Optimum circulating water flow rate of dry-cooling system at different working conditions of power generating unit

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Abstract. The optimum water flow rate of dry-cooling system is studied under various power loads of power plant. Taking the typical thermal power plant as example, the numerical modeling of the dry-cooling system and the mathematical process of the entire water team circulation are developed and coupled with each other. The total coal consumption as the summation of the unit coal consumption and the circulating water pump coal consumption is focused, because the water flow variation has effects on the two sections. By iterative solution, the results show that the total coal consumption increases slightly at first but then declines as the water flow increases under various power loads. Therefore, under each power load working condition, there exits the optimum flow rate of circulating water for the cooling system. Additionally, with operation at the optimum water flow rate of the dry cooling system, more energy-saving benefits can be achieved for the power plant operating at the higher power load.

Keywords: dry-cooling system, thermal power plant, varied power load, circulating water flow rate, total coal consumption.

1. Introduction
Many thermal power generating units adopt the dry-cooling technology owing to its environmental-friendly advantages such as the water-saving, lack of mechanical noise and recirculation of hot fluid plume [1]-[2]. Generally, dry-cooling system coupled with the condenser of the coal-fired power plant will reject more than half of the thermal energy from the boiler, and the energy efficiency relates closely to the condensate temperature [3]. Therefore, it’s worth exploring the energy-saving operation for the dry cooling system (NDDCS) that the power plant can better satisfy its economic and environmental requirements.

When the ambient temperature or crosswind increases, the thermal and flow behavior of NDDCS becomes weakened. Researchers have brought forward many pragmatic methods to improve its cooling performance. Xia et al. [4] conducted the numerical process of pre-cooling performance of water spray, pointing out that the air temperature as well as relative humidity impacts significantly on the maximum fully evaporated mass flow which can get improved by enhancing the turbulence intensity. Chen et al. [5] evaluated the large-dimension dry cooling system with precooling of water spray, concluding that the variable humidity leads to achievable range of performance. Additionally,
heat transfer rate of the rear sector rises when the air inlet temperature decreases; the turbine back pressure also reduces if the water spray flow rate increases [6]. Sun et al. [7] studied the arrangement pattern for the nozzle of the pre-cooling system to achieve maximum heat transfer efficiency with the minimum water usage, when using 5 nozzles, hot air can be cooled by $6.3^\circ\text{C}$ with cooling efficiency improved by 51.2%. Subsequently, the height of injection, radial interval and injecting direction of the typical nozzle LNN1.5 have been further investigated [8]. Additionally, the pre-cooling effects on the tower with height of 20 m were demonstrated for the first time in the world, and the tower performance was lifted by 6.68% with the optimized spray system [9]. By comparing the performance of the complete dry-cooling system, dry-cooling system with pre-cooling and wet cooling system, the pre-cooling type can enhance cooling efficiency by 46% during the adverse summer condition with 70% reduced water consumption [10]. What’s more, the hybrid dry-cooling and wet cooling system was recommended to combine their specific water saving and better cooling advantages [11]-[13].

In addition to the aforementioned pre-cooling and hybrid cooling techniques, air flow guiding such as the air deflectors and windbreakers are generally regarded as effective measures to recover the deteriorated cooling performance by ambient wind. The air deflector can reduce the deviation angle of air flow for nearly all cooling deltas, which improves the cooling performance of the former cooling columns [14]. Thermo-flow performances of the lateral sectors can be intensified by windbreakers, and the exterior installation shows superior to the interior installation [15]. Gu[16] discovered that the windbreaker measure has higher cooling efficiency than other configurations such as the cross-wall, crossline screen or louver. The optimal setting angle of windbreakers equals the air flow deviation angle or zero in the absence or presence of airflow separation outside the tower, which shows quite effective to strengthen the cooling performance of NDDCS [17]. When dividing the outside windbreaker into three rotatable columns, wall-form of the lateral column improves the performance over 17% at wind speed of 8m/s [18]. The swirling plume of air induced by nozzle has been proposed recently, which greatly enhances the cooling capacity of NDDCS with creating the additional draft force [19]. Besides, hot air extraction was put forward to enhance the aerodynamic and heat performance of dry cooling system, besides the maximum back pressure reduction could reach as high as 9.28% [20].

These aforementioned researches provided solid theoretical supports and directions to strengthen the flow and thermal behavior of cooling system indeed, however they mostly paid attention to the single section of the dry cooling system or the cold end system coupling with condenser. As a significant cooling part, the performance of NDDCS actually relates closely to the varied power loads of the entire power plant. In such a case, it is essential to extend the corresponding investigation for the entire coupled dry-cooling system and water-steam system.

As also pointed out, previous researches concerned mainly on intensifying the air-side cooling capability by introducing the pre-cooling and hybrid cooling technologies, or improving the air-side flow field through windbreaker, deflector, and air recovery measures. Fairly few works have carried out the performance enhancement of NDDCS from the water-side except for researches [21]-[29]. With hot water re-distribution in the air-cooled heat exchanger under crosswind conditions, the cold fluid/cooling air and the hot fluid/circulating water match better which increases the cooling performance by 18% under 4 m/s [21]. With proper water flow distribution, cooling performance gets improved at any wind speed, but excessive redistribution between the lateral sector and frontal sector has adverse influences under small crosswinds [22]. When combining the air side optimization by windbreak wall, water redistribution makes good use of the non-uniform airflow that the flow and thermal behavior of the Gatton NDDCT gets further lifted [23]. Dramatic economic and environmental benefits can be realized if reconstructing the wet-cooling system into the dry-cooling system, besides further economic outcomes can be achieved with applying the water flow redistribution [24]-[25]. What’s more, during operation in freezing condition, the anti-freezing capability of the windward air-cooled sector gets strengthened with water flow redistribution [26]-[27], meanwhile the energy-efficient operation of the cold-end system could reach the chocking turbine back pressure with simultaneous adjustment of louver and circulating water [28]-[29].
It can be seen that, these works focused on redistributing the water flow rate among various air-cooled sectors at a constant total value, and the research models were aimed at the single dry-cooling system or the cold end system coupling the condenser, however the water flow influences on energy-saving operation of the whole power plant haven’t been disclosed yet. As a matter of fact, the variation of circulating water flow rate of the dry cooling system influences both the power consumption of water pump and turbine back pressure. In such a case, this research will explore the optimum water flow rate of NDDCS under various output power loads of the power plant.

2. Description of coupling model

2.1. Thermodynamic cycle of power plant with NDDCS
Dry-cooling system in a typical 660 MW thermal power plant is adopted in this research. Generally, its water-steam cycle incorporates the boiler, the turbine, the re-heaters, the coupled condenser and dry-cooling system, and the circulating water pumps, as shown in Figure 1. Table I has listed each section of the water-steam system as well as the pipeline loop in detail.

![Figure 1. Thermodynamic cycle of the typical 660 MW power plant with NDDCS](image)

**Table 1. Key Sections of the Thermodynamic Cycle**

| Item number | Equipment                                    |
|-------------|----------------------------------------------|
| No. 1       | Boiler                                       |
| No. 2       | High pressure cylinder                       |
| No. 3       | Intermediate pressure cylinder               |
| No. 4       | Low pressure cylinder                        |
| No. 5       | Condenser                                    |
| No. 6       | Condensate pump                              |
| No. 7-9     | Low pressure heater                          |
| No.10-12    | High pressure heater                         |
| No.13       | Deaerator                                    |
| No.14       | Feed water pump                              |
| No.15       | Circulating water pump                       |
| No.16       | Natural draft dry cooling system             |
| No.17       | Main steam                                   |
| No.18       | Reheat steam                                 |
| No.19       | Circulating water                            |

Process of the thermodynamic cycle: (1) Exhausted steam from low pressure cylinder condenses in condenser with releasing its latent heat; (2) then the condensate water flows into the regenerative system and gets pre-heated sequentially; (3) afterwards, it enters the boiler and converts into dry steam with high temperature and pressure; (4) the dry steam will be heated further in the boiler and then flow into the high pressure cylinder to drive the shaft of turbine; (5) whereafter, it returns to the boiler to be reheated before expanding in both the intermediate and low pressure cylinders; (6) finally, after the conversion of internal energy to mechanical energy, this process goes back to the first step.
2.2. Physical model of NDDCS

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Circulating water acts as the thermal connection of the water steam system and NDDCS. Figure 2 displays the prototype model of NDDCS with hyperbolic tower shell, vertical air-cooled heat exchanger, and X-form supports. Air-cooled heat exchanger consists of 176 cooling deltas with the height of 22 m, and each of them includes two finned tube columns with the angle of 49°. Table II presents the specific geometric parameters of this dry-cooling system.

![Figure 2. Geometric configuration of NDDCS](image)

| Parameter                                         | Value  |
|---------------------------------------------------|--------|
| Tower height (m)                                  | 150    |
| Height of tower throat (m)                        | 113.5  |
| Height of air-cooled heat exchanger (m)           | 22     |
| Tower outlet diameter (m)                         | 87     |
| Tower throat diameter (m)                         | 84     |
| Tower bottom diameter (m)                         | 142    |
| Cooling delta number                              | 176    |

3. Numerical process of NDDCS

3.1. Governing equations and thermo-flow model

Ambient air is considered to be the impressible ideal gas during the numerical simulation of NDDCS with steady state. The continuity, momentum, and energy equations are summarized together with the turbulent model to describe the flow and thermal performance of NDDCS. The general form is expressed as follows [14]-[16], [30]-[35]:

$$\frac{\partial \rho u \varphi}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \Gamma_{\varphi} \frac{\partial \varphi}{\partial x_j} \right) + S_{\varphi} (j = 1, 2, 3) \tag{1}$$
When modeling the thermal and flow process between water and cooling air, the pressure difference \( \Delta p \) and heat transfer rate \( Q_{\text{total}} \) take the following forms by adopting the heat exchanger model [26]-[29], [32]-[34].

\[
\Delta p = \frac{1}{2} f \rho u_{\text{Amin}}^2 \quad (2)
\]

\[
Q_{\text{macro}} = \varepsilon_{\text{macro}} m_w c_{\text{pa}} (t_{w1} - t_{a1})_{\text{macro}} \quad (3)
\]

\[
Q_{\text{total}} = \sum Q_{\text{macro}} \quad (4)
\]

where \( f \) means the coefficient of flow loss, \( \rho \) represents the mean density of cooling air, \( u_{\text{Amin}} \) equals to the velocity at the minimum area of air flow, \( \varepsilon_{\text{macro}} \) means the heat exchanger effectiveness, \( m_w c_{\text{pa}} \) is the water heat capacity rate, \( Q_{\text{macro}} \) represents the macro heat rejection rate.

3.2. Numerical domain, boundaries, and solution method

Figure 3 presents the 2000 m × 1000 m × 1000 m computational domain that is far larger than NDDCS to avoid the boundary influences. Referring to researches [22], [26], [34]-[35], the hexahedral meshes are adopted for heat exchanger and cooling tower, while the tetrahedral/hexahedral combined meshes are applied for the central block with NDDCS. Mesh cases as 2,726,157, 3,624,582 and 4,068,592 are created to check the grid independence under the designing crosswind of 4 m/s. Heat rejection of NDDCS varies as small as 0.17\% at the two denser cases that the grid number as 3,624,582 is finally chosen.

With crosswind, the upwind domain surface is appointed as velocity inlet, meanwhile the crosswind \( u_z \) is calculated based on the power-law equation [13]-[15], [26]-[29], [36]. The side and top surfaces, as well as the symmetry plane are given the symmetry condition; the rear surface is given the outflow condition. The ground, support between delta, as well as tower shell are assigned to the thermal isolation wall.

\[
u_z = u_{w1} \left( \frac{H}{H_{\text{ref}}} \right)^c \quad (5)
\]

During the numerical solving process, the upwind differencing method with second-order is used to discretize the summarized mathematical equations. Solving algorithm of SIMPLE is chosen to conduct the iterative coupling of air pressure and velocity with residuals set smaller than 10\(^{-4}\) [32]-[33].

Figure 3. Numerical domain of NDDCS with the corresponding boundaries
3.3. Mathematical process of off-design operation of power plant

For the typical dry-cooling power plant, the mathematical process of the varied working condition involves the turbine, the condenser, and the heat re-generative system. Under the varied working condition, the main steam and extraction steam flow rates changes, so the iteration is adopted to obtain their values. The initial flow rate of main steam $D_{ms0}$ is calculated from:

$$\frac{D_{ms}}{D_{ms0}} = \frac{N}{N_0} \tag{6}$$

where $N$ means the power load, the symbol 0 represents the designing condition.

The Flugel formula is adopted to obtain the extraction pressure [38]:

$$\frac{D}{D_0} = \left(\frac{P_{1}^{2} - P_{2}^{2}}{P_{10}^{2} - P_{20}^{2}}\right) \frac{T_{10}}{T_{1}} \tag{7}$$

in which $D$, $P$, and $T$ mean the mass flow rate, pressure, and temperature of the steam from a certain stage group; the symbols 1, 2 mean the before stage group condition and after stage group condition.

In general, the temperature variation under off-design operation could be neglected, besides $P_2/P_1$ is fairly small among the stages, so Eq. (7) gets simplified as follows:

$$\frac{D}{D_0} = \frac{P_1}{P_{10}} \tag{8}$$

Based on the above equation, the extraction steam flow rate is regraded proportional to the main steam flow rate, therefore the extraction pressure $P_r$ is given as:

$$\frac{P_r}{P_{10}} = \frac{D_{ms}}{D_{ms0}} \tag{9}$$

In effect, the back pressure both relates to the off-design condition of the water-steam system and the performance of NDDCS, as more details illustrated in section D. Therefore, when calculate the varied working condition of the power plant, it is assumed constant. For each stage, the relative internal efficiency could be acquired following the steam turbine performance curve that the corresponding enthalpy can be obtained:

$$h_{2i} = h_{1i} - H \eta_i \tag{10}$$

in which $h_1$, $h_2$ mean the inlet and outlet steam enthalpy; $\eta$ and $H$ represent the relative internal efficiency and the isentropic enthalpy drop of the stage group; the subscript i means the stage group number.

Extraction steam mass flow could be acquired following the thermal conservation equation, which gives as:

$$D_{exj} = \frac{D_{ex,i} \tau_j - D_{d,j} \gamma_j}{q_j} \tag{11}$$
in which \( D_{es} \) means the extraction steam flow rate, \( D_{fw} \) means the feed water flow rate, and \( D_{d} \) means the drainage water flow rate; \( \tau \) means the enthalpy increase of feed water, \( \gamma \) means the heat transfer rate of drainage water, \( q \) means the heat transfer rate of steam; The symbol \( j \) means the extraction number.

In sequence, mass flow of exhausted steam \( D_c \) is obtained according to the mass conservation law:

\[
D_c = D_{ms} - \sum D_{ex,j} \tag{12}
\]

When the steam thermal parameters and the extraction steam mass flow rate are obtained, the unit output power load \( P_e \) can be calculated as:

\[
P_e = \frac{D_c \Delta h_c + \sum D_{ex,j} \Delta h_{ex,j}}{\eta_m \eta_g} \tag{13}
\]

where \( \Delta h_c \) means the enthalpy drop of exhaust steam, \( \Delta h_{es} \) means the enthalpy drop of extraction steam; \( \eta_m \) represents the mechanical efficiency, \( \eta_g \) represents the generator effectiveness with the constant value in this work.

During iteration, when the relative difference of \( P_e \) and \( N \) reaches smaller than 0.01%, the involved parameters in Eqs. (6-13) can be accepted, otherwise restart the procedure.

The turbine heat consumption rate \( q \) and the hourly unit standard coal consumption \( B \) can be obtained as follows:

\[
q = 1000 \times \frac{D_{ms} (h_{ms,2} - h_{ms,1}) + D_{reh} (h_{reh,2} - h_{reh,1})}{N} \tag{14}
\]

\[
B = \frac{D_{ms} (h_{ms,2} - h_{ms,1}) + D_{reh} (h_{reh,2} - h_{reh,1})}{\eta_b \eta_p Q_{coal}} \tag{15}
\]

where \( D_{reh} \) means the reheat steam mass flow rate, \( Q_{coal} \) means the standard coal heat, \( \eta_b \) and \( \eta_p \) are the boiler effectiveness and the pipeline effectiveness.

For maintaining the closed flow cycle of circulating water connecting the NDDCS and condenser, the required power consumption of water pump \( P_w \) can be calculated as [38]:

\[
P_w = \pi d^2 \Delta p_w G_w Ln_b n_b / 4 \rho_w \tag{16}
\]

where \( \Delta p_w \) and \( G_w \) are expressed as:

\[
\Delta p_w = \frac{fG_w^2}{2 \rho_w d} \tag{17}
\]

\[
G_w = \frac{4 M_{w,cl} n_{wp}}{\pi d^2 n_{th} n_{ba}} \tag{18}
\]

\( P_w \) can be converted into the standard coal consumption of water pump \( B' \) which gives as follows:
Consequently, the total standard coal consumption $B_{\text{total}}$ relating to the power load and circulating water of NDDCS can be obtained:

$$B_{\text{total}} = B + B'$$

### 3.4. Iterative process of coupling NDDCS and water-steam cycle

Input the above parameters $h_c$ and $D_c$, as well as the corresponding enthalpy of condensing water $h_{cw}$, and then calculate the heat rejection of condenser $Q$ as follows:

$$Q = D_c (h_c - h_{cw})$$

Based on the heat transfer principle between exhausted steam and circulating water, the water inlet temperature $t_{c1}$ and the water outlet temperature $t_{c2}$ are obtained:

$$Q = K_c A_c \frac{t_{c2} - t_{c1}}{\ln \frac{t_s - t_{c2}}{t_s - t_{c1}}}$$

$$Q = M_w c_p_w (t_{c2} - t_{c1})$$

in which $K_c$ means heat transfer coefficient of condenser [39], $A_c$ means heat transfer surface area; $M_w$ means water flow rate, $c_p_w$ means specific heat capacity; $t_s$ means the steam saturated temperature.

Circulating water shows as the connecting medium for the condenser and NDDCS, hence the $t_{c2}$ of condenser is actually the $t_{w1}$ of NDDCS. Then by numerical simulation, the heat transfer of NDDCS $Q'$ could be acquired. Figure 4 displays the iterative heat transfer process of the cold end system, when the relative error of $Q$ and $Q'$ presents smaller than 0.05%, the assumed back pressure is accepted [22, 30]. In this research, circulating water flow rates from 70% to 150% of the design condition with the value of 70000 $\text{m}^3/\text{h}$ are considered under various power loads from 60% to 100%. The optimum water flow rate can be approached when the total standard coal consumption $B_{\text{total}}$ has the minimum value.
3.5. Model validation
To validate this mathematical modeling, the heat consumption rate $q$ of steam turbine is calculated and compared with the actual operating parameter at four power load working conditions, as presented in Figure 5. Their difference shows to be 128 kJ/kWh, -39 kJ/kWh, -187 kJ/kWh, -103 kJ/kWh at power load of 100%, 85%, 70%, 60%, respectively. The relative error between them varies from 0.49% to 1.62%, which is fairly small. Hence, the mathematical model is accurate enough for analyzing the off-design operation of the coupled system.

4. Results and discussions
The turbine back pressure as well as the power consumption of water pump at various flow rates versus different power loads are analyzed. As pointed out, since the variation tendency is quite similar for different power loads, the results at the condition of 100% power load are specifically analyzed as the typical example, which are presented in Figure 6. As clearly observed, the back pressure declines with the increase of water flow, besides the maximum gap can reach nearly 5.6 kPa between the 70%
and 150% water flow rates. While with the increased water flow, the power consumption of water pump rises, and the changing gradient also becomes larger.

![Figure 6. Comparison of turbine back pressure with power consumption of water pump at 100% power load](image)

![Figure 7. Total coal consumption at various circulating water flow rates for different power loads](image)

The total coal consumption versus various water flows and power loads is displayed in Figure 7. As observed, the result drops violently with the decrease of power load, because decreased fuel is required in boiler with the decreased turbine output work. With the increased water flow rate at a certain power load, the total coal consumption increases slightly at first but then decreases. It’s also noted that, the effect of the water flow rate is far smaller than the power load, meanwhile if the power load declines, the impact from the water flow rate will reduce further.

The relative indicator $\Delta B_{\text{total}}$ for the total standard coal consumption is introduced to reveal its vibration more clearly, as defined follows:

$$\Delta B = \frac{B_{\text{total}} - B_{\text{total, initial } M_p}}{B_{\text{total, initial } M_p}} \times 100\% \quad (24)$$
Figure 8. Relative indication total coal consumption at various circulating water flow rates under different power loads

Figure 8 gives the relative value versus various water flows and power loads. As can be seen, at each power load, $\Delta B$ decreases at first but then increases as the water flow increases. Therefore, there exists the optimal value of flow rate at a certain power load that $\Delta B$ has the minimum value. Additionally, at a certain water flow rate condition, as the power load decreases, the exhausted steam flow rate will decrease, so the decreasing trend of turbine back pressure gets weakened, which implies that the coal consumption declination will become mildly.

In addition, Figure 9 presents the accurate optimal water flow rate at each power load, meanwhile Figure 10 reveals the corresponding minimum coal consumption and the maximum value. As can be seen, the optimum water mass flow rate equals 148%, 125%, 118%, 109%, 103% of the design value at the power load of 100%, 90%, 80%, 70%, 60%, respectively. What’s more, the largest gap between the minimum and maximum total consumptions reaches 4.7 ton per hour at the 100% power load. Besides, the smallest gap still shows to be 0.65 ton per hour at the 60% power load.

Figure 9. Optimum circulating water flow rate of NDDCS under various power loads
Figure 10. The minimum and maximum total coal consumptions under the same power load

The results demonstrate that power plant can achieve the significantly energy-efficient operation with the optimal circulating water of NDDCS, meanwhile more energy-saving benefits can be realized under larger power load.

5. Conclusion
The numerical process for NDDCS and the mathematical process for the varied working condition of power plant are coupled. Through iterative solution, the optimum water flow rate of NDDCS is explored so as to achieve the energy-efficient operation for power plant under various off-design conditions. The main conclusions of this research are illustrated as follows.

(1) The turbine back pressure decreases while the power consumption of water pump increases with the increased water flow rate. The gap of back pressure can reach as high as 5.6 kPa between the maximum and minimum water flow rates. The variation gradient of power consumption of water pump becomes higher as water flow rate increases.

(2) When the power load declines, the total coal consumption drops significantly. If the water flow rate increases at a certain power load, the total coal consumption will increase firstly but then decrease. Impact of the water flow rate shows obviously smaller compared with the power load, which also reduces further with the decreased power load.

(3) At each power load working condition, there exists the optimal water flow rate of NDDCS that the coal consumption reaches smallest. What’s more, when the power load increases, more energy-saving benefits can be achieved for power plant with the optimal water flow of NDDCS.

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