Multi-objective optimization of the generator air cooler based on genetic algorithm

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Abstract—Generator unit No.1 in an electric power plant cannot run at full load owing to the high temperature of generator cooling air. Analysis has proved that the problem lay in the large cooling load and low air side heat transfer coefficient, and thus, the cooling requirement cannot meet in summer. Therefore, a generator air cooler with circular fin structure is optimized under summer operating conditions. The heat transfer coefficient, air side pressure drop, water side pressure drop resistance and overall weight are used as the optimization objectives using genetic algorithm, to optimize structure parameters of the generator air cooler. Through analysis, the best structure combination can be selected to manufacture a product. The optimized generator air cooler could meet the generator cooling requirements, operate at full load, and also reduce the investment and operation costs.

1. Introduction
The heat-supply steam turbine generator (model: QF-155-2) in a phase I plant basically runs at full load to maintain the plant power consumption. After summer arrives, the increased ambient temperature leads to an elevation in cooling water and air temperature. As a result, when the winding temperature in generator unit No.1 approaches 109°C, overheating may induce potential safety hazards, thus causing the unit to run at reduced load. The original generator air cooler is the type of the cooling tube with metal wire, The observations of operations as well as inspection and repair of the cooler reveal insignificant water-side scaling and pollutants and restricted cooling effects of the additionally installed pipe pump, thus scaling, fouling, and small flow are excluded. The heat transfer coefficient of air cooler calculated based on operating parameters is 5.479 W/(m\textsuperscript{2}·K), which is far lower than the designed coefficient of 87.266 W/(m\textsuperscript{2}·K). The field measurement demonstrates that the air- and water-side differential pressures of the cooler are 397 Pa and 95.35 kPa, respectively. Additionally, the high line loss and cooling load of the generator as well as the high external thermal resistance of air cooler cooling tube result in a low heat exchange capacity, the failure in meeting the cooling requirements of the generator under summer operating conditions, and the affected unit operations.

2. Study status of cooling of generator and air cooler
Thanks to increased unit capacity, novel technologies and approaches have been employed in studies on cooling of air-cooled generator. For example, the internal ventilation and cooling of generators are
currently being investigated through the numerical simulation and prediction and experimental verification overseas \cite{1-4}. While domestic studies primarily focus on theoretical analysis and numerical calculation and simulation in rotor, stator, and ventilation conditions of generators \cite{5-8}. Lu et al. \cite{9-11} conducted extensive studies in this field, including studies on simulation of complex flows featuring multiple flow transition characteristics in large air-cooled turbo-generators with stator multipath ventilation \cite{9}, impacts of changes in turbulence model on heat flow field of the steam turbine generator rotor \cite{10}, and the comparison of cooling effects of multiple ventilation plans of large air-cooled turbo-generator rotor \cite{11}. Additionally, some manufacturers and scientific research institutes conducted empirical studies \cite{12-14}. Currently, improvement of cooling intensity and ventilation loss should be extensively investigated in studies on cooling of generators, and the numerical parameters of generator air cooler as the major cooling component should be optimized through investigation. In studies on air cooler, Tang \cite{15} and Ji \cite{16} conducted numerical calculation and optimization of flow field and temperature field for the air cooler of medium-sized high-voltage electric motor, respectively. Zhang et al. established the computational model of regional heat transfer within generator air cooler, and provided the calculation method of heat transfer under different temperature conditions inside the internal air duct \cite{17}. In brief, engineering applications primarily focus on reformation of capacity expansion and cooler replacement, yet the design and optimization of the structure arrangement to improve the cooler’s performance have been rarely explored. In this study, the air cooler was designed, and the original air cooler was replaced through optimization of structure parameters. Additionally, the observations of two minor repair cycles proved that such design and optimization were feasible and significant cooling effects were demonstrated, thereby providing references for design and optimization of other units.

3. Air cooler modeling

3.1. Geometric modeling and quality calculation

The cooling water enters from the lower tube bundle of air cooler, and flows out from the upper tube bundle after heat exchange through the two tube passes. Figure 1 (a case study on the staggered arrangement of four rows of tubes with six tubes in each row) shows the schematic diagram for structure arrangement of air cooler.

$$M = \frac{1}{4} \pi n \left[ (d_r^2 - d_l^2) \rho_l + (d_r^2 - d_o^2) \rho_o + (d_f^2 - d_v^2) \rho_v \right]$$

(1)
3.2. Calculation of resistance and pump power consumption

The optimization of the air-side resistance and resistance at the cooling water side of air cooler is conducive to reducing the power consumption of driving fans and water pump. When air-side forced ventilation is conducted at the circular annular fin, the resistance loss can be calculated through the following formula [18]:

\[
\Delta P_a = f_a n_\text{max} \frac{G^2}{2 \rho_a} \quad (2)
\]

When the tube bundles are arranged in staggered rows, the air-side frictional resistance coefficient \( f_o \) can be calculated through the following formula [18]:

\[
f_o = 37.86 Re_o^{0.316} \left( \frac{P_i}{P_o} \right)^{0.515} \quad (3)
\]

Single-phase flow occurs in the tube, yet no phase changes occur. The resistance loss equals to the sum of frictional resistance loss along the tube length, resistance loss in the tube case, and resistance loss at the inlet and outlet [19], namely:

\[
\Delta P_l = \zeta (\Delta P_f + \Delta P_r) + \Delta P_N \quad (4)
\]

Where, \( \zeta \) denotes the fouling correction coefficient, which is related to fouling thermal resistance.

The pressure loss of fluid in the straight tube section can be calculated through the following formula:

\[
\Delta P_f = f_i \frac{G^2}{2 \rho_w} \frac{N_pr}{d_i} L \frac{\mu_w}{\mu_h}^{0.14} \quad (5)
\]

Calculation of frictional resistance coefficient in the tube [19]:

If \( Re_i < 10^3 \),

\[
f_i = 67.63 Re_i^{0.9873} \quad (6)
\]

If \( 10^3 = Re_i = 10^5 \),

\[
f_i = 0.4513 Re_i^{0.2653} \quad (7)
\]

If \( Re_i > 10^5 \),

\[
f_i = 0.2864 Re_i^{0.2258} \quad (8)
\]

The resistance loss in the tube case:

\[
\Delta P_f = 4 N_pr \frac{G^2}{2 \rho_w} \quad (9)
\]

The resistance loss at the inlet and outlet:

\[
\Delta P_N = 1.5 \frac{G^2}{\rho_w} \quad (10)
\]

After the air- and water-side resistance losses are obtained, the pump power for overcoming the intratubal and extra-tubal resistance consumption is estimated through the following formula:

\[
W_p = \frac{m_a \Delta P_o}{\eta_o \rho_a} + \frac{m_w \Delta P_f}{\eta_i \rho_w} \quad (11)
\]
Where, $\eta_o$ denotes the efficiency of air-side fan, and $\eta_i$ denotes the efficiency of water-side driving water pump. The fan blades are installed at both ends of the generator rotor to drive air circulation. In this study, the efficiency of the air-side fan and water pump were 0.6 and 0.75, respectively.

3.3. Calculation of heat transfer and air cooler

The total heat transfer coefficient of the heat exchanger equals to the reciprocal of the sum of all thermal resistances. The total heat transfer coefficient based on the external surface area of the light tube in the heat exchange tube can be calculated through the following formula [19]:

$$K = \frac{1}{\frac{1}{h_o} + \frac{A_i}{A_o} + R_o + R_i + R_t + R_j}$$  \hspace{1cm} (12)

Where, $h_o$ denotes the air-side heat transfer coefficient; $h_i$ means the water-side heat transfer coefficient; $A_o$ and $A_i$ indicate the external surface area of the light tube and the internal surface area of the heat exchange tube, respectively; and $R_o$, $R_i$, $R_t$, and $R_j$ represent air-side thermal resistance, water-side thermal resistance, tube-wall thermal resistance, and thermal contact resistance, respectively. When forced ventilation is performed at the circular annular finned tube bundle [18,20], the air-side heat transfer coefficient:

$$h_o = h_f \eta_f \frac{A_o}{A_b}$$  \hspace{1cm} (13)

Where, $\eta_f$ denotes fin efficiency, $A_o$ indicates the total surface area outside the tube, and $A_b$ refers to the external surface area of the light tube in the heat exchange tube. The tube bundles are arranged in staggered rows, and the heat transfer coefficient based on the total surface area outside the tubes can be calculated through the following formula [20]:

$$h_f = 0.1378 \frac{d_f}{d_f} R_e^{0.718} P_r^\frac{1}{2} \frac{P_r}{H_f} \frac{\delta_f}{H_f}^{0.296}$$  \hspace{1cm} (14)

The heat transfer coefficient in the tube can be calculated through the following correlation [19]:

If $Re_i \leq 2100$:

$$h_i = 1.86 \frac{\lambda}{d_i} \left[ Re_i Pr \frac{d_i}{L} \right]^{0.14} \frac{\mu_w}{\mu_b}$$  \hspace{1cm} (15)

If $2100 < Re_i \leq 10^4$:

$$h_i = 0.116 \frac{\lambda}{d_i} \left[ Re_i \left( \frac{d_i}{L} \right) - 1.25 \left( \frac{d_i}{L} \right)^2 \right] \frac{Pr}{Pr_f} \frac{\mu_w}{\mu_b}^{0.14}$$  \hspace{1cm} (16)

If $Re_i > 10^4$:

$$h_i = 0.027 \frac{\lambda}{d_i} Re^{0.8} Pr^{0.5} \frac{\mu_w}{\mu_b}^{0.14}$$  \hspace{1cm} (17)

3.4. Cost calculation

The air cooler cost primarily includes initial investment cost and operating cost. In this study, the calculation was simplified, and the time value of funds was not considered. In view of this, the air cooler cost can be calculated through the following formula:

$$C = C_{io} + C_{op}$$  \hspace{1cm} (18)

It is approximately considered that the initial investment cost of air cooler is directly proportional to the air cooler mass, namely:

$$C_{io} = c_{io} C_M M$$  \hspace{1cm} (19)

According to the optimization and design in this study, the initial investment quality cost of the
partner plant was investigated and comprehensively compared. The coefficient is: \( C_{\text{ini}} = 95 \) yuan/kg, and \( c_M \) denotes the ratio of the total mass of air cooler (including shell) to the mass of the core heat-exchanging components of air cooler. Based on the investigation, the ratio of the total mass of the rolled air cooler (including shell) \( c_M \) was approximately taken as 1.2 in this study. \( M \) indicates the total mass of the heat-exchanging components of air cooler.

The operating cost of air cooler primarily includes the cost of operating power consumption and inspection and repair cost. However, the inspection and repair costs were not considered in this study. Therefore, the operating cost equals to the sum of the air- and water-side driving costs, namely, the total cost of pump power consumption. The operating cost of air cooler can be calculated through the following formula:

\[
C_{\text{op}} = \frac{c_{\text{op}} W_p N_y t_y}{1000}
\]

Where, \( c_{\text{op}} \) denotes the electricity price, the on-grid electricity price \( c_{\text{op}} = 0.3525 \) yuan/kWh, \( W_p \) indicates the total pump power consumption, \( N_y \) represents the normal service life of air cooler, and \( t_y \) refers to the average annual operating hours of air cooler within the normal service life during three major inspection and repair periods. The average annual operating time of the generator unit in our plant was 7614.5 hours.

4. Design and optimization of air cooler

Based on the actual conditions and transformation requirements of our plant, the design and optimization of air cooler structure were conducted. Specifically, the multi-objective optimization was achieved through increasing the heat transfer coefficient and reducing the air- and water-side resistances, so as to control the cost. When the design and optimization are performed, the aforementioned requirements should be met as much as possible, and the optimal values should be provided to the decision makers for reference and selection.

4.1. Implementation of optimization method

Different structure parameters correspond to different optimized target values, and a series of combinations of structures are available, facilitating the acquisition of the optimal values through optimization and design of targets under the given constrained conditions. Therefore, the design and optimization aim at seeking the optimal combination of design parameters, so that the optimal values can be obtained for optimized target functions. Additionally, the process of design and optimization is in line with the operating process of genetic algorithm. Figure 3 shows the corresponding relationships between genetic algorithm and various optimized parameters of heat exchangers.

![Genetic algorithm and heat exchanger optimization relationship diagram](image)

Fig. 3 Relationship of parameters between genetic algorithm and optimization of heat exchangers
The chromogene in genetic algorithm (e.g., number of tube rows, number of tubes per row, tube diameter, and fin pitch) correspond to the design variables of the heat exchanger. The operation rules in genetic algorithm (e.g., selection, crossover, and mutation) are equivalent to the search method during the design and optimization of heat exchangers. The genetic algebra of genetic algorithm represents the iteration number of design and optimization of heat exchangers.

4.2. Design of calculation conditions
Design and optimization conditions include the known parameter conditions, design and optimization objectives and variables, and constrained conditions throughout the optimization process of air coolers. Parameter conditions include operating parameters, structure parameters, and physical property parameters. During the design and optimization, some structure parameters can be used as known conditions. Based on relevant regulations in JBT2728.3-2008 “Gas Cooler for Motor”, the air cooler parameters in practical application and manufacturing capacity of manufacturers, the parameters were selected (as shown in Table 1).

| Structure parameters | δ₁ | δ₂ | δ₃ | L  | W  | H  |
|----------------------|----|----|----|----|----|----|
| Value                | 1  | 0.4| 0.4| 3400| 560| 510|

Based on relevant regulations in JBT 2728.3-2008 “Gas Cooler for Motor” and actual requirements for water quality, BFe30-1-1 was selected as the tube material, and T2 was selected as the fin material. Physical property parameters include the relevant thermophysical properties of extra-tubal and intratubal media, tube materials, and fins. Specifically, the physical property parameters of air are obtained at the average inlet and outlet temperature of 55℃ and the operating pressure of 1 bar (910 kPa), while the physical property parameters of water are obtained at the average inlet and outlet temperature of 40℃ and the operating pressure of 0.2 MPa.

4.3. Optimization of variables and objectives and constrained conditions
With a view to comprehensively describing issues to be handled, multiple impacts should be weighed to achieve the overall optimal performance of the optimized targets, thus providing comprehensive reference information. As a result, a series of optimal solutions, corresponding to different values of optimization objectives, can be obtained for multi-objective optimization. In this study, multi-objective optimization of air cooler was performed. Specifically, the total heat transfer coefficient $K$, air-side resistance loss $\Delta P_o$, water-side resistance loss $\Delta P_i$, and the core component mass of air cooler $M$ were taken as the optimization objectives.

During the design and optimization, the values of optimization targets can be changed through changing the values of design and optimization variables (as shown in Table 2). According to the genetic algorithm, the initial population is randomly generated, namely, the initial values of design and optimization variables are randomly generated. However, due to the actual engineering process requirements and scope of application of the selected correlation, the values of various design variables in the air cooler, must be in line with the requirements in relevant air cooler specifications.

| Design parameters          | Symbol | Unit      | Value range | Value requirements   |
|----------------------------|--------|-----------|-------------|----------------------|
| Number of tube rows        | $N_t$  | Row       | 2-22        | Rounded number       |
| Number of tubes per row    | $n_t$  | Pcs       | 2-20        | Even number          |
| Fin pitch                  | $P_f$  | mm        | 2-6         | Accurate to 0.1 mm   |
| Tube external diameter     | $d_e$  | mm        | 15-38       | Rounded number       |
| Fin height                 | $H_f$  | mm        | 10-25       | Accurate to 0.1 mm   |
The design and optimization in this study should be subject to three constrained conditions, namely, constraint on design variable value, constraint on geometric structure, and physical constraint. Specifically, the constraint on geometric structure indicates the logical constraint on the heat exchanger structure. For the rolled air cooler, the constraints on geometric structure are as follows:

1. The tube spacing is greater than the external diameter of tube fins, i.e., \( P_t > d_f \).
2. The heat exchange tubes are arranged in a staggered isosceles triangle, \((P_t/2)^2 + P_l^2 \geq d_f^2\).

The physical constraint indicates the limitation of performance parameters of heat exchangers. In this study, based on the actual application environment and performance requirements of air coolers, the following physical constraints were established:

1. The air-side resistance loss of a single air cooler is lower than 500 Pa, i.e., \( \Delta P_o < 500 \text{ Pa} \)
2. The water-side resistance loss of a single air cooler is lower than 50 kPa, i.e., \( \Delta P_i < 50 \text{ kPa} \)
3. The total mass of the heat-exchanging components of a single air cooler is lower than 2t, i.e., \( M < 2t \)

5. Analysis of optimized results & project implementation
When multiple parameters are taken as the optimization objectives, there are usually no single results leading to the simultaneous acquisition of optimal solutions for all optimization objectives, because all parameters are mutually affected and constrained. Therefore, the results of multi-objective optimization are a series of values, which correspond to the value combinations of all optimization objectives. The final optimal values should be selected only from the aforementioned values depending on actual issues.

5.1. Analysis of optimized results
To demonstrate the multi-objective optimization results based on genetic algorithm, Pareto front was used to indicate the distribution of optimal objective parameters in this study. Pareto front indicates the distribution of all effective solutions obtained based on Pareto optimality principle in issues pertaining to multi-objective optimization, serving as the common method of handling multi-objective optimization.

When the initial population number of genetic algorithms is 750, a total of 263 sets of optimal solutions can be obtained through 300 iterative computations. Figures 4-9 show the Pareto distribution diagrams for design results of multi-objective optimization of rolled air cooler based on genetic algorithm. Such figures show the spatial distribution and mutually changing relationship of solutions for the total heat transfer coefficient \( K \), air-side resistance loss \( \Delta P_o \), water-side resistance loss \( \Delta P_i \), and mass \( M \). Based on the specification requirements (JB/T 2728.3-2008 “Gas Cooler for Motor”), solutions for \( K \) that is greater than 880 W/(m²·K) and \( \Delta P_o \) and \( \Delta P_i \) that are smaller than 400 Pa and 40 kPa respectively were selected, and a total of 32 sets of solutions were in line with the requirements.

5.2. Selection of optimized results and equipment customization
Based on the heat-exchanging performance, operations, investment costs, and actual manufacturing conditions, the 11th set of values were selected for customized processing, installation, and transformation. The tube length was 3400 mm, the fin length was 0.4 mm (four rows, six tubes/row), the tube diameter was 35 mm, the fin pitch was 2.7 mm, the core component mass was about 500 kg, and the initial investment cost was about 60,000 yuan/set.
Fig. 4 Optimization results distribution ($K$ & $\Delta P_o$)

Fig. 5 Optimization results distribution ($K$ & $\Delta P_i$)

Fig. 6 Optimization results distribution ($K$ & $M$)

Fig. 7 Optimization results distribution ($\Delta P_o$ & $\Delta P_i$)

Fig. 8 Optimization results distribution ($\Delta P_o$ & $M$)
5.3. Project implementation and operation effects

According to the manufacturing indicators, inspection and repair system of the plant, the replacement and transformation of air cooler of the unit No.1 were included in the repair plan A. Upon the arrival of customized equipment, the original air cooler was removed and replaced with a new air cooler during the inspection and repair, the cooling water pipeline was transformed, and a filter screen was installed. The drain holes at the lower part of the generator air chamber were inspected and dredged so as to eliminate moisture condensation. The automatic exhaust steam valve at the high level of return pipe was set to exhaust accumulated gas.

After start-up, the flow was measured through the ultrasonic flowmeter, and the differential pressure and temperature of inlet and outlet air of air cooler were measured. After the cooler was put into operation, the operating records of the new and original coolers were compared (as shown in Table 3).

| Item                        | Unit | Before  | After  |
|-----------------------------|------|---------|--------|
| Ambient temperature        | °C   | 31.5    | 33.2   |
| Atmospheric pressure       | KPa  | 905     | 891    |
| Humidity                   | %    | 30      | 44     |
| Unit load                  | MW   | 127.79  | 156.03 |
| Water inlet temperature    | °C   | 33      | 34.7   |
| Water outlet temperature   | °C   | 38.5    | 46.3   |
| Water flow                 | t/h  | 55.7    | 64.9   |
| Hot air temperature        | °C   | 86.9    | 67.2   |
| Cold air temperature       | °C   | 52.1    | 39.5   |
| Air-side differential pressure | Pa  | 1397    | 473    |
| Water-side differential pressure | KPa | 95.35   | 37.29  |

The comparison with the operating parameters during the same periods last year suggests that the air and temperature cooling requirements of generator are met, and the generator can run at full load. The reduced resistance of air cooler, ventilation loss of generator, and water-side differential pressure contribute significantly to lowering down the plant power consumption. Under the external high temperature conditions, the cooling requirements of the unit operating at full load are met.

6. Conclusion

In this study, the air cooler was optimized and replaced, so as to tackle the issue that the generator could not run at full load because of high air temperature. During the design of the air cooler, the basic structure parameters of air cooler were improved through multi-objective optimization based on genetic algorithm. Additionally, the heat transfer coefficient, quality, resistance, cost, and economic benefits of the air cooler were effectively improved. The operational observations exhibited that the replaced air cooler was under normal operation and met transformation requirements, and the unit
optimization and transformation were successfully performed. In addition, it was proved that such
optimization method could be employed for the design and optimization of generator air cooler
structure, thereby providing practical references for transformation of other units.

Nomenclature

- **A**: Area, m²
- **A_b**: External surface area of the light tube, m²
- **A_o**: Total external surface area, m²
- **c**: Cost coefficient
- **C**: Cost, yuan
- **c_p**: Constant pressure specific heat, J/(kg·K)
- **d**: Diameter, m
- **d_h**: Hydraulic diameter, m
- **G**: Mass velocity, kg/(m²·s)
- **h**: Coefficient of heat transfer, W/(m²·K)
- **H**: Height, m
- **K**: Total Coefficient of heat transfer, W/(m²·K)
- **L**: Length, m
- **m**: Mass flow, kg/s
- **M**: Weight, kg
- **n_f**: Number of fins
- **n_t**: Number of tubes per row
- **N_p**: Number of tube passes
- **N_t**: Number of tube rows
- **P_f**: Fin spacing, m
- **P_l**: Row spacing, m
- **P_r**: Prandtl number
- **P_t**: Tube spacing, m
- **ΔP**: Pressure loss, Pa
- **Q**: Heat exchange amount, W
- **r**: Fouling thermal resistance, (m²·K)/W
- **R**: Thermal resistance, (m²·K)/W
- **R_j**: Thermal contact resistance, (m²·K)/W
- **Re**: Reynolds number
- **t**: Temperature, °C
- **T**: Temperature, K
- **v**: Flow rate, m/s
- **W**: Width, m
- **W_p**: Total pump power, W
- **ξ**: Tube side pressure loss fouling correction
- **δ**: Thickness, m
- **η**: Efficiency
- **λ**: Coefficient of thermal conductivity, W/(m·K)
- **μ**: Dynamic viscosity, kg/(m·s)
- **ρ**: Density, kg/m³

Subscript

- **a**: Air
- **b**: Wall surface
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