POWER PLANT

by

Srdjan B. MILIĆ, Milan V. PETROVIĆ*, Milan B. BANJAC, Dejan R. DJUKANOVIĆ, and Srećko M. NEDELJKOVIĆ

Faculty of Mechanical Engineering, University of Belgrade, Belgrade, Serbia

Original scientific paper
https://doi.org/10.2298/TSCI170903242M

The condenser plant has a huge impact on the economy of a steam turbine power plant. Deterioration of the parameters during operation could lead to a significant decrease in electrical output and to an increase in the heat rate of the thermal power plant. Detailed calculations of the performance under different operating conditions were carried out for the condenser of the Morava thermal power plant. Comprehensive testing of the condenser was carried out and experimental data were compared with the numerical results. The effects of deviations in the condenser behavior on the main thermodynamic parameters and the overall economics of the power plant were evaluated. Guidelines for operation of condensation plants are given in the conclusion.

Key words: *steam power plant, condenser, condensation pressure, heat rate, testing*

Introduction

The operation of the condenser has a large impact on the economy of a steam turbine power plant. The pressure in the condenser, which determines the resulting enthalpy decrease and the internal work in the turbine, depends on many parameters [1]. In the literature, papers can be found which examine the effect of particular parameters, such as cooling water mass flow rate [2, 3], cooling water temperature at the condenser inlet [4-7], and turbine exhaust steam mass-flow rate [8] on the condenser performance and the overall heat rate, and power of entire systems.

Deterioration of the operating characteristics of the condenser may lead to a significant decrease in the electrical output and to an increase in the heat rate of the thermal power plant (TPP) [9, 10]. Therefore, an analysis of the condenser plant with the aim of identifying problems and determining the causes of irregular operation is very important. For precise diagnostics, it is necessary to conduct detailed tests of the steam turbine plant and also perform calculations on the behavior of condenser and other systems at the design and at off-design conditions [11-13].

For the condenser of the 120 MW Morava TPP in City of Svilajnac, Serbia, detailed calculations of the performance of the condensing system under different operating conditions have been carried out. Also, comprehensive tests on the condenser were conducted and the experimental data were compared with the numerical results. The impact of deviations in the condenser performance on the main thermodynamic parameters and on the overall economy

* Corresponding author, e-mail: mpetrovic@mas.bg.ac.rs

of the plant was calculated. In conclusion, guidelines for overhauls and operation of the condensation plant are given.

Design parameters of the condenser and the operating conditions

The Morava TPP has been in operation since 1969 with an installed nominal electrical output of 120 MW. Since the first synchronization with the network, the plant has had more than 220.000 operating hours. It uses lignite, brown coal, and hard coal as fuel.

The heat balance diagram of the plant with parameters relevant for the calculation of the condenser is given in fig. 1. A control calculation of the condenser for a given heat transfer surface area, *i. e.*, for a given number and the geometry of the pipe [14], was conducted. The calculation results are given in tab. 1. The in-house codes developed by the Laboratory of Thermal Turbomachinery, University of Belgrade (LTT/UB) were applied for the thermodynamic design and analysis and the condenser (Condpro) and for the calculation of the heat balance diagram of the steam turbine plant (STCycle) [15, 16].

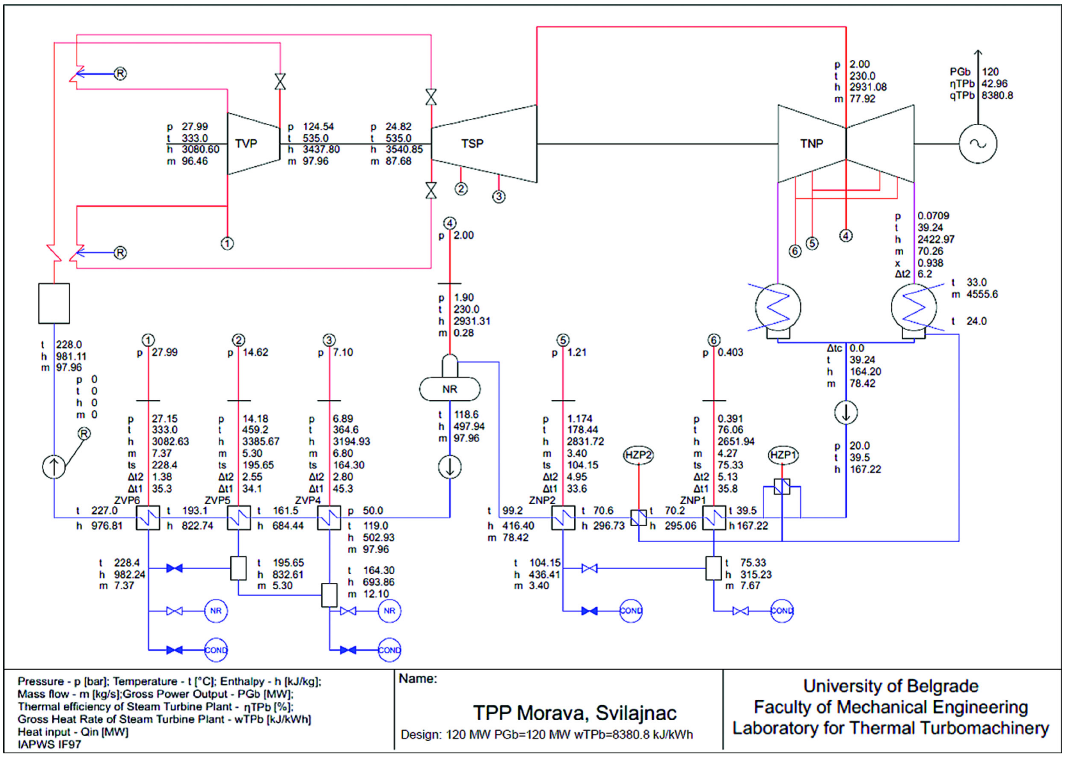

Figure 1. Results of the calculation of the heat balance diagram for the Morava TPP under design conditions (STCycle) (*for color image see journal web site*)

Comparing the condenser design pressure given by the plant manufacturer and the resulting pressure obtained from the control calculations, it can be concluded that the geometry, *i. e.*, the surface area of the condenser, is well determined. A cleanness factor of the tubes of 0.85 is also selected in accordance with the usual practice and corresponds to the operating conditions for average foiled tubes.

Table 1. Input data and calculation results for the condenser in the Morava TPP under design conditions (CONDPRO)

Operating parameters for condenser with P_{Gb} = 120 MW	Unit	Value
Steam flow rate at the condenser inlet	[kgs^{-1}]	70.26
Enthalpy of the steam	[$kJkg^{-1}$]	2422.97
Flow rate of the cooling water	[kgs^{-1}]	4555.6
Temperature of the cooling water at inlet	[°C]	24
Sub-cooling the condensate	[°C]	0.0
Tube material	[–]	Brass
Cleanness factor of tubes	[–]	0.85
Velocity of the cooling water in the tube	[ms^{-1}]	1.94
Number of strokes of the cooling water	[–]	2
Temperature of the cooling water at exit – control calculation	[°C]	33
Condensing pressure – design	[bar]	0.0709
Condensing pressure – control calculation	[bar]	0.0700

Test of the condenser

Since the costs of the burned fuel are increasing from year to year, the power plant management considers options to modernize the plant with the aim of improving the operating performance and to meet new legal requirements. The LTT/UB was engaged to carry out tests of the steam turbine plant [17]. The boiler plant was excluded from this analysis.

The aim of the study was to:
– determine the gross heat rate/thermodynamic efficiency,
– evaluate the reference conditions the turbine plant, and
– identify deviations in the behavior of the individual components.

Based on the results, recommendations for further operation were to be made and a list of measures for future modernization of the plant is to be prepared.

For a full diagnostic test, the possibility of calculating the efficiency/performance of all individual components of the power plant is essential. Therefore, a comprehensive test with a large number of measuring points was undertaken. All measuring points were equipped with high-accuracy instruments to obtain the most accurate information.

In scope of testing the steam turbine plant of the Morava TPP in February 2016, detailed testing the condenser was conducted. Figure 2 shows the heat balance diagram with measuring points for testing the steam turbine plant in the Morava TPP. The tests were performed at the following operating loads: 110 MW (test T110), 105 MW (T105), 90 MW (T90 and T90K), and 80 MW (T80).

Measurements of the following parameters were performed:
– the condenser pressure at seven measuring points,
– the cooling water temperature at the inlet at the two measuring points, and
– the cooling water temperature at the outlet at the four measuring points.

Based on the test results and calculation of the mass and energy balance of the turbine plant, the following results were determined:
– steam mass-flow rate at the inlet to the condenser,

– condition of the steam (enthalpy and entropy) at the inlet to the condenser, and
– mass-flow rate of the cooling water.

The results of the condenser test at the design load are presented in tab. 2.

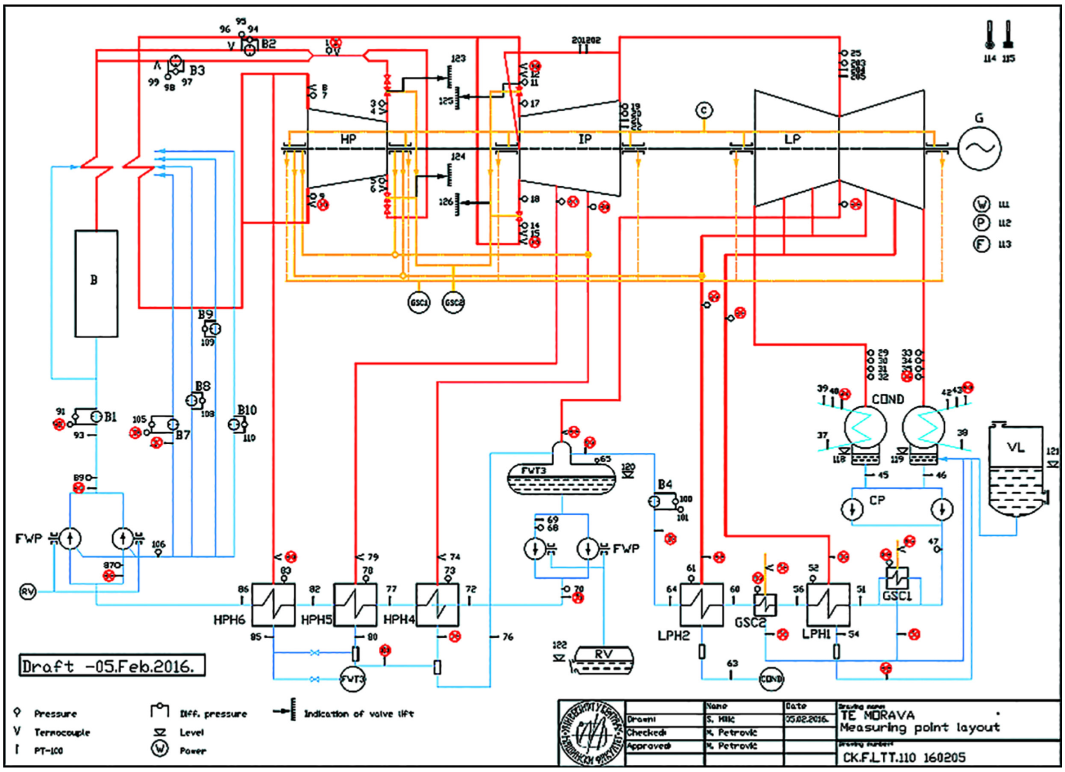

Figure 2. Measuring points diagram for testing the 120 MW Morava TPP
(*for color image see journal web site*)

Operating conditions and their influence on the condenser performance

The parameters that affect the performance of the condenser vary during operation, so the condensation pressure depends on:

– the flow rate and temperature of the cooling water,
– the plant load, *i. e.*, the flow rate and properties of steam at the turbine outlet, which further depends on the turbine efficiency and on the overall condition of the plant, and
– the current state of the condenser, *i. e.*, the active surface for heat exchange, fouling of the tubes, air leakage, quality of vacuum pump operation, *etc.*

Hence, there are a number of parameters that may lead the condensation pressure to be above or below the design value, even in the case of a completely correct state of the condenser and all related equipment. Therefore, deviations in the condenser behavior in everyday operation are difficult to observe. It is possible to locate problems in operation only by detailed tests and calculations.

Evaluation of the test data showed that:

Table 2. Test results for the condenser at different loads and results of calculation of the heat balance diagram for design operating conditions at 120 MW

Parameter	Unit	Design 120	T80	T90K	T90	T105	T110
Steam at condenser inlet, $\dot{m}$	[kgs^{-1}]	70.26	53.00	56.49	58.41	68.46	71.80
Steam at condenser inlet, *p*	**[bar]**	**0.0709**	**0.0519**	**0.0446**	**0.0637**	**0.0765**	**0.0823**
Steam at condenser inlet, t	[°C]	39.24	33.56	30.87	37.25	40.66	42.04
Steam at condenser inlet, h	[kJkg^{-1}]	2423.0	2478.8	2446.0	2484.5	2477.8	2476.9
Condensate from LPH1, $\dot{m}$	[kgs^{-1}]	4.27	2.46	2.90	2.73	3.14	3.30
Condensate from LPH1, $\dot{m}$	[kJkg^{-1}]	315.23	172.96	157.53	186.68	201.55	207.27
Condensate from LPH2, $\dot{m}$	[kgs^{-1}]	3.90	3.66	4.60	4.37	5.07	5.31
Condensate from LPH2, h	[kJkg^{-1}]	436.41	388.17	391.39	397.25	416.95	422.26
Condensate from HPH, $\dot{m}$	[kgs^{-1}]	0.00	0.77	3.81	3.76	0.00	1.17
Condensate from HPH, h	[kJkg^{-1}]	693.9	644.2	651.4	655.4	690.3	699.0
Cooling water inlet, $\dot{m}$	[kgs^{-1}]	4555.6	2384	3441	2394	2365	2382
Cooling water inlet, t	[°C]	24.00	6.77	7.00	6.93	6.68	6.73
Cooling water inlet, h	[kJkg^{-1}]	100.79	28.59	29.56	29.26	28.24	28.45
Cooling water exit, t	[°C]	32.90	19.34	16.35	20.83	22.83	23.56
Cooling water exit, h	[kJkg^{-1}]	141.8	81.29	68.76	87.51	95.87	98.90
Main condensate, $\dot{m}$	[kgs^{-1}]	78.42	59.12	63.98	65.51	76.67	80.41
Main condensate, t	[°C]	39.20	33.05	29.36	36.80	39.86	41.13
main condensate, h	[kJkg^{-1}]	164.20	138.48	123.09	154.17	166.95	172.27
Increase in cooling water temperature, Δt_1	[°C]	8.00	12.58	9.35	13.90	16.15	16.82
Temperature difference of condenser, Δt_2	[°C]	7.2	14.2	14.5	16.4	17.8	18.5
Sub-cooling of the main condensate	[°C]	0.04	0.51	1.51	0.45	0.8	0.9
Heat exchanged in condenser	[MW]	160.91	125.63	134.87	139.44	159.99	167.84

– the condenser pressure at all operating points was near the design value of 0.07 bar, but considerably higher than the expected values for the current cooling water temperature (about 7 °C), which was much lower than the design value (24 °C) and
– the cooling water-flow rate obtained by calculation from the balance was considerably lower than the design value (4555.6 kg/s).

Cooling water-flow rate

The cooling water mass-flow rate was determined applying two methods:
– from the test results based on the energy balance of the plant and
– by measurement of the cooling water mass-flow rate with an ultrasonic flow meter.

A comparison of the cooling water-flow rate calculated from the energy balance of the plant and directly measured with the ultrasound flow meter is presented in tab. 3.

Table 3. Comparison of the cooling water-flow rate calculated from the energy balance of the plant and directly measured with the ultrasound flow meter for tests at different loads

Parameter	Unit	T80	T90K*	T90	T105	T110
Cooling water – measurement, $\dot{m}$	[kgs^{-1}]	2410	3618	2412	2415	2417
Cooling water – energy balance of condenser, $\dot{m}$	[kgs^{-1}]	2391	3457	2401	2374	2395

* During the test T90K, two cooling water pumps were in operation. In other tests, only one pump was in operation.

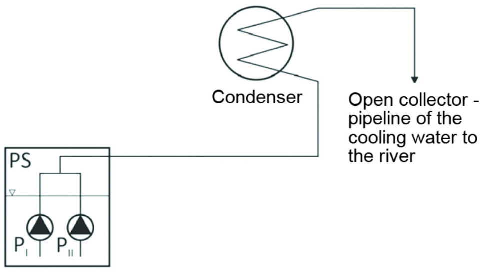

Figure 3. Simplified scheme of pump stations and the cooling water system in the Morava TPP

In standard operation of the plant, during the winter period with a low cooling water temperature only one pump is in operation. However, from the results (tab. 3), it is clear that the designed cooling water-flow rate (4555.6 kg/s) can not be achieved even if both pumps are in operation.

Figure 3 presents a simplified diagram of the pump station and the cooling water system in the Morava TPP, and fig. 4 shows the pump performance diagram with one and two pumps connected with pipelines. Two centrifugal pumps were installed in the cooling water pump station. The declared operating parameters for the pumps are Q = 2.4 m^3/s, H = 13.5 m, and n = 480 min^{-1} and for the electric motor P_{E1} = 450 kW and η_{E1} = 0.92 [14]. The pumps are mutually interconnected in such a way that either of the two pumps may operate in a stand-alone mode or, if an increase in the cooling water-flow rate is required, the pumps may operate in a parallel mode.

Figure 4 shows a diagram of the coupling of the cooling water pumps in stand-alone or parallel mode. The pump characteristics of head, power and efficiency as a function of flow rate were established based on the test report for the pumps [18]. The approximate operating characteristic of the pipeline is obtained based on the cooling water-flow rate measured with the ultrasonic flow meter which was installed on the cooling water pipeline. The measured values were $Q \cong 2.4$ m^3/s in the pump stand-alone operation and $Q \cong 3.6$ m^3/s in the parallel mode of two-pump operation: (tab. 3).

In the case of the pump stand-alone operation (operating point: OP$_{PI}$), the efficiency was $\eta_{(OPPI)}$ = 91% and the pump power was 365 kW. In the parallel mode of the two pumps (operating point OP$_{(I+II)}$), the flow rate of each pump was Q = 1.8 m^3/s. For more precise analysis of the pumps in the cooling water system, it is necessary to perform calculations on the basis of data collected in the power plant (the declared performance by the manufacturer or the data obtained from the on-site tests, disposition of the pump station and the levels) and corresponding pipes and valves in the cooling water system (dimensions, elevation, suction strainers, valves, elbows, *etc.*). However, for the present investigation, the results obtained in this way were sufficient.

It is obvious that the selected cooling water pumps in parallel mode can not achieve the design cooling water mass-flow rate of 4555.6 kg/s. In the existing situation, a cooling water flow rate of 2400 kg/s may be realized if one pump operates, or 3600 kg/s if two pumps operate. This deficit in the cooling water-flow rate causes a significant increase in the condenser pressure, tab. 4, to about 10-30 mbar compared with the design value (0.0709 mbar).

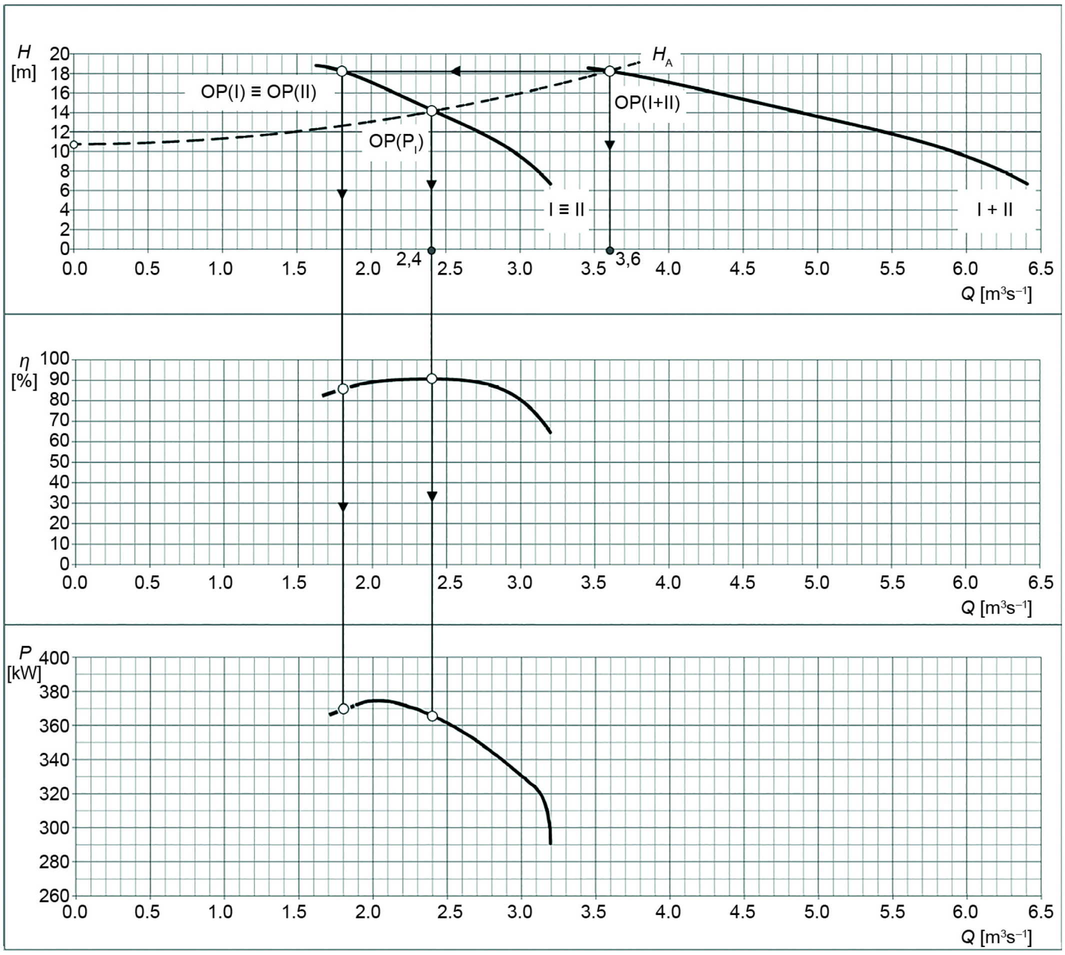

Figure 4. Diagram of coupling of one or two cooling water pumps

Table 4. Calculation of the condensation pressure at the design parameters for different cooling water flow rates

	Design	Present: two pumps in operation	Present: one pump in operation
Cooling water flow rate [kgs^{-1}]	4555.6	3600	2400
Pressure in condenser [bar]	0.0709	0.081	0.100

Current state of the condenser

In order to separate the effect of deterioration of the condenser pressure due to the reduced flow of cooling water from the effects caused by the current state of the condenser, calculations of the condenser performances for different loads were performed using the software package CONDPRO. A cooling water temperature of 7 °C, which was measured during the plant tests in February 2016, was taken as reference.

The calculated results were compared with the test results, figs. 5 and 6. It can be seen that the condenser pressure was 10-20 mbar higher than the value that would have been achieved if the condenser, in the given circumstances, had operated with the design performance.

In fig. 7, a comparison between the measured and calculated values of the temperature difference, Δt_2, between the steam condensation temperature and the cooling water temperature at the condenser outlet is presented. It can be seen that the values obtained in the tests were considerably higher, by approximately 8-10 °C, than the design values. The reasons for this are:

– reduced heat transfer coefficient on the water side of the tubes,
– leak and air penetration into the condenser, and
– fouling of tubes and reduced surface area for heat exchange.

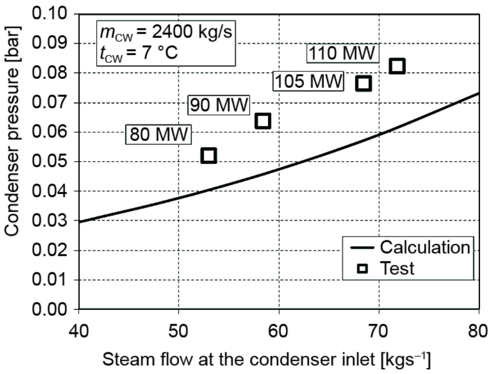

Figure 5. Condensation pressure at a cooling water-flow rate of 2400 kg/s; measured data and calculation results for the given cooling water flow rate and a temperature of 7 °C

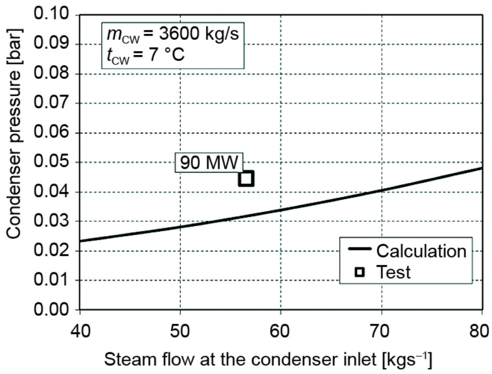

Figure 6. Condensation pressure at a cooling water-flow rate of 3600 kg/s; measured data and calculation results for the given cooling water flow rate and a temperature of 7 °C

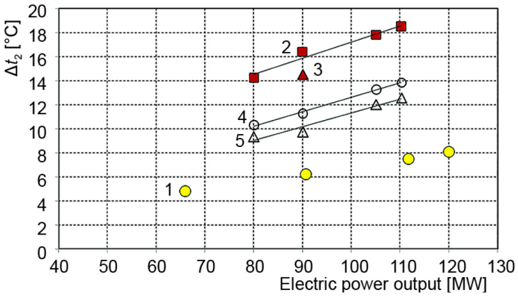

Figure 7. Comparison of the measured and calculated values of the temperature difference, Δt_2, between the steam condensation temperature and the temperature of the cooling water at the condenser outlet; *1 – design – cooling water temp 24 °C, 2 – test – operation with 1 vacuum pump and 1 cool water, cooling water temp 7 °C, 3 – test – operation with 2 vacuum pump and 2 cool water, cooling water temp 7 °C, 4 – calculation – operation with 1 cool water, cooling water temp 7 °C, 5 – calculation – operation with 2 cool water, cooling water temp 7 °C*

From fig. 7, it is seen that, owing to the decrease in the cooling water-flow rate, Δt_2 increases by about 4 °C at a flow rate of 3600 kg/s or about 6 °C at a flow rate of 2400 kg/s compared with the design state. This effect was caused by the decrease in of the cooling water velocity in the pipes, which was reduced from the design value of 1.94 m/s to 1.53 m/s at a flow rate of 3600 kg/s and to 1.02 m/s at a flow rate of 2400 kg/s. This significantly decreases the heat transfer coefficient from the water side and, as a consequence, increases Δt_2, *i. e.*, the condensation pressure.

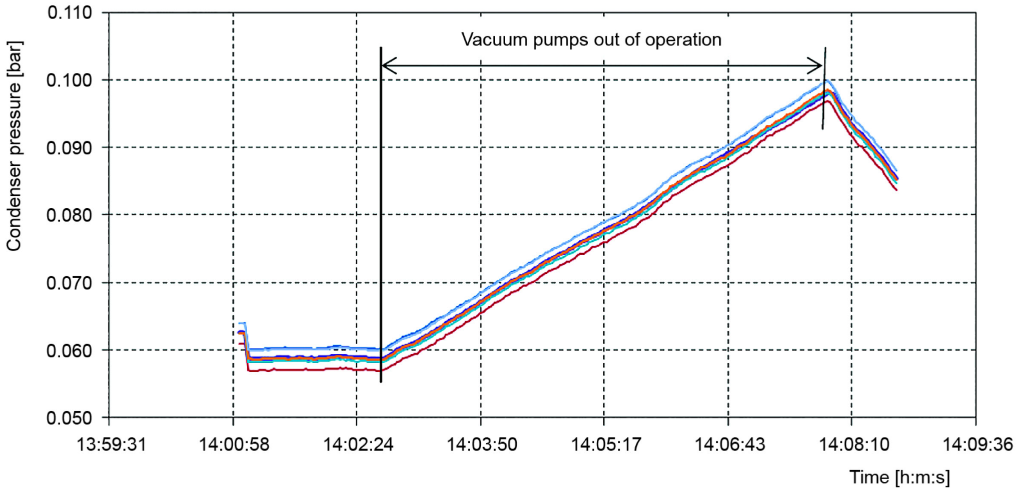

Figure 8. Condenser air leakage test. The increase in the condenser pressure with the vacuum pumps out of operation was 7.45 mbar per minute, while the recommended value is 2.67 mbar per minute

The second group of problems involve leakage and penetration of air into the condenser. Figure 8 shows a diagram for a test with a change/increase in the condenser pressure if the vacuum pumps were out of operation. The results show that this increase was about three times higher than allowed. The vacuum pumps are not able to evacuate such a large air intake in a satisfactory manner since they are not designed for such a large capacity.

The third group of problems are:

– the reduced condenser area because a relatively large number of tubes are closed due to damage and
– tubes fouled by deposits on the water side, which cause a decrease of the heat transfer coefficient.

It should be noted that the condenser tubes have not been replaced since the plant came into operation, and that here there is a considerable potential for improvement of the plant.

Impact of the higher pressure of the condenser on the plant performance

The design cooling water flow rate can not be achieved in the existing plant without major reconstruction. Therefore, the value of the cooling water-flow rate here that can be achieved by two pumps in parallel operation, *i. e.* 3600 kg/s, was taken as reference.

Table 5 presents the calculated impact of the cooling water-flow rate and the state of the condenser on the most important performance parameters of the plant: the electrical power output and the gross heat rate. The calculation was performed using the software package STCycle [15]. It can be seen that the electrical power is reduced by about 3.5 MW due to the condenser state (air leakage, smaller area for the heat exchange, and fouling and deposits) and to about 2.3 MW due to the decreased cooling water-flow rate (2400 instead of 3600 kg/s).

Table 5. Influence of deviations of the operating conditions from the design values in the condenser during the test of the steam turbine plant (test T110)

Parameter	Unit	Design*	Test	Difference	Deficit in electrical power		Increase in heat rate	
					[MW]	[%]	[kJkW^{-1}h^{-1}]	[%]
Condenser pressure – one CW pump in operation	[bar]	0.0570	0.0823	0.0253	–3.57	–3.10	269.41	3.03
Condenser pressure – two CW pumps in operation	[bar]	0.0400	0.0570	0.0170	–2.42	–2.12	181.37	2.02
Condenser – overall	[bar]	0.0400	0.0823	0.0423	–5.99	–5.15	450.5	5.12

* It was assumed that the design cooling water-flow rate is 3600 kg/s; it is possible to achieve this value with the existing pumps and pipeline system.

Calculation of the condenser performance for a wide range of operating conditions

As already stated, the design cooling water-flow rate of 4555.6 kg/s can not be realized with the existing pumps and pipeline system. Therefore, using the software package CONDPRO [16], calculations on the condenser were carried with one pump (flow rate 2400 kg/s) and two pumps (flow rate 3600 kg/s) in operation and at different temperatures of the cooling water, loads and the cleanness factor of the tubes. The results are shown in fig. 9. The diagrams are aimed for use in the operation of the plant. Comparing the results of the calculation from fig. 9 and the condenser pressure achieved at different operating conditions, the staff in the power plant will be able to draw conclusions regarding the state of the condenser, *i. e.*, to define maintenance measures if necessary.

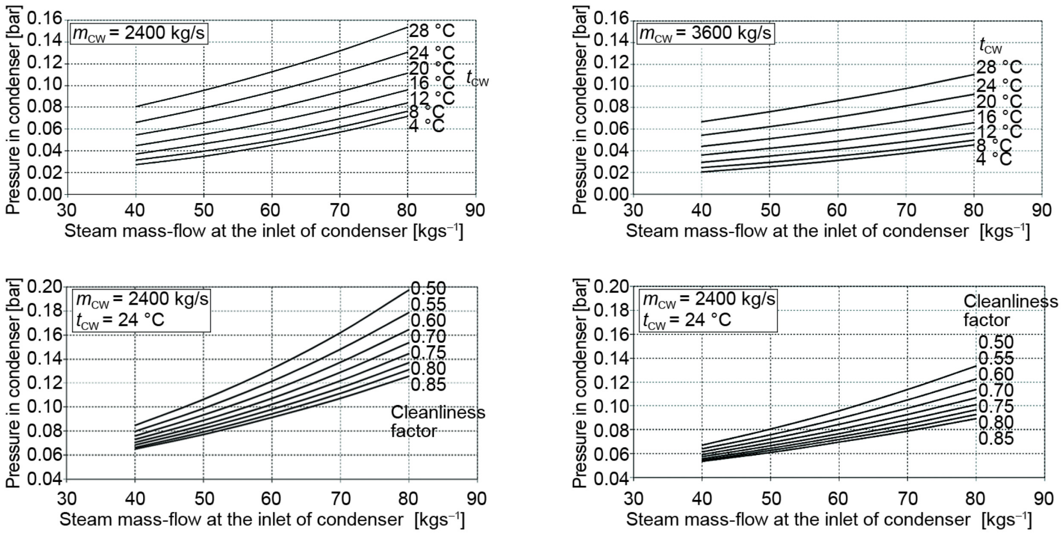

Figure 9. Behaviour of the condenser when one CW pump (flow rate 2400 kg/s) and two CW pumps (flow rate 3600 kg/s) are in operation and at different CW temperatures, loads and cleanness factor of the condenser

Conclusions

A detailed analysis of the operation of the condensing plant in the 120 MW Morava TPP was carried out. Parallel to the calculations, comprehensive tests of the power plant were conducted and the experimental data were compared with the theoretical results. It was found that the pressure in the condenser had been significantly increased. At a load of 110 MW (test T110), the measured condenser pressure was 82.3 mbar instead of the 40 mbar that would be achieved if the condenser operated according to the design. This deterioration of the condenser performance caused a deficit in the plant electrical output of more than 5 MW during the test at 110 MW (T110). The reasons for this lies in the decreased flow rate of the cooling water and the poor state of the condenser because the pipes had not been replaced since the original commissioning in 1969. The cooling water-flow rate was reduced because only one pump (instead of two) was in operation. The considerable air penetration due to leakages, the decreased surface area for heat exchange due to the plugged tubes, tube deposits, and fouling lead to additional steam pressure increases in the condenser. All this causes a large decrease in the plant electrical power and an increase in the heat rate. The analyses of the Morava TPP show that it is necessary to operate with two cooling water pumps, even at low cooling water temperatures, and to carry out a major overhaul of the condenser in which the tubes should be replaced and the leakages and air penetration eliminated.

The procedure applied here which consists in a comprehensive testing combined with a detailed theoretical analysis and a calculation of the condenser behaviour over a wide range of loads is able to detect successfully deviation and problems in operation of condensers in steam turbine power plants. Solving of the found problems may significantly improve thermodynamic and financial performances of the power plants.

Nomenclature

H – head [m]
h – enthalpy [$\mathrm{kJkg^{-1}}$]
$\dot{m}$ – mass-flow rate [$\mathrm{kgs^{-1}}$]
n – rotation speed [$\mathrm{s^{-1}}$]
P – electrical output, power [MW]
p – pressure [bar]
Q – volume flow rate [$\mathrm{m^3s^{-1}}$]
t – temperature [°C]
Δt_1 – increase in the CW temperature [°C]
Δt_2 – temperature difference in the condenser [°C]
x – wetness [–]

Greek symbol

η – efficiency [–]

Subscripts

I – first
II – second

Acronyms

CW – cooling water
El – electrical
HPH – high-pressure heater
LPH – low-pressure heater
OP – operating point
P – pump
PS – pump station

References

[1] De Souza, G. F. M., *Thermal Power Plant Performance Analysis,* Springer, London, 2014
[2] Ganan, J., *et al.*, Influence of Cooling Circulation Water on the Efficiency of a Thermonuclear Plant, *Appl. Therm. Eng.*, *25* (2005), 4, pp. 485-495
[3] Harish, R., *et al.*, Theoretical Model for Evaluation of Variable Frequency Drive for Cooling Water Pumps in Sea Water Based Once through Condenser Cooling Water Systems, *Appl. Therm. Eng., 30* (2010), 10, pp. 2051-2057
[4] Haseli, Y., *et al.*, Optimum Temperatures in a Shell and Tube Condenser with Respect to Exergy, *Int. J. Heat Mass Transfer, 51* (2008), 9, pp. 2462-2470

[5] Atria S. I., The Influence of Condenser Cooling Water Temperature on the Thermal Efficiency of a Nuclear Power Plant, *Ann. Nucl. Energy, 80* (2015), 6, 371-378
[6] Laskowski, R., *et al.*, Cooperation of a Steam Condenser with a Low-Pressure Part of a Steam Turbine in Off-Design Conditions, *Am. J. Energy Res., 3* (2015), 1, pp. 13-18
[7] Durmayaz, A., Sogut, O. S., Influence of Cooling Water Temperature on the Efficiency of a Pressurized-Water Reactor Nuclear-Power Plant, *Int. J. Energy Res, 30* (2006), 10, pp. 799-810
[8] Wang, W. *et al.*, Feasibility Analysis of Changing Turbine Load in Power Plants Using Continuous Condenser Pressure Adjustment, *Energy, 64* (2014), Jan., pp. 533-540
[9] Harpster, J. W., Putman, R. E., The Economic Effects of Condenser Backpressure on Heat Rate, Condensate Sub-Cooling and Feedwater Dissolved Oxygen, *Proceedings,* 2000 International Joint Power Generation Conf., Miami Beach, Fla., USA, 2000
[10] Walker M. E., *et al.* Economic Impact of Condenser Fouling in Existing Thermoelectric Power Plants, *Energy*, *44* (2012), 1, pp. 429-437
[11] Putman, R. E., Jaresch, D., The Cleaning of Air Cooled Condensers to Improve Performance, *Proceedings,* 2002 International Joint Power Generation Conf., Scottsdale, Ariz., USA, 2002
[12] Saxon, G. E., Putman R. E., Improved Plant Performance and Reduced CO_2 Emission through State-of-the-Art Condenser Cleaning and Air In-Leakage Detection, *Proceedings,* 2003 International Joint Power Generation Conf., Atlanta, Geo., USA, 2003
[13] Roy, R. P., Computational Model of a Power Plant Steam Condenser, *J. Energy Resour. Technol*, *123* (2000), 1, pp. 81-91
[14] ***, Morava TPP Technical Documentation
[15] ***, STCycle – A Code for Heat Balance Calculation and Optimization of Steam Turbine Plants, University of Belgrade, Faculty of Mechanical Engineering, Laboratory of Thermal Turbomachinery, Belgrade, 2015
[16] ***, CONDPRO – A Code for Design and Analysis of Condensers in Steam Turbine Power Plants, University of Belgrade, Faculty of Mechanical Engineering, Laboratory of Thermal Turbomachinery, Belgrade, 2015
[17] Petrovic, M., *et al.*, Thermal Tests and Analysis of Operation of the Steam Turbine Plant in 120 MW Morava TPP , Report LTT-02/16, IC Faculty of Mechanical Engineering, Belgrade, 2016
[18] ***, Test Report: The Centrifugal Pump 95Vi/A Litostroj-Ljubljana, Institute for Turbines, Ljubljana, 1967

Paper submitted: September 3, 2017
Paper revised: November 4, 2017
Paper accepted: November 10, 2017