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In-situ effectiveness of cooling water flow at different operating loads: A case study of a 1070MW ultra-supercritical steam power plant

Didi Supandi*, Eflita Yohana, Tony Suryo Utomo, Syarif Dwi Priyanto, Muhammad Bagir

ABSTRACT

The cooling water systems play a vital role in large steam power plants as they directly affect the condenser back pressure, turbine efficiency, and overall plant heat rate. This paper presents an in-situ detailed case study of the performance of the cooling water (CW) system in a 1070 MW ultra-supercritical coal-fired steam power plant at three steady-state load levels, i.e., 100% (1070 MW), 75% (803 MW), and 50% (535 MW). Actual operating data obtained from the Distributed Control System (DCS), such as condenser pressure, CW inlet/outlet temperatures, temperature rise (ΔT), CW mass flow rate, exhaust steam flow, heat duty, logarithmic mean temperature difference (LMTD), and total heat transfer coefficient (U), are reviewed and compared to results from a verified steady-state simulation model. The research results show a deviation between the measured and simulated condenser pressure values, with a validation margin of less than 5% for the main parameters. This indicates that cooling water (CW) flow varies under partial load conditions to avoid excessive auxiliary power consumption, and provides quantitative insights for load-dependent optimization of cooling water (CW) flow. The condenser pressure at 50% load is 60.6 mbar, considerably less than the projected 73 mbar, thus affirming the existence of surplus cooling capacity and the opportunity to decrease pump flow. These results are useful in developing better operational strategies for increasing thermal efficiency and decreasing specific auxiliary consumption of large-scale fossil-fuel power plants.

Keywords: Part-load performance, Cooling water flow, Steam power plant, Surface condenser, Condenser backpressure, Thermal optimization, Simulation validation, DCS data.

1. INTRODUCTION

Steam power plants are still a major source of electricity generation worldwide and continue to meet the increasing global energy demand (Khattak et al., 2016;

Yohana et al., 2025). In a steam Rankine cycle, the primary method of heat removal is through the condenser, which uses a cooling water (CW) system for heat exchange. This directly affects the turbine exhaust gas pressure and the overall thermal efficiency of the power plant (Mohammed et al., 2019; Pořkas et al., 2024). A gradual increase in condenser pressure of approximately 1 kPa can reduce power plant output by about 0.5–1% and also increase the heat ratio, depending on the turbine design (Pattanayak et al., 2019).

The performance of the condenser is influenced by several parameters, namely the cooling water (CW) mass flow rate, the cooling water (CW) inlet temperature, the tube-side fouling conditions, the exhaust steam flow and quality, and the overall heat transfer coefficient U (Howell and Saxon, 2017; Putman and Harpster, 2002). Previous studies have shown that high cooling water (CW) inlet temperature or gradual fouling can significantly reduce thermal efficiency. Ibrahim and Attia (2015) found that increasing seawater temperature caused nuclear facilities to lose 2–3% of their efficiency. Li et al., (2016) has developed an online scaling monitoring model for condensers in coal-fired power plants. Wang et al., (2012) proposes a model in which the cleanliness coefficient decreases over time to improve the optimization of cooling water circulation.

For large utility units, the operating plant load is often varied in response to grid demand, economic dispatch, or maintenance schedule. At reduced load, the flow of exhaust steam is approximately proportional to the reduction in load, with a corresponding reduction in the heat duty of the condenser. However, many facilities function with nearly constant cooling water (CW) flow settings, causing underutilization of heat-transfer driving force, an overabundance of cooling margin, and excessive use of auxiliary pump power. Cooling water pumps can consume 1–3% of gross plant output; Anozie and Odejebi, (2011) and Laskowski et al., (2020) have shown that variable cooling water (CW) flow control can save 10–30% of the pump power at partial load; however, few publications present in-situ validation data for large units (>800 MW).

Another operational problem is fouling of condenser tubes. Howell and Saxon, (2017) classified fouling of condenser tubes into five categories: microbiological, scale, particulate deposition, corrosion product, and macrofouling, with each having different mitigation strategies. Cregan and Flynn, (2006) used genetic-algorithm optimization to develop a composite, non-linear fouling model with weighted sub-models to quantify economic maintenance costs. Putman & Harpster (2002) showed that fouling losses and air-ingress losses must be measured separately, since they have similar effects on performance but different paths of remediation. Wang et al., (2016) developed experimental correlations of CaCO_3 fouling characteristics with flow velocity, in laminar, transitional, and turbulent regimes in smooth tubes, which offer quantitative guidance for selection of optimal cooling water (CW) velocity.

On-line and off-line cleaning technologies have also been assessed. Ma et al., (2018) demonstrated that rubber-ball on-line cleaning devices reduce fouling resistance by 74.5% and increase chiller COP by up to 10.2% with a dynamic pay-back period of 1.5 years. Nebot et al., (2006) did in-situ experiments at a 550 MW coastal station, and they found that a residual chlorine concentration of 0.2 mg/L decreases biofouling accumulation by about 90%, which is the accepted design cleanliness factor of 85%.

Several researchers have advocated for the simulation and off-design modeling of condenser systems. Wróblewski et al., (2013) analyzed parallel versus serial configurations of condensers for a 900 MW ultra-supercritical unit utilizing the HEI model, discovering that a serial configuration combined with an asymmetric LP turbine enhances cycle efficiency by 0.26 percentage points. Qureshi and Zubair, (2016) proposed a predictive equation based on dimensional analysis for UA degradation in Rankine cycle power generation systems. They demonstrated a logarithmic decline in performance due to simultaneous scaling in the boiler and condenser. Laskowski (2016) developed universal off-design correlations for the operational efficacy of steam condensers based on input characteristics that offer a pragmatic framework for real-time performance assessment. Hu et al., (2024) analyzed the thermal performance of a combined cycle gas-steam system with deep cooling in the condenser, demonstrating the feasibility of incorporating cooling water (CW) side optimization into the overall cycle design.

However, there is still a practical gap in the literature: few published in-situ case studies with DCS-validated simulation findings for very large (>1000 MW) ultra-supercritical units at numerous discrete load levels. This study fills this gap by looking at the in-situ efficacy of cooling water flow in a 1070 MW ultra-supercritical coal-fired steam power plant under three operational loads (100%, 75%, and 50%) using actual DCS data integrated with simulation. The particular objectives are: (i) To record measured condenser and cooling water (CW) parameters at all load levels; (ii) To quantify the deviations between measured data and simulation predictions; (iii) To assess the effectiveness of the current cooling water (CW) flow strategy; (iv) To identify opportunities for load-dependent cooling water (CW) flow optimization to reduction auxiliary power consumption.

Working Principles of Condenser

The surface condenser is a single- or double-pass shell-and-tube heat exchanger that collects exhaust steam from the LP turbine and condenses it on the outside of thousands of cooling water tubes. The heat rejection rate Q is governed by the energy balance between the latent heat release on the steam side and the sensible heat gain of the cooling water (CW). The turbine backpressure (and thus the available specific work output per kg of steam) is determined by the condenser saturation pressure, which is a function of the cooling water (CW) inlet temperature, cooling water (CW) flow rate, area of heat transfer A , and overall heat transfer coefficient U (Pattanayak *et al.*, 2019; Laskowski *et al.*, 2020).

U is computed by the standard HEI method:

$$U = U_1 \times F^W \times F^M \times F_C \quad (1)$$

U : Heat Transfer Coefficient (W/m²K)

U_1 : Uncorrected Heat Transfer Coefficients

F^W : Inlet Water Temperature Correction Factor

F^M : Tube Material and Gauge Correction Factor

F_C : Cleanliness Factor

U_1 is the base coefficient taken from HEI standard curves (function of cooling water (CW) velocity and tube outer diameter), F^W is the cooling water (CW) inlet temperature correction factor, F^M is the tube material and gauge correction factor, and F_C is the cleanliness factor which reflects fouling state. Progressive fouling reduces F_C below the design value (usually 0.85 for once-through seawater cooling) while increasing condenser pressure and turbine heat rate.

Issues of Part-Load Operation

As the load is reduced, the exhaust steam flow decreases roughly proportionally, as does the condenser heat duty. If the flow of the cooling water (CW) is maintained constant, the cooling water (CW) temperature increase, ΔT , is decreased, and this leads to excess cooling margin. This means that CW pumps have unnecessary auxiliary power, which can consume 1 to 3% of gross plant output (Anozie and Odejebi, 2011). Control strategies for variable cooling water (CW) flow, e.g., pump speed modulation or valve throttling, can potentially save 10-30% of the pump power at part-load (Anozie and Odejebi, 2011; Laskowski *et al.*, 2020). The optimum trade-off between condenser performance (low backpressure) and pump power, however, should be found using plant-specific in-situ data and a validated thermal model.

2. METHODS AND MATERIALS

System Description

This case study is an ultra-supercritical coal-fired power generation unit with a capacity of 1,070 MW, operating at nominal main steam conditions of approximately 26.5 MPa and 600°C, equipped with a reheating system. The condenser features a single-shell, double-flow design and is equipped with a single-pass cooling water system that draws water from a nearby source. The nominal cooling water inlet temperature is 30°C, with a design temperature rise of approximately 9–11°C at full load. Table 1 summarizes the main design parameters of the condenser.

Table 1. Condenser design specifications

Parameter	Design Value
Type	Steam surface condenser
Heat transfer surface area	52,606 m ²
Design heat duty	1,117.0 MW
Design exhaust pressure (vacuum)	85.7 mbar(abs)
Nominal CW volumetric flow	143,910 m ³ /h
CW inlet temperature (design)	30°C

Parameter	Design Value
Number of CW pumps (full load)	2
Design cleanliness factor (F_C)	0.85

Illustrates the schematic layout of the steam turbine–condenser system at the study plant, showing the LP turbine exhaust connections, the shell-and-tube condenser, the cooling water inlet and outlet water boxes, and the CW pump configuration. Fig. 1 shows an illustration of steam turbine with condenser.

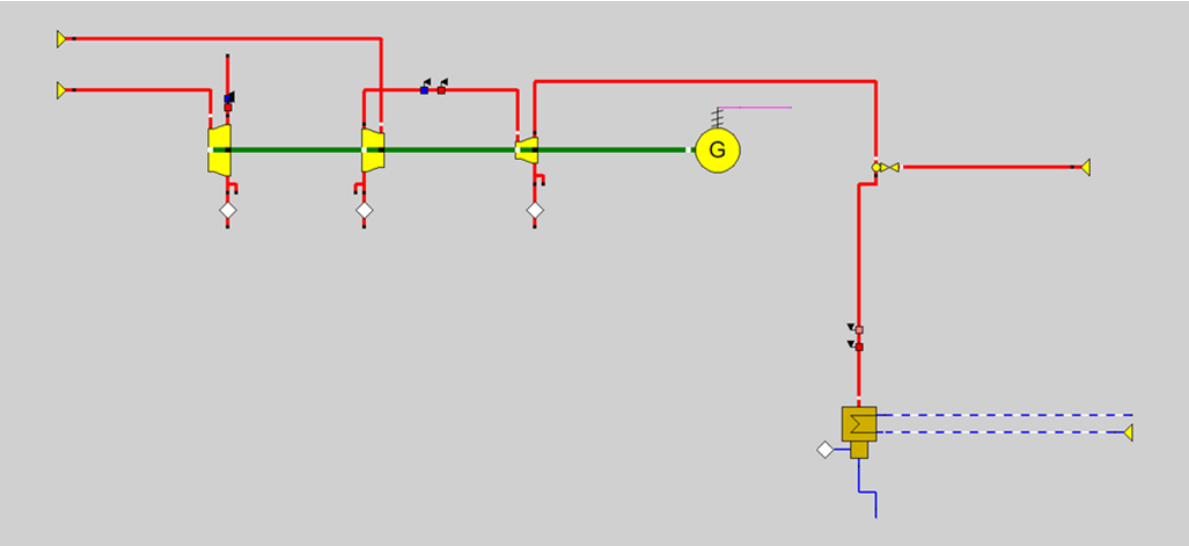


Figure 1. Illustration of Steam Turbine with Condenser

Operating Load Conditions & DCS Data Collection

The investigation was conducted at three different steady-state load levels: 100% (1070 MW), 75% (803 MW), and 50% (535 MW). For each load, operational data were collected over time windows representing steady, transient-free operation from the plant Distributed Control System (DCS). Within each window, the following parameters were recorded and averaged:

- a. Gross and net generation (MW)
- b. Absolute condenser pressure (mbar).
- c. Cooling water temperature at the inlet and output (°C).
- d. The number of operational CW pumps and the CW volumetric flow rate (m3/h).
- e. Percentage of valves open at the water box outlet pipe (%)
- f. Temperature of saturation at measured condenser pressure (°C).

Data pre-processing involved statistical filtering of outliers caused by sensor noise or short-term process disturbances, so that only representative steady-state values were used for the following analysis. Table 2 shows summarize the simulation scenarios tested.

Table 2. Simulation scenario matrix based on load and cooling water (CW) flow settings

Load Gross (MW)	CW Pumps Running	Outlet WB Valve Opening (%)	CW Flow Rate (m³/h)	CW Inlet Temp (°C)
1070 (100%)	2	52	143,910	30
803 (75%)	1	38	93,705	30
535 (50%)	1	38	93,705	30

Thermal Performance Formulation

The condenser thermal balance is formulated using standard energy conservation between the steam and cooling water streams (Pattanayak et al., 2019; Laskowski et al., 2020). The total heat rejection rate is expressed as:

$$\dot{Q} = \dot{m}_{cw} (C_p)_{cw} (T_{cwout} - T_{cwin}) = \dot{m}_s (h_s - h_{cond}) \quad (1)$$

Based on the equation, the value of cooling water heat absorption is obtained using the following equation (2) (Attia, 2015):

$$\dot{Q}_{cw} = \dot{m}_{cw} (C_p)_{cw} (T_{cwout} - T_{cwin}) \quad (2)$$

- $\dot{m}_{cw}$ = Cooling water mass flow rate (kg/s)
- $(C_p)_{cw}$ = The specific heat capacity of liquid water (4.18 kJ/kg·°C)
- ΔT_{cw} = Temperature rise of cooling water (°C)
- $\dot{m}_s$ = Steam mass flow rate (kg/s)
- h_s = Enthalpy of inlet steam (kJ/kg)
- h_{cond} = Enthalpy of condensate (kJ/kg)

The heat transfer rate is also expressed through the Fourier (LMTD) equation (Laskowski et al., 2020; Wróblewski et al., 2013):

$$\dot{Q} = U A \Delta T_{ln} \quad (3)$$

- U = Heat Transfer Coefficient (W/m²K)
- A = Heat transfer surface area (m²)
- ΔT_{ln} = Log Mean Temperature Difference (°C)

The logarithmic mean temperature difference (LMTD) is obtained from Eq. (4):

$$\Delta T_{ln} = \frac{(T_s - T_{cwin}) - (T_s - T_{cwout})}{\ln \left(\frac{T_s - T_{cwin}}{T_s - T_{cwout}} \right)} \quad (4)$$

- T_s = Saturation temperature of steam (°C)
- T_{cwout}/T_{cwin} = outlet/inlet CW temperatures (°C)

where T_s is the steam saturation temperature corresponding to the measured condenser pressure. U represents the overall heat transfer coefficient (Wróblewski *et al.*, 2013), the standard formulation of the Heat Exchanger Institute (HEI):

$$U = U_1 \times F^W \times F^M \times F^C \quad (5)$$

Where U_1 is the uncorrected HEI base coefficient (function of CW velocity and tube outer diameter), F^W is the CW inlet temperature correction factor, F^M is the tube material and gauge correction factor (constant for a given tube type), and F^C is the cleanliness factor. At design conditions, $F^C = 0.85$. The condensed pressure $T_{cw out}$ and Q are iteratively determined by solving the equations (2)–(5) in each load case scenario (Vodeniktov et al., 2023; Laskowski et al., 2020).

Simulation Model and Verification

For the heat transport correlations, the steady-state simulation model was constructed with professional CFD-compatible engineering software based on the HEI Standards (Qureshi and Zubair, 2016). The model inputs for each load level are the measured exhaust steam mass flow rate and inlet enthalpy, cooling water (CW) inlet temperature, cooling water (CW) volumetric flow rate, and fixed condenser geometry (surface area, tube configuration). The model outputs condenser pressure, cooling water (CW) outlet temperature, heat duty Q , and U .

The simulation results were first compared to the plant heat-balance design data for 100% load (Table 3). The validated model was then run at 75% and 50% load and compared to actual in-situ measurements of DCS (Tables 4 and 5). Agreement is expressed in relative deviation terms:

$$Error\ (\%) = \frac{Measured - Simulated}{Measured} \times 100\%$$

(6)

3. RESULTS

100% Load (1070 MW) Model Validation

Table 3 compares the simulation prediction with the plant heat-balance design specification at full load (1070 MW, 143,910 m³/h CW flow, $T_{CW\ In} = 30^{\circ}C$). All main parameters agree within 1 %, confirming the model fidelity at the design operating point. Figure 2 shows the simulated system diagram at heat balance 100% load. Table 4 shows further validation and compares the validated simulation against in-situ DCS data at 100% load.

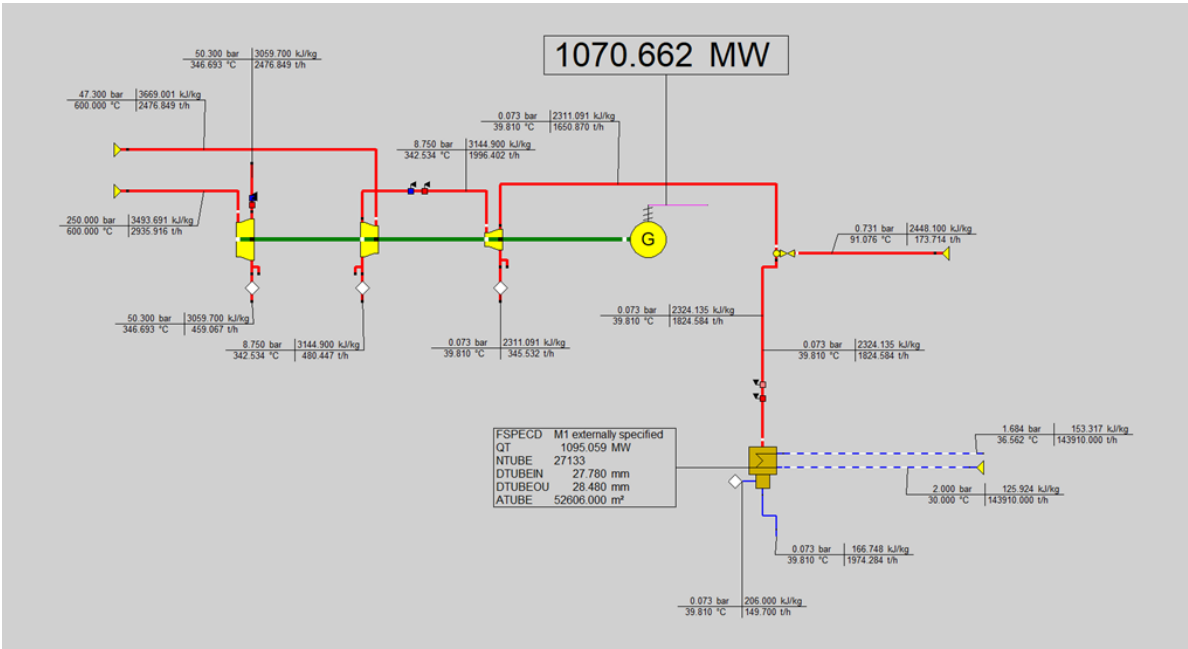


Figure 2. Simulated system diagram at heat balance 100% load

Table 3. Validation of simulation against heat-balance design data — 100% load (1070 MW)

Case	T_CW, in (°C)	T_CW, out (°C)	CW Flow (m³/h)	Load (MW)
Simulation	30.00	36.56	143.910	1070.66
Heat Balance Design	30.00	37.00	143.910	1070.00
Deviation (%)	0.00%	1.19%	0.00%	0.06%

Table 4. Simulation vs. In-Situ DCS data — 100% load (1070 MW)

Case	T_CW, in (°C)	T_CW, out (°C)	CW Flow (m³/h)	Load (MW)
Simulation	30.00	36.56	143.910	1070.66
In-Situ (DCS)	29.20	36.40	143.910	1068.00
Deviation (%)	2.74%	0.44%	0.00%	0.25%

75% load (803 MW) In-Situ Performance

One CW pump is operating at 75% load with a reduced flow of 93705 m3/h and an outlet waterbox valve opening of 38%. The simulated and in-situ comparison is displayed in Table 5. Given the abundant supply of cooling water (CW) for the lowered heat duty, the condenser pressure should be low, and the drop in steam flow and condenser heat duty are proportionate to full load. Figure 3 shows the simulated system diagram at 75% load.

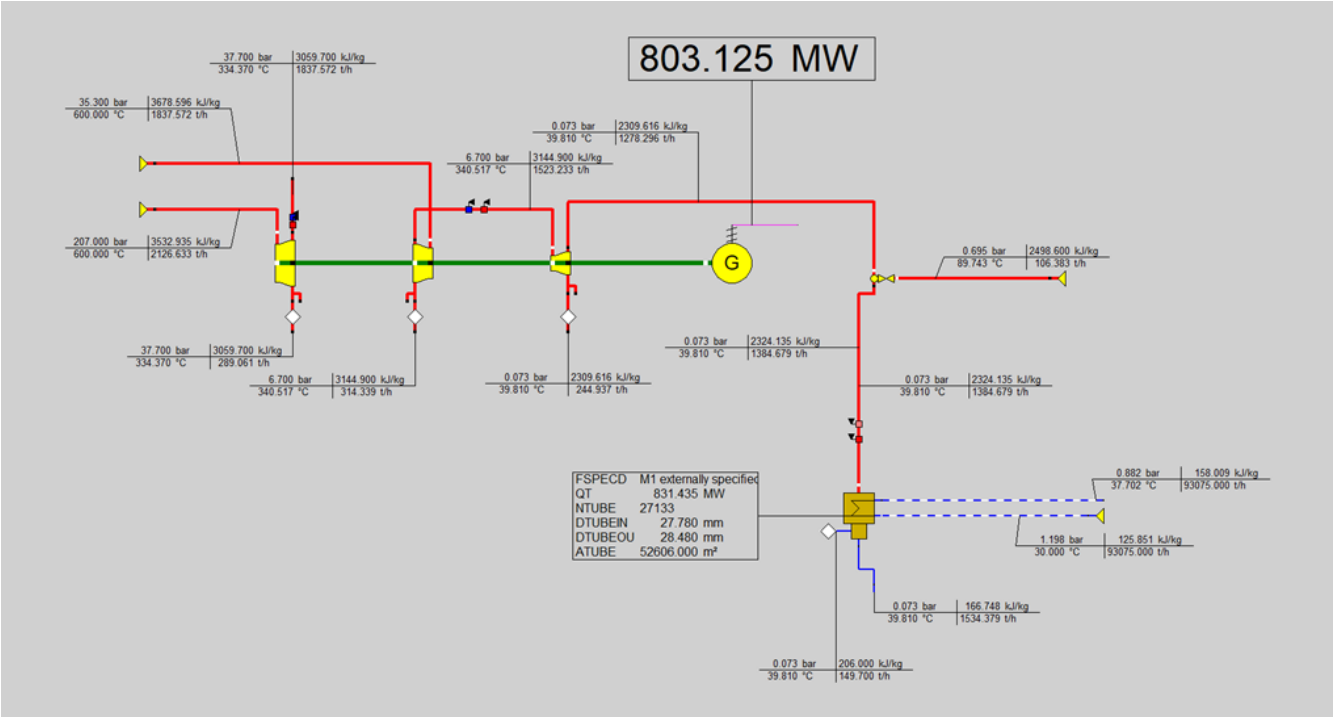


Figure 3. Simulated system diagram at 75% load (803 MW)

Table 5. Simulation vs. In-Situ DCS data — 75% load (803 MW)

Case	T_CW, in (°C)	T_CW, out (°C)	CW Pumps	CW Flow (m³/h)	Load (MW)
Simulation	30.00	37.70	1	93,705	803.13
In-Situ (DCS)	27.50	36.70	1	93,705	790.00
Deviation (%)	9.09%	2.72%	—	0.00%	1.63%

The cooling water (CW) outlet temperature deviation of 2.72% and load deviation of 1.63% are within acceptable modeling tolerance. The 9.09% deviation in cooling water (CW) inlet temperature (30.00°C simulated vs. 27.50°C measured) reflects actual seasonal sea-surface conditions at the time of data acquisition being cooler than the design assumption. The resulting lower cooling water (CW) inlet temperature reduces condenser pressure further below the simulated value (69.1 vs. 73.0 mbar), confirming that the plant operates with significant excess cooling margin at this load level - a key finding with implications for pump power optimization.

With only one CW pump running and 38% waterbox valve opening, the cooling system supplies a cooling water (CW) flow approximately 35% lower than at full load.

In-Situ Performance at 50% Load (535 MW)

At 50% load, the steam flow to the condenser is approximately half the design value. CW pump configuration is identical to the 75% case (one pump, 93,705 m³/h, 38% valve). Table 6 and figure 4 shows simulation vs. in-situ comparison.

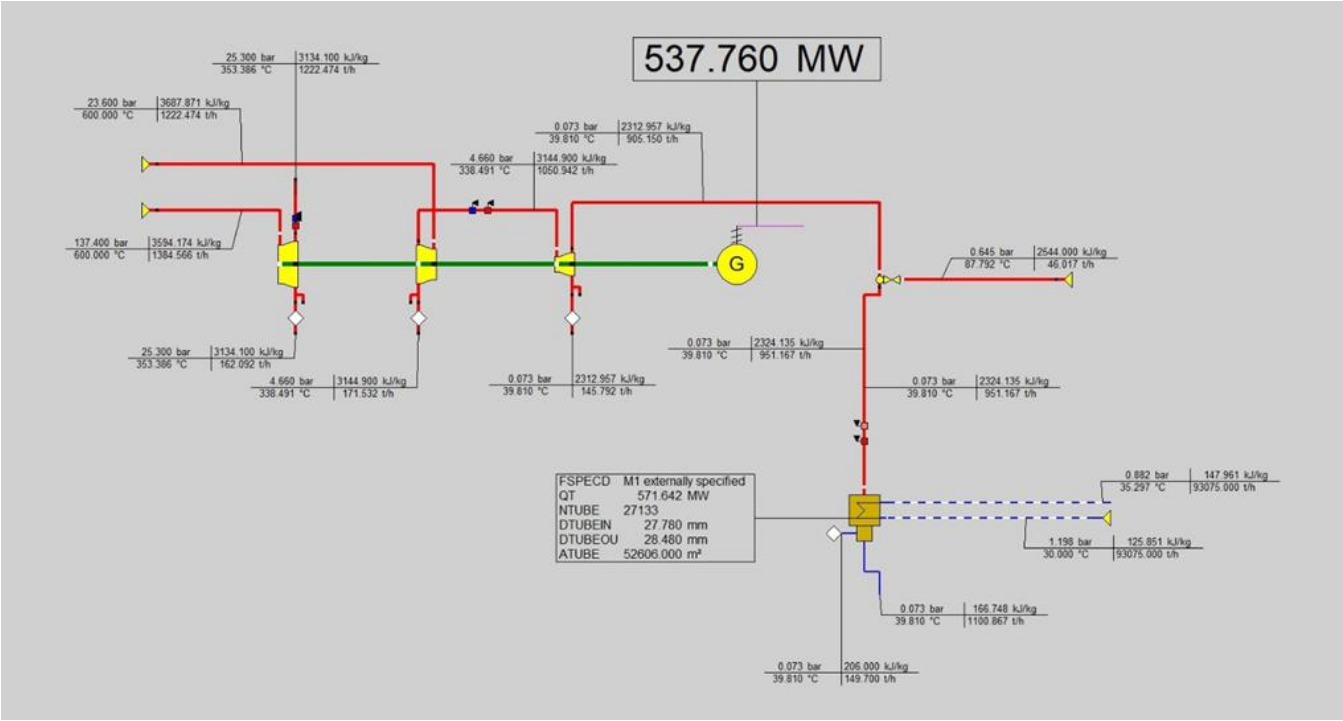


Figure 4. System diagram with simulation for 50% loading (535MW)

Table 6. Simulation vs in-situ DCS data – 50% load (535 MW)

Case	T_CW, in (°C)	T_CW, out (°C)	CW Pumps	CW Flow (m³/h)	Load (MW)
Simulation	30.00	35.30	1	93,705	537.76
In-Situ (DCS)	28.50	35.20	1	93,705	538.00
Deviation (%)	5.26%	0.28%	—	0.00%	0.04%

At 50% load, there is an excellent agreement between the simulation and DCS data for cooling water (CW) outlet temperature (0.28%) and load (0.04%), confirming the reliability of the model over the entire load range. A very high 35.2°C is observed for the cooling water (CW) outlet temperature, against a cooling water (CW) inlet temperature of 28.5°C, giving a ΔT_{CW} of only 6.7°C, well below the 11.3°C temperature rise at full load. This small increase in temperature is a clear indication of a CW oversupply relative to the condenser heat rejection requirement.

The simulation assumes the design CW inlet temperature of 30°C, and the actual DCS value is 28.5°C. Actual backpressure is far below the design point due to the 1.5°C lower-than-design condition and the excess cooling water (CW) flow vs. heat duty. The backpressure of 60.6 mbar is within the turbine operating envelope and is a favorable efficiency condition.

However, the condenser pressure at 50% load is well below the feasible lower bound, even with one pump operating, indicating that further cooling water (CW) flow reduction (e.g. by reducing the pump speed or increasing the valve throttling) may be achievable without degrading turbine exhaust conditions. As Laskowski (2016) pointed out, the optimum condenser operation at part load is a compromise between the turbine efficiency gain due to the lower backpressure and the auxiliary power cost of the CW pumps. The current data confirm that the current strategy is sub-optimal from an auxiliary power perspective.

4. DISCUSSIONS

Analysis Based on variative Unit Load

Table 7 presents the results of key in-situ measurements at three load variation, allowing for a direct comparison between cooling water (CW) conditions and condenser behavior.

Table 7. Combined in-situ DCS measurements at all load levels

Parameter	100% Load (1070 MW)	75% Load (803 MW)	50% Load (535 MW)	Unit	Trend
CW Inlet Temp (T _{CW, in})	29.2	27.5	28.5	°C	↓ at PL
CW Outlet Temp (T _{CW, out})	36.4	36.7	35.2	°C	↓ at PL
ΔT CW (Outlet – Inlet)	7.2	9.2	6.7	°C	↓ at PL
CW Flow Rate	143,910	93,705	93,705	m ³ /h	↓ at PL
No. CW Pumps Running	2	1	1	—	↓ at PL
Condenser Pressure (P _{cond})	71.5	69.1	60.6	mbar	↓ at PL
Saturation Temp (T _s)	38.9	38.4	36.6	°C	↓ at PL
Generator Output	1068	790	538	MW	↓ at PL

PL = part load.

The data exhibit a consistent pattern: reduction in plant load, reduction in condenser pressure, and increase in CW temperature even with a reduction in CW flow (one pump off at 75% load). This confirms that the cooling water (CW) system has excess cooling capacity at low loads. At 75% load, the CW temperature rise ΔT_{cw} : 9.2°C is in fact higher than at 100% load ΔT_{cw} : 7.2°C because the reduced CW flow with a single pump is partially compensated by reduced heat duty. At 50% load, the lowered heat duty dominates, with ΔT_{cw} to 6.7°C, clearly indicating an over-supply of CW.

The progressive decrease of condenser pressure from 71.5 mbar (100%) to 60.6 mbar (50%) shows that under the present CW operation strategy, the turbine exhaust conditions are better at lower load conditions. While this improves turbine work output, the low ΔT_{cw} at 50% load is due to pumping and can be avoided. Qureshi and Zubair (2016) state that the degradation of system performance is logarithmically related to the reduction in condenser UA (fouling or reduction in flow). Similarly, excess UA can be kept by over-supplying CW; however, the benefit in terms of backpressure diminishes after a certain point.

Implications for optimization of CW flows

The in-situ data collectively indicate that the current CW operating strategy at part load is conservative. The plant is operated at a fixed CW flow configuration (one pump, fixed valve position) at both 75% and 50% load without any change to accommodate the significantly different heat rejection requirements. This results in unnecessary consumption of auxiliary power of the CW pump. The CW pumps at this plant are typically 4-8 MW each; running a pump at 50% higher flow than required results in a wasteful auxiliary load of 1-3 MW, lowering net plant output by 0.2-0.6% (Anozie and Odejebi, 2011).

A load-adaptive CW flow approach (e.g., variable-speed pump drives or staged valve control) can reduce the CW flow at 50% load to approximately 70% of the existing single-pump flow without increasing the condenser pressure beyond 75 mbar, thereby ensuring safe operation. This is consistent with the findings of Laskowski (2016). Hu et al., (2024) show that optimizing the CW side can significantly increase the net output of the combined cycle.

Long-term fouling management is also necessary to maintain the present favorable backpressure values. According to Wang et al., (2012) the cleanliness coefficient drops in a non-linear asymptotic way with runtime, with the maintenance threshold ranging from 45 to 90 days depending on water quality. Cregan and Flynn, (2006) showed that coupling a composite fouling model with genetic-algorithm optimization can be used to identify the least-cost cleaning schedule. A cleaning interval of 60–90 days is recommended for

the study plant based on a design cleanliness factor of 0.85, consistent with the practices reported by Howell and Saxon, (2017) and Ma et al., (2018).

5. CONCLUSION

The in-situ effectiveness of cooling water flow at three operational load levels (100%, 75% and 50%) in a 1070 MW ultra-supercritical coal-fired steam power plant was investigated using actual DCS data validated against a steady-state thermal simulation model. Our main results are as follows:

1. The plant's HEI-based thermal model was validated, with simulation predictions within 1.2% of design data for all major variables at 100% load. The part load variances are mostly due to real CW inlet temperatures were 1.5-2.5°C lower than the design value estimate of 30°C, indicating and suggesting favorable ambient sea conditions at the time of data collection periods.
2. Confirmation of conservative CW operation at part load: confirmation that the plant is operating with a significant excess cooling margin. The 6.7°C of ΔT_{CW} at 50% load is well below the design range of 9-11°C, which directly indicates over-supply of CW flow in relation to heat duty.
3. An auxiliary power saving opportunity was identified due to the use same CW pump configuration at 75% and 50% loads (one pump, 93,705 m³/h, 38% water box valve) does not take advantage of potential further flow reduction at the lower load level. A load-adaptive approach, i.e., reducing CW flow at 50% load by 20-30% using variable-speed drives or additional valve throttling, could save 1-3 MW of pump auxiliary power without compromising condenser vacuum, improving net plant efficiency by 0.1-0.3 percentage points.
4. Fouling management remains a critical key: The present favorable backpressure conditions rely on maintaining the design cleanliness factor of 0.85. Based on published fouling growth models for once-through coastal cooling systems, the findings suggest a tube-cleaning interval of 60-90 days to avoid the observed CW temperature deviation and backpressure advantage from being eroded by progressive scale and biofouling accumulation from observed CW temperature value performance.
5. The validated simulation model is a useful tool for real-time performance evaluation and pilot testing of alternative CW operating strategies. Integrating this model into the DCS automation system can enable load-responsive CW flow management, thereby continuously optimizing the balance between condenser vacuum and power for auxiliary pump power consumption.

Recommendation for Future Research

Future investigations should also address the seasonal variability of CW source temperature, the prolonged effects of fouling growth as determined by a time-variable cleanliness coefficient model, and an economic evaluation of variable-speed pump investments against the potential operational power plant benefits, energy auxiliary power savings, and improved condenser performance.

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Author Contributions

Didi Supandi designed and conducted conceptualization, methodology, software. Tony Suryo and Eflita Yohana provided supervision for the data curation, writing- original draft preparation, visualization, investigation. Mohammad Bagir and Syarif Priyanto supervised software, validation, writing- reviewing and editing.

Ethical issues

Not applicable. This study does not involve any experiments on humans or animals. Hence, ethical approval was not required.

Informed consent

Not applicable.

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Conflict of Interest

The authors declare that they have no conflicts of interest, competing financial interests or personal relationships that could have influenced the work reported in this paper

Data and materials availability

Data that support the findings of this study are embedded within the manuscript.

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