Adaptive Modeling and Operation Optimization for the Cold End System of Thermal Power Units based on Mechanism and Statistical Analysis

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Abstract. The operation optimization for the cold end system is an efficient means to improve the economy of steam turbine units. To compensate for the inadequacy of the traditional mechanism analysis utilized in obtaining actual operating characteristics of the cold end system, the prediction model of the exhaust pressure was established on the basis of mechanism analysis combined with data from the operation process. An online adaptive updating strategy was introduced to guarantee the modeling accuracy. A discrete model of the cooling tower outlet water temperature (CTOWT) was constructed based on the operation data partitioned into different groups according to the pump operating mode change (POMC). Combining the above two models, the coupled model of the cold end system was therefore obtained. A model-based operation optimization system was then implemented for the cold end system in a coal-fired power plant. Experimental trials authenticate that the optimization suggestions provided by the system can effectively enhance the benefit of power generation.

1. Introduction

In recent decades, with large-scale renewable energy connected to the power grid, coal-fired power plants more frequently run in cycling load operation mode to compensate for the intermittency of renewable energy [1]. As a significant auxiliary system of steam turbine, the cold end system has direct impact on the economic operating of the units when the load varies in a wide range. Note that for every 1kPa decrease in the back pressure, the steam turbine output will increase (1%-2%)[2], whereas the reduction of the back pressure is at the expense of extra power consumption of the circulating pumps. It is estimated that the circulating water pumps consume about (1%-1.5%) of the total power of the plant[3].

There has been quite a bit of attention attracted on the cold end system modeling methods and operation optimization strategies[4–6]. Li et al.[7] applied the least square support vector machine algorithm (LSSVM) to modify the deviation of operation data and the results obtained from mechanism equations. However, it is worth noting that the exhaust pressure is essentially different from the condenser pressure. Research shows that the uneven flow field distribution in the complicated exhaust passage seriously affects the thermal performance of the condenser[8]. In terms of cooling tower modeling, CFD (computational fluid dynamics) models were employed to investigate cooling tower characteristics[9–11], which were generally computational expensive. Recently, data-driven approaches have been gaining increasingly popularity in modeling the cooling tower. Considering the operation data within the same operating mode have the similar characteristics, Pan et al.[12] developed a set of local linear models on the basis of data partitioned into several groups by the fuzzy c-mean clustering algorithm. Similarly, Wang et al.[13] proposed a dimensionless index to describe the cooling capability of a cooling tower.

In this paper, the implementation of operation optimization is founded on the conception of evaluating the impact of a specific pump combination change on the cold end system. In view of this, a discrete model of CTOWT is therefore constructed with the data partitioned by different pump operating mode changing way. An adaptive update strategy is introduced to correspond with the actual process. Thereafter, the prediction model of exhaust pressure is further established on the basis of mechanism model of with least squares support vector machine (LSSVM) algorithm. Embedded with the coupled model, a cold end optimization system is developed to provide guidance for the actual operation.

2. Overview of the system background

Structure schematic diagram of the studied cold end system shared by two turbo-generator units is depicted in figure1. On the premise of safety, the circulating water
system generally operates in the following five modes: two units with two pumps (P2), three pumps (P3), four pumps (P4), five pumps (P5) and six pumps (P6). The research aims to investigate which pump operating mode under a certain condition can simultaneously fulfill the requirements of safety and economy.

3. Modeling of the cold end system

3.1. Modeling of steam exhaust pressure

3.1.1. Thermodynamic characteristic model of the condenser.

An adaptive heat transfer coefficient updating strategy is considered to enhance the accuracy of the model. The thermodynamic characteristic model of the condenser is shown in figure 2.

The energy balance equation of the process is expressed as follows:

\[ k A_c \Delta \tau_m = D_e C_p (t_{w_2} - t_{wi}) \]

Equation (2) calculates the logarithmic mean temperature difference:

\[ \Delta \tau_m = \frac{\Delta \tau_{\text{max}} - \Delta \tau_{\text{min}}}{\ln \frac{\Delta \tau_{\text{max}}}{\Delta \tau_{\text{min}}}} = \frac{(t_e - t_{wi}) - (t_e - t_{w_2})}{t_e - t_{w_2} - t_e - t_{w_2}} \]

Combining equations (1) and (2), the OHTC can be obtained as follows:

\[ k = \frac{D_e C_p}{A_c} \ln \frac{t_e - t_{wi}}{t_e - t_{w_2}} \]

The Antoine equation utilized to calculate the saturated vapour pressure is presented below:

\[ \ln p_v = a - \frac{b}{t_e + c} \]

Thus, the saturation temperature can be inversely derived from equation (4):

\[ t_e = \frac{3826.36}{9.3876 - \ln p_v} - 227.68 \]

Note that the range applicable to the saturation temperature is 17°C to 227°C [15].

With equations (3) and (5), the overall heat transfer coefficient of the condenser is presented as follows:

\[ k = \frac{D_e C_p}{A_c} \ln \frac{3826.36}{9.3876 - \ln p_v} - 227.68 \]

\[ k = \frac{D_e C_p}{A_c} \ln \frac{3826.36}{9.3876 - \ln p_v} - 227.68 - t_{w_1} \]

The update logic diagram is depicted in figure 3. Based on the latest N \( \hat{k} \) values in the database, the overall heat transfer coefficient utilized to calculate the thermal characteristics of the condenser can be obtained as follows:

Based on the latest N \( \hat{k} \) values in the database, the overall heat transfer coefficient utilized to calculate the thermal characteristics of the condenser can be obtained as follows:
\[ K = \sum_{i=1}^{N} \mu_i k_i \]  

(7)

The terminal temperature difference is expressed as equation (8):

\[ \delta t = \frac{\Delta t_w}{\exp(Kt_w/1000C_pD_w) - 1} \]  

(8)

From equation (1) to equation (8), the condensation temperature of steam in the condenser can be attained:

\[ t_s = t_w_1 + \Delta t_w + \delta t \]  

(9)

The condenser pressure can be obtained from the following empirical formula:

\[ p_c = \left( \frac{t_s + 100}{57.66} \right)^{7.46} \times 9.806 \times 10^{-6} \text{(MPa)} \]  

(10)

The temperature of circulating water at the outlet of the condenser is as follows:

\[ t_{w_2} = t_{w_1} + \Delta t \]  

(11)

From equation (1) to equation (11), the calculation model of condenser outlet water temperature and pressure is established:

\[ t_{w_2} = f_i(D_c,t_{w_1},D_w) \]  

(12)

\[ p_c = f_p(D_c,t_{w_1},D_w) \]  

(13)



| Error | #1 cooling tower | #2 cooling tower |
|-------|------------------|------------------|
|       | Absolute error /°C | Relative error /% | Absolute error /°C | Relative error /% |
| Max   | 1.280            | 5.006            | 1.311              | 5.977              |
| Min   | 0.003            | 0.000            | 0.000              | 0.000              |
| Mean  | 0.325            | 1.884            | 0.338              | 1.513              |
| Standard deviation | 0.237 | 0.014 | 0.244 | 0.013 |

Table 1. Error statistics of the CTOWT model.

3.1.2. Deviation modification based on LSSVM.

Numerous investigations revealed that the aerodynamic performance of exhaust passage is influenced by multi factors like inlet swirl distribution[14], steam wetness[15], non-uniformity flow field distribution[16]. The major operating parameters such as load, condenser inlet water temperature, exhaust flow and condenser pressure were reasonably selected as the input of the data-driven model, and LSSVM algorithm was then applied to establish the deviation modification model.

3.2. Modeling of the cooling tower

An analysis program was developed in-house to capture start-stop action happening time of circulating water pumps by scanning historical data. Once the action captured, working conditions of a certain POMC are recorded subsequently and hence the POMC dataset are generated, namely, \(D_1, D_2, \ldots, D_n\). Each subset \(D_i\) consists of data that categorized by one specific pump operating mode changing way. Imbedded with statistical analysis conducted on POMC dataset in the previous section, LSSVM algorithm was utilized to fit the relationship between the variation of CTOWT with the load, and the ambient parameters under different POMC, namely:

\[ \Delta t_2 = f_{\text{ct}}(N,t,u,p,A,B) \]  

(14)

Thus, the CTOWT can be calculated as follows:

\[ t_{w_2} = t_{w_1} + \Delta t_2 \]  

(15)

3.3. Coupling model of the cold end system

\[ t_{w_2} = t_1, t_{w_1} = t_2 \]  

(16)

According to equation (16), the models of exhaust pressure and the cooling tower are coupled together to predict the exhaust pressure when the pump operation mode switches between different operation modes under given load and environmental factors, that is:

\[ p_1 = f(N,D_c,t,u,p,D_w,t_2,A,B) \]  

(17)

The calculation flow chart of the exhaust steam pressure is presented in figure 4.

3.4. Model validation

The 2×1050MWe coal-fired power plant is in Taizhou, China. Figure 5, figure 6 and table 2 present the accuracy of the cooling tower models. Figure 7 and table 2 present the accuracy of the coupled model. The average relative error is about 1.7%, indicating high consistency between the model and the actual process. Hence, the model of exhaust pressure can be reasonably further applied to the operation optimization.
Ambient Conditions: $p, u, t$
Operating Conditions: $N, A, D_c, D_w, t_2$

Assumption:
Pump Operating Mode Changes to B

Match the POMC in Model Database: from A to B

Calculate Variation of CTOWT: $\Delta t_2$

Calculate Thermal Parameters in Condenser: $\delta t_2, \delta t, t_2, t_2, t_2$

Calculate Condenser Pressure: $p_c$

Deviation Modification

Turbine Exhaust Pressure $p_t$

Fig. 4. Calculation flow chart of cold end system model

Table 2. Error statistics of steam turbine exhaust pressure model.

| Error    | Steam turbine exhaust pressure                                      | Absolute error /kPa | Relative error /% |
|----------|---------------------------------------------------------------------|----------------------|-------------------|
| Max      | 0.358                                                               | 4.986                |
| Min      | 0.000                                                               | 0.002                |
| Mean     | 0.141                                                               | 1.727                |
| Standard deviation | 0.079                                                                | 0.998                |

Fig. 5. Comparison between measured and predicted #1 CTOWT

Fig. 6. Comparison between the measured and predicted #2 CTOWT

Fig. 7. Comparison between the actual and predicted exhaust pressure

Input:
Ambient Conditions, Operating Conditions and Current Pump Operating Mode

Assumption:
Pump Operating Mode Changes to Another Possible Mode

Predict Exhaust Pressure Under the Assumed Mode Based on the Model of Cold End System

Calculate Additional Power Output Based on Power Characteristic to Back Pressure of Steam Turbine

Calculate Unit Load Under the Assumed Mode

Calculate Energy Consumption of the Assumed Mode

Calculate Net Power Output Under the Assumed Mode

Is the Traversal Completed?

No

Yes

Compare Net Power Output Under All Possible Modes

Select the Optimal Mode with the Highest Net Power Output

Output:
The Optimal Mode and Predicted Parameters of the Cold End System

Fig. 8. Flow chart of the operation optimization for the cold end system
4. Optimization strategy and implementation

Figure 8 presents the optimization process. Take a pump operating mode change at 13:30 on August 3, 2018 as an example. Before the switch, the loads of unit #1 and unit #2 were 800MW and 815MW, respectively. The pump operating mode was set at P6, and the ambient temperature was 34°C, the humidity 63%. At the moment, the cold end optimization system suggested the operating mode of P5 based on the embedded model. Following the recommendation, the on-duty operator stopped a pump and the parameters of the cold end reached a stable state after about 35 minutes. Table 3 summarizes the economic indexes of the whole plant after a possible pump operation mode change under such circumstance. Compared with five running pumps, the increased output power of the turbines is little to offset the more energy consumption under the operating mode of six pumps.

5. Conclusion

(1) The exhaust pressure was no longer considered equivalent to the condenser pressure but the result further modified with LSSVM algorithm on the basis of condenser pressure. Forgetting factors were introduced in modeling the condenser pressure as an adaptive update strategy of the overall heat transfer coefficient. The mean relative error is within 2% for the exhaust pressure model.

(2) A discrete model of the cooling tower outlet water temperature was established based on the data partitioned into different groups according to the pump operating mode changing way. The prediction accuracy and calculating speed revealed that the proposed modeling approach could fulfill the requirements of field application.

(3) A cold end optimization system was developed based on the adaptive model. The reliability and capability of the implemented system were demonstrated in experimental trials. The whole work offers a practical opportunity for the operators to conserve energy in the operation of the cold end system.

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Table 3. Economic indexes of the whole plant after a specific POMC

| POMC         | Net power/kW | Variation of auxiliary power consumption/% | Variation of heat consumption/(kJ/(kW·h)) | Variation of coal consumption/(g/(kW·h)) |
|--------------|--------------|--------------------------------------------|------------------------------------------|------------------------------------------|
| From P6 to P5| 2341         | -0.215                                     | -1.224                                   | -0.145                                   |
| From P6 to P4| 1440         | -0.899                                     | 26.594                                   | -0.089                                   |
| From P6 to P3| -2796        | -1.347                                     | 63.806                                   | 0.173                                    |
| From P6 to P2| -40548       | -1.801                                     | 262.756                                  | 2.509                                    |
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Nomenclature

\( k \) overall heat transfer coefficient, kJ/(m\(^2\)·h·K)

\( A_c \) outside surface of tube bundles, m\(^2\)

\( \Delta t_m \) logarithmic mean temperature difference, K

\( D_c \) condensate mass flow rate, t/h

\( D_w \) circulating water flow rate, t/h

\( C_p \) specific heat capacity of water, kJ/ (kg·K)

\( t_{w1} \) inlet water temperature of condenser, K

\( t_{w2} \) outlet water temperature of condenser, K

\( t_s \) condensation temperature of exhaust steam, K

\( p_c \) condenser pressure, MPa

\( K \) updated overall heat transfer coefficient, kJ/(m\(^2\)·h·K)

\( \delta t \) terminal temperature difference of condenser, K

\( \Delta t_w \) temperature rise of circulating water in condenser, K

\( t_1 \) inlet water temperature of cooling tower, K

\( t_2 \) outlet water temperature of cooling tower, K

\( N \) load, MW

\( t \) dry bulb temperature, K

\( u \) relative humidity, %

\( P \) atmospheric pressure, kPa

\( \Delta t_2 \) outlet water temperature variation of cooling tower after pump operating mode change, K

\( A \) pump operating mode before change

\( B \) pump operating mode after change

\( t_{2,1} \) outlet water temperature of cooling tower before pump operating mode change, K

\( t_{2,2} \) outlet water temperature of cooling tower after pump operating mode change, K

\( D \) dataset

\( \Delta p_t \) deviation between the exhaust pressure and the condenser pressure

\( \Delta N_n \) net power of the units, kW

\( \Delta N_t \) turbine additional power after pump operating mode change, kW

\( \Delta N_p \) increased energy consumption of pumps after pump operating mode change, kW

Subscripts

\( A \) pump operating mode before change

\( B \) pump operating mode after change

\( w \) circulating water

\( c \) condenser

\( s \) saturation state

\( m \) mean

\( t \) turbine

\( P \) pump

Abbreviations

CTOWT cooling tower outlet water temperature

FOMC pump operating mode change

LSSVM the least square support vector machine algorithm

CTIWT cooling tower inlet water temperature

OHTC overall heat transfer coefficient

EP exhaust pressure

CP condenser pressure

Greek symbols

\( \mu \) forgetting factor