Field Test and Research on Heat-transfer Performance of Plate Air-cooled Condenser

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Abstract

Based on field test data, the performance of plate air-cooled condenser unit is studied and simulated using CFD software, and numerical results agree well with actual data. Then with VAV (variable air volume) numerical simulation, the correlation formula between Nusselt number and Reynolds number is acquired. And with numerical simulation of variable surrounding DB (dry bulb) temperature, the outlet air temperature is shown linear to the surrounding. When surrounding temperature gets 34.2°C, the outlet reaches 69°C. That high surrounding temperature could reduce output of condenser. Aiming at the problem of condenser output limit, the spray humidification system is presented here to reduce inlet air temperature. Then temperature drops of different contact factors are studied, and results show the higher surrounding temperature, the bigger temperature drop, which means spray humidification system could reduce inlet air temperature effectively, especially in high surrounding temperature.

Keywords: plate air-cooled condenser; correlation formula; variable surrounding DB temperature; spray humidification system

1. Introduction

Air-cooling has been found favorable not only in water-deficient area but also in water-abundant area due to environmental and economical needs. Air-cooling system has shown increasingly wide application in air-cooled condenser of steam turbine in garbage power station, gas-steam combined cycle, air-cooled condenser in steam turbine or tow turbine of mining enterprise autonomous power station and air-cooled condenser in dry-wet jointly combined cooling system. Development & application appears more urgent due to severe shortage of water in China, which has shown brilliant prospect of application. [1] Strengthening of heat-transfer, reduction of flow resistance as well as enhancement of compactness has been considered crucial for air-cooler due to the minor air heat-transfer coefficient. Plate heat-transfer has been featured by high efficient and well-compactated in structure. Plate air-cooler, which is composed of plate heat-transfer and air-cooling technology has shown broad prospect of application in market due to its structural advantage against traditional tube type or finned tubular air-cooler. [2]
This article has carried out analog calculation of change in major parameter such as different ventilation rates, environmental air DB temperature based on spot checking in terms of numerical simulation so as to come up with suggestion for application of spray humidification system in view of higher efficiency.

2. Field test for Plate Air-cooled Condenser

2.1 Description of field test

One saturated & waste steam power generation project of certain enterprise introduced 5 pieces of foreign generator units with power plant plate steam-condensing device (corrugated steel sheet will be adopted as heat-transfer element) being utilized for dead & condensed steam device. 20 downstream plate bundles and 4 counter-current plate bundles are contained in site project with width of 2.13m while consisting of 174 plate & pipe (spacing 6.4mm). Downstream plate bundles and counter-current plate bundles are 6×0.2 and 5.4×0.2 respectively.

2.2 Data record

Air side parameter: Average air temperature in entrance and exit are \( t_1 = 27.1^\circ C \) (12 checking points) and \( t_2 = 62.8^\circ C \) (21 checking points), average temperature in plate surface is \( 75.1^\circ C \) (8 checking points), average wind speed in air-cooled condenser plate beam is 1.58m/s (5 checking points).

Steam lateral parameter: Steam flow \( W = 30.17t/h \), exhaust steam pressure \( p_c = 51.23kPa \), exhaust steam temperature \( t_c = 80.8^\circ C \), condense water temperature \( t_w = 66.3^\circ C \).

2.3 Collection and calculation of test data

Beam bundle windward acreage \( A_u = 2.13 \times (6 \times 20 + 5.4 \times 4) = 301.608m^2 \)

Air volume flow \( Q_t = A_u V = 301.608 \times 1.58 \times 3600 = 1715546m^3/h \)

Tested thermal load of air-cooled condenser

\[ \Phi = \rho A_u V c_p (t_2 - t_1) = 1.11 \times 301.608 \times 1.58 \times 1.005 \times (62.8 - 27.1) = 18.978MW \]

\( \rho \) - air density, kg/m\(^3\); \( A_u \) - side flow acreage of air, \( m^2\); \( V \) - average wind speed, m/s; \( c_p \) - specific heat of air, kJ/(kg·\( ^\circ C \)); \( t_2 \) - air temperature in entrance, \( ^\circ C \); \( t_1 \) - entrance air temperature \( ^\circ C \)

Check calculation of steam side thermal load

\[ \Phi' = W (h_e - h_c) = 30.17 \times (2646.4 - 277.6) / 3.6 = 19.851MW \]

Where, \( W_e \) - steam flow in air-cooled condenser is 30.17t/h; \( h_e \) - exhaust steam heat enthalpy, kJ/kg which can be calculated presently as 2646.4kJ/kg in accordance to exhaust steam pressure \( p_e = 51.23kPa \) and exhaust temperature \( t_e = 80.8^\circ C \); \( h_c \) - condense water heat enthalpy, kJ/kg, \( h_c = c_p t_w = 4.1868 \times 66.3 = 277.6kJ/kg \).

Test data is reliable when the relative error between \( \Phi \) and \( \Phi' \) is controlled within 5%. Therefore the measured data can be used as boundary condition & checking data for numeral simulation of air-cooling unit.

3. Numeral Simulation of Air-cooled Unit Under Test Condition

3.1 Physical model
Heat transfer plate bundle displayed in ridge shape is comprised by 174 plate tube, each of which is the heat-transfer element in plate air-cooled condenser. Air-cooled condenser is comprised by several air-cooling units, each of which is formed by 6 plate bundles together with bottom blower station as well as the ambient wind retaining wall. Air-cooling units are relatively simplified in modeling process in that rectangle as well as its internal structure will be adopted for replacement (size 7m×7m×10m), as shown in fig.1.

![Fig.1 Physical model of air cooling unit](image)

3.2 Control equation

Flow& heat transfer of air should be in accordance to continuity equation, \( k - \varepsilon \) momentum equation as well as energy equation. \([3][4]\).

Condense process of steam in heat-transfer plate bundles is simplified by the author merely taking condensing heat release into consideration thus add source item to energy equation while irrespective of specific condensing process of steam in heat-transfer plate bundle. Thermal load of steam condensing process is assumed as 4.96MW.

Heat transfer plate bundles are considered as porous medium area\([5]\), which mean that inertial loss item will be added to momentum equation. As for simple& uniform porous medium:

\[
S_i = \frac{p}{n} = C_2 \cdot \frac{1}{2} \cdot \rho \cdot u_{mag} \cdot \vec{u}_i
\]  

(1)

\( S_i \) is the additional inertial loss item in momentum equation; \( p \) represents pressure; \( \vec{n} \) is normal vector of heat-transfer plate bundle; \( u_i \) is velocity rise(free flow velocity), \( u_{mag} \) is absolute value of velocity; \( C_2 \) is inertial resistance coefficient.

The relation of additional source item with air free flow velocity in momentum is crucial for calculation. Relation (2) of resistance with heat-transfer plate bundles velocity \( u_i \) can be fitted as per test data in document \([6]\) (variable wind resistance performance test of heat-transfer plate bundles under working condition).

\[
\Delta p = 62.75 \cdot b \cdot u_i^2
\]  

(2)

\( b \) is the width of plate, \( C_2 = 21.45 \) can be obtained by fit relation.

Entrance boundary of calculation area is considered as air-cooling units fan entrance while exit boundary is taken as model top surface with heat-transfer plate bundles regarded as porous medium area.
and wall boundary adopted as surrounding partition. Configuration of module work station shall be Intel Xeon E5420, 2.5G Hz primary frequency with 8G memory.

3.3 Confirmation of test result

Air-cooling unit \( Y = 0 \) sectional temperature distribution drawing with uniform distribution of air temperature in air-cooling unit is shown in Fig. 2. Air temperature from internal side of heat-transfer plate bundles and average temperature value in entrance are exactly 300K and 334.7K respectively when air temperature in other regions is between them. Numeral model is reliable when numeral value entrance temperature 334.7K is consistent with test entrance temperature 335.8K, based on which further research on heat-transfer performance of air-cooling units is necessary.

![Fig.2 Sectional temperature distribution drawing of air-cooling unit Y=0](image)

4. Numeral Simulation of Variable Wind Rate

Average operation efficiency of fan is only 60% with 40% balance under working condition. Varying curve of heat transfer coefficient in air-cooling unit with internal air flow velocity can be obtained by changing test model wind rate as well as relative parameter under working condition (simulated wind rate varying ranges 100 ~ 240kg/s). Heat-transfer coefficient of air-cooling unit increases as the addition of air flow, which is shown in Fig.3.

![Fig.3 Heat transfer coefficient changes in accordance with the air flow speed](image)

Correlation formula for heat-transfer under forced air turbulence can be gained as below as \( Pr \) work as constant for as for air. \( Nu = f(Re) \)

Correlation formula between aerial \( Nu_a \) and \( Re_a \) can be obtained as per calculation method listed in chapter 2.3.2:

\[
Nu_a = 0.007Re_a^{1.031}
\]

\[
2184.5 \leq Re_a \leq 5230.9
\]
Constant coefficient in correlation formula is determined by numeral simulation representing the law of homogeneous heat-transfer phenomenon therefore is suitable for analysis & calculation of similar phenomenon as per similar theory.

5. Spray Humidification Air Cooler

5.1 Temperature numeral simulation of variable environmental DB

Aerial DB temperature range 13.4°C~34.2°C in July, the hottest month in for power plant can be obtained as per typical meteorological annual parameter of DB temperature in the hottest month[8]. Aerial entrance DB temperature $t_{out}$ has been found linear increasing trend with ambient temperature $t_{sur}$ by changing test model aerial entrance temperature under working condition, which is shown in Fig.4.

$$t_{out} = t_{sur} + 34.8$$

Fig.4 Temperature of entrance air DB changes in accordance with environment temperature

DB air temperature in entrance reaches 69°C when ambient temperature is 34.2°C, predicting that application of air-cooling technology is substantially influenced by ambient DB temperature, which is shown in Fig.6. During hot seasons especially in summer, spray humidification air-cooling system is recommended for enhancement of working efficiency due to the failure of entrance temperature of air cooler machine cooled by medium to technical requirement

5.2 Spray humidification air cooler

Fundamental rationale of spray humidification air cooler is that vaporization and cooling of entrance air will be conducted for reduction of its temperature to WB temperature, after which heat-transfer surface will be created for bigger heat-transfer difference and enhancement of heat-transfer.

Air of condition 1 can be ultimately converted into 3 if sufficient heat-humidity exchange can be guaranteed during the contact of air with water whereas in actual process condition 2 will be realized due to improper exchange. Meanwhile, contact coefficient $\eta_2$ can be used to represent perfect degree of aerial change for expression of heat-humidity exchange effect, which is shown as follows:

$$\eta_2 = \frac{t_1-t_3}{t_1-t_2} = \frac{t_1-t_2}{t_1-t_{s1}}$$

Contact coefficient 60%, 70%, 80%, 90% and 100% can be for research on cooling effect as shown in Fig.5. Temperature drop can be 5°C when ambient temperature is 34.2°C, $\eta_2 = 100\%$ (cooling limit, which means entrance air is cooled to WB temperature), even if $\eta_2 = 60\%$, 3°C drop can be detected, which means that humidification system can show higher efficiency in hot season since DB temperature drop has shown increasing trend with ambient temperature.
6. Conclusion

Calculation model has been established based on test operation data of plate air-cooled condenser in one power plant and typical simulation result under working condition is consistent with actual result. Further research on heat-transfer performance of air-cooled condenser can be carried out with the aid of this model.

Correlation formula between $Nu_a$ and $Re_a$ has been obtained by simulating variable wind value and is suitable for analysis and calculation of similar phenomenon.

Linear increasing trend of aerial entrance DB temperature with ambient temperature can be gained by simulation of variable environmental DB temperature value. Aerial entrance temperature arrives at 69°C when ambient temperature is 34.2°C, which means that application of air-cooling technology is substantially influenced by ambient temperature.

Spray humidification air-cooling system is recommended for enhancement of efficient. Temperature drop value will be calculated by the use of different contact coefficient. Temperature drop can be 5°C when ambient temperature is 34.2°C, $\eta_2 = 100\%$ even if $\eta_2 = 60\%$, 3°C drop can be detected as well. DB temperature shows an increasing trend, which means that spray humidification will enhance working efficiency under working condition.

References