Multi-parameter Joint Optimization Based on Steam Turbine Thermal System Characteristic Reconstruction Model

S Liu¹,², J Shen¹,³, P H Wang¹
¹School of Energy & Environment, Southeast University, Nanjing 210096, Jiangsu Province, China
²School of Mechanical & Electrical Engineering, Jinling Institute of Technology, Nanjing 211169, China
E-mail: shenj@seu.edu.cn

Abstract. Steam turbine thermal system characteristic reconstruction model for multi-parameter joint optimization is proposed in this paper. This paper has two main improvements. First, a steam turbine thermal system characteristic reconstruction model is conducted, which includes turbine, condenser and heater. There are three iterative nested algorithms in this model. In the innermost layer, a variable condition calculation model of condenser is nested, and the exhaust flow and exhaust steam pressure can be calculated. In the outermost layer, the power equation is applied as the iteration ends. In this model, simulation of thermal system of steam turbine three kinds of disturbing factors demand can be realized well and the joint optimization of thermal system of steam turbine flow section and the cold end can be realized. Through calculation and comparison, the model constructed in this paper is proved to have high accuracy, and the calculation process is stable and convergent. At the same time, the joint optimization method involving turbine main steam pressure and exhaust pressure is proposed here. In this method, heat consumption rate is the optimization objective, with steam turbine thermal system characteristic reconstruction model, the optimal main steam condenser and vacuum pressure were obtained. The result of a 600MW condensing unit showed that the method could be synchronized to determine the optimal combination of circulating pump, the best vacuum and sliding pressure curve of main steam under different load and different ambient temperature. Comparison results also showed that the joint optimization solution of main steam pressure and exhaust pressure is superior to optimized solution main steam pressure individually.

1. Introduction
The optimization of steam turbine thermal system operation parameters is of great significance to the optimization operation and system evaluation of thermal power units. Researchers have conducted in-depth research on the main operating parameters of steam turbine thermal system, i.e. main steam pressure and exhaust steam pressure, and achieved some results.

Main steam pressure optimization is also known as steam turbine hot-end optimization. Existing literature often focus on the separate optimization of main steam pressure parameters. The methods adopted can be summarized as test optimization and analytical optimization [1-6]. Exhaust steam pressure optimization of steam turbine is related to the cold end equipment of steam turbine. Exhaust steam pressure optimization is also known as cold end optimization, and is one of the important contents of optimization of operation parameters of thermal system. Similar to the main steam
pressure optimization, it can be divided into two categories: experimental optimization and analytical optimization [7-11].

At present, the optimization of main steam pressure and exhaust steam pressure of steam turbines is carried out independently, without considering their interaction [12-14]. However, in the actual thermal system, the change of main steam pressure will lead to the change of the steam state line of the thermal system, which will change the final exhaust steam volume and the condenser vacuum (the exhaust steam pressure of the steam turbine); when the circulating water volume changes, the exhaust steam pressure will also change, so the output power of the unit will change, leading to the change of the optimal main steam pressure change. The main steam pressure and exhaust steam pressure are two coupled parameters, so it is necessary to discuss their synchronous optimization. In the actual production process, the staff who carried out the sliding pressure optimization test also realized that the set value of the main steam pressure should be a function determined jointly by the load and the unit back pressure, rather than a function corresponding only to the single value of the load [15].

At the same time, the objective function of steam turbine exhaust pressure optimization is usually to maximize the difference between the output power of steam turbine and the incremental power consumption of the circulating water pump, while the objective function of the main steam pressure optimization is to minimize the heat consumption rate of steam turbine-generator units. There is a mismatch between these objective functions.

In summary, it is necessary to present a model for simulation and prediction of the thermal system characteristics to study the joint optimization. The model includes the off-design characteristics of the steam turbine, the heater in regenerative system and the condenser. In order to meet the need of joint optimization, the model adopts three layers of iteration cycles in the algorithm design. The inner layer is nested with the variable working condition calculation model of condenser. The exhaust flow rate, exhaust pressure and exhaust enthalpy can be calculated iteratively in a synchronous way, in order to realize the joint optimization of the flow passage and the cold end of the steam turbine thermodynamic system. The middle layer adopts a step-by-step variable working condition calculation that is combined with the stage group off-design calculation (intermediate stage). The outermost layer uses the power equation as the constraint condition at the end of iteration.

In this paper, the combined optimization of main steam pressure and exhaust steam vacuum for a 600 MW condensing unit is carried out with the heat consumption rate as the objective function. The optimal combination of circulating water pumps with different loads and ambient temperatures is calculated. The influence of ambient temperature on the optimization results is analyzed, and the sliding pressure curves at different temperatures are given. Finally, this paper will focuses on the difference between the joint optimization and the main steam pressure optimization, and provides results for both joint and individual optimization.

### 2. Steam Turbine Thermal System Characteristic Reconstruction Model

It is necessary to present a model for simulation and prediction of the thermal system characteristics to study the joint optimization. Steam turbine thermal system characteristic reconstruction is established in this paper. Variable working conditions are influenced by three classes of factors: external factors (these correspond to changes of load, environment temperature and running mode), loop parameters (that correspond to changes of main steam pressure, main steam temperature and reheating steam temperature) and equipment energy efficiency (correspond to steam turbine efficiency and temperature difference of heater). In applying the method sets of working condition parameters can be generated, which real units haven’t reach. By calculation test data, the turbo set state reconstruction model can reflect the respond changes of panel points of unit with different running states, and calculation results fit with real working condition data.

#### 2.1. Various Operation Conditions of Steam Turbine
- Grade group characteristic
The turbine itself is composed of a high-pressure, a medium-pressure and a low-pressure cylinder. Each cylinder contains a certain number of stages. According to performances in transient and off-design conditions, that correspond to changes in the operating conditions, where, the amount of vapor into the steam turbine is changed, the stages can be divided into three parts—a governing stage, an intermediate and a final stage, based on nozzle adjustment.

- The governing stage

The pressure ratio before and after the stage changes with the flow, so does the ideal enthalpy drop. When the turbine flow decreases, the pressure ratio gradually decreases and the governing stage enthalpy drop increases gradually. When the first governing valve is fully open and the second valve is just about to open, the pressure ratio achieves the minimum, and therefore the ideal enthalpy drop reaches its maximum. The ideal enthalpy drop increases, while the velocity rate and reaction degree decreases, deviating from the optimum value, and the stage efficiency is reduced accordingly.

The governing stage performs in a complex way, and generally calculate its efficiency in accordance with the adjustment. For a specific unit, it may also be calculated by polynomial curve fitting of the “flow-efficiency” scatter value provided by the manufacturer, or fitting the efficiency data according to the design conditions.

- The intermediate stage

When operating conditions change, the pressure ratio remains the same in the intermediate stage. Taking the conditions of Flugel formula into account and in order to simplify the modeling, the intermediate stage can be divided into a number of intermediate groups based on extraction ports. The ideal enthalpy drop of steam turbine is the function of the temperature before the stage and the pressure ratio. The steam temperature before the stage at all intermediate groups is essentially the same when operating conditions change not too much. The ideal steam enthalpy, the velocity rate and reaction degree of the groups remain unchanged, so the efficiency of the class group is unchanged.

- The final stage

According to Flugel formula application conditions, several stages from the extraction port to the exhaust end of the final heater are classified as the final stage. When the steam flow rate decreases, the heat transfer in the condenser doesn’t change too much, leading to small turbine exhaust pressure change, while the pressure before the stage reduces relatively more. Pressure difference before the stage and after is greater in transient and off-design conditions, since the pressure after the stage reduces less than the pressure before. Therefore, the pressure ratio increases, the stage ideal enthalpy drop decreases, and the pressure ratio of the final stage and ideal enthalpy drop has the greatest change. The velocity rate and the reaction degree of the stage increases as the ideal enthalpy drop decreases. The velocity rate deviates from the optimum and the stage efficiency is reduced accordingly. For a specific unit, it may also be calculated by the “flow-efficiency” curve provided by the manufacturer.

2.2. Variable Conditions of Condenser

Condenser is an important auxiliary in the steam turbine system variable operating conditions of condenser, i.e. condensing steam consumption, cooling water inlet temperature, and cooling water flow will change with the change of load.

The operating conditions of the condenser in the off-design conditions is called the Variable Conditions of Condenser.

Condenser vacuum depends on the condensation temperature $t_s$, mainly depends on the turbine exhaust steam load, cooling water flow rate $D_w$ and cooling water inlet temperature $t_{w1}$:

$$t_s = t_{w1} + \Delta t + \delta t$$  \hspace{1cm} (1)

where: $\Delta t$ is the circulating water temperature rise. According to the energy balance equation in the condenser:

$$\Delta t = \frac{D_w Q}{D_w C_w} = \frac{h_t - h_{w}}{4.1816 * m}$$  \hspace{1cm} (2)
where: $D_c$ is the exhaust steam consumption of steam Turbine, [t/h]; $C_w$ is the heat capacity of cooling water, [$kJ / (kg \cdot ^{\circ}C)$]; $m$ is the condenser circulating ratio. $\Delta t$ is the condenser terminal temperature difference, which is calculated by:

$$\Delta t = \Delta t \cdot \left( \exp(\frac{K \cdot F}{4186.8 \cdot D_w}) - 1 \right)^{-1}$$

Where: $F$ is the Condenser heat transfer area, $m^2$; $K$ is the Condenser heat transfer coefficient [$kJ / (m^2 \cdot h \cdot K)$], now commonly used in the modified Germain formula:

$$K = 4070 \beta_c \beta_w \beta_1 \beta_z \beta_d \beta_m \beta_a$$

Where: $\beta_c$, $\beta_w$, $\beta_1$, $\beta_z$, $\beta_d$, $\beta_m$, $\beta_a$ are the cleaning, flow rate, temperature, flow, steam load, pipe with wall thickness and air correction factor.

The above calculation method is the theory calculation method of the condenser variable conditions, analysis shows that, for the same state of the circulating pump and cooling water, when disturbance load or thermal parameters, Only the steam turbine exhaust impact the vacuum regenerator heater features.

Regenerative heater is an important equipment for thermal power units to improve the economy. Its operation performance directly affects the thermal economy of the entire unit. Its features are related to extraction flow, extraction parameters, and enter water parameters of the heater. When load changes, the parameters of steam turbine main flow and the extraction flow are changed, causing variable parameter operation of the heater. In addition, when the condenser outlet water temperature changes caused by the change of circulating water temperature the heater inlet water temperature changes and the heater working conditions change. Therefore, heater terminal temperature difference is always changing with the unit load in variable operating conditions. For the formula can be derived:

$$\frac{\Delta t}{\Delta t_h} = \left( \frac{D_h C_p}{K_h} - 1 \right) \cdot \frac{\Delta t}{\Delta t_h}$$

where: $A$ is the heater heat transfer area, $m^2$; $D_h$ is the feed water flowing through the heater; $\Delta t_h$ is the temperature rise of the feed water; $\theta$ is the temperature of heater; $K_h$ is the heat transfer coefficient of heater and $C_p$ is the specific heat at constant pressure of the feed water, [$kJ / (kg \cdot ^{\circ}C)$].

Since the Heater Terminal Temperature Difference is the main indicator for the evaluation of the heater, to transform formula (5) can be obtained the Heater Terminal Temperature Difference calculation formula under variable conditions:

$$\Delta t_h = \left( e^{\frac{D_h C_p}{K_h}} - 1 \right) \cdot \theta$$

The calculation of the heat transfer coefficient in Function has a variety of ways, when the heater in a normal operating state, $K_h$ in variable conditions can be approximately considered constant. At the same time, the Heater Terminal Temperature Difference and the unit load rate law has relationship. Law relationship can be used to simplify the heater Variation Calculation for the analysis of the specific unit of Variable Conditions.

2.3 Multi-parameter Joint Optimization Based on Steam Turbine Thermal System Characteristic Reconstruction Model

The algorithm of the multi-parameter joint optimization based on steam turbine thermal system characteristic reconstruction model is shown in figure 1.
Figure 1. Multi-parameter Joint Optimization Process

Figure 1 shows the calculation process, when the unit operating conditions change, according to the external power requirements, ambient temperature and operating mode requirements to determine the operating conditions of the unit. First assuming that the main steam flow remains the same, then cause the parameters which change the operating conditions into the corresponding computational model to calculate the extraction parameters of the flow section, according to the variable conditions model of regenerative heater and condenser to get the new heater parameters and exhaust parameters; according to the newly calculated Steam-Water Distribution to get the unit working fluid acting used external output power balance equation to get new main steam flow; New main steam flow re-substituting the previous steps, repeat counting the steps to get a new main steam flow and the unit thermal efficiency values. If the difference between the new value and the last calculated value to meet the accuracy calibration requirements End the calculation, otherwise repeat the iterative calculation until the results meet the requirements, and the calculation is finished.

3. Solution of Joint Parameter Optimization

3.1. Determination of feasible solution interval of system independent variables
Based on steam turbine thermal system characteristic Reconstruction Model, this paper studies the joint optimization method of main steam pressure and exhaust steam pressure of steam turbine. In this paper, the main steam pressure and circulating water volume are chosen as the independent variables of the system. The range of optimization parameters is analyzed as follows: for the sake of unit operation safety, the upper and lower limits of main steam pressure are 16.67 MPa and 12 MPa, and the step of pressure change in optimization calculation is 0.1 MPa.

According to the characteristics of condenser under different working conditions, when the inlet temperature of circulating water is constant, the circulating water flow rates is the decisive factor of condenser vacuum or exhaust pressure. This paper focuses on the heat consumption rate of thermal system under different circulating water flow, and the optimal circulating water flow is the optimal exhaust pressure. The typical domestic 600 MW unit that is studied here in paper consists of two double-speed pumps. Two-speed pumps can operate at high and low speeds.

3.2. Objective function and constraints
Under a certain load of the unit, the change of the main steam pressure will lead to the change of the unit flow rate and exhaust steam volume, which will affect the change of the steam flow into the condenser, and thus change the steam load of the condenser. Similarly, under the condition of constant cooling water volume, changing the inlet water temperature and steam load will cause the change of the condenser pressure. The change of the pressure of the generator will affect the change of the load of the unit, and then the change of the optimum main steam pressure. It can be seen that the condenser vacuum and the main steam pressure of the steam turbine are a pair of coupled parameters. The objective of multi-parameter coupling optimization is to minimize net heat consumption, so the objective function of the optimization model is as follows:

$$\min (HR) = F(N, p_0, n, T)$$  \hspace{1cm} (7)

Where: $N$ : Unit load; $p_0$ : Main Steam Pressure for Steam Turbine Sliding Pressure Operation $p_{0_{\min}} \leq p_0 \leq p_{0_{\max}}$; $T$: Ambient temperature of unit operation $T_{w0}$; $n$: Operation mode of circulating water pump.

Six values of ambient temperatures (5, 10, 15, 20, 25, 30), and partial loads (50%, 60%, 70%, 80%, 90%, 100%) as well as five combinations of circulating water pumps are selected to find the optimal combination of circulating water pumps and main steam pressure under different unit loads at the lowest heat consumption rate.

3.3. Determination of feasible solution interval of system independent variables
Through the joint optimization model established above, the main steam pressure and exhaust steam vacuum of a 600 MW condensing unit are optimized jointly. The sliding main steam pressure determined by joint optimization is shown in table 1, table 2 and figure 2, figure 3.

**Table 1. Optimum Exhaust Pressure under Different Load and Environmental Temperature after Joint Optimization**

| Condensing Vacuum(KPa) | 300MW | 360 MW | 420 MW | 480 MW | 540 MW | 600 MW |
|------------------------|-------|--------|--------|--------|--------|--------|
| 5°C                    | 2.14  | 2.18   | 2.57   | 3.03   | 3.19   | 3.11   |
| 10°C                   | 2.74  | 2.77   | 3.21   | 3.71   | 3.46   | 3.71   |
| 15°C                   | 3.59  | 3.62   | 3.55   | 4.75   | 4.39   | 3.95   |
| 20°C                   | 4.73  | 4.76   | 4.65   | 5.17   | 4.87   | 5.03   |
| 25°C                   | 6.20  | 5.50   | 6.09   | 5.86   | 6.23   | 6.31   |
| 30°C                   | 8.06  | 7.18   | 7.90   | 7.62   | 7.48   | 8.27   |
Table 2. The Optimum Initial Pressure Result of Sliding Pressure of Unit at Ambient Temperature of 5 °C

| Name                  | Unit  | Value |
|-----------------------|-------|-------|
| Unit load             | MW    | 300   | 360  | 420  | 480  | 540  | 600  |
| Main steam pressure   | MPa   | 10.9  | 12.2 | 13.6 | 14.8 | 16.5 | 16.5 |

Figure 2. Sliding Pressure Curve of Unit at Ambient Temperature of 5 °C

Figure 3. Slip Pressure Curve Optimized Jointly at Ambient Temperature of 5 °C and 30 °C.
4. Conclusions

Through the above calculation, the sliding pressure curve under the optimal operation mode of the combined optimization circulating water pump is obtained, and the sliding pressure optimal initial pressure curve under the fixed operation mode of a circulating water pump is compared and analyzed. The results are shown in figure 4.

From the above analysis, it can be concluded that the economy of the joint optimization slip pressure curve is higher than that of the single optimization slip pressure curve. The reason is that when the sliding main steam pressure is optimized separately, the operation mode of circulating water selected in the whole load area is parallel operation of high-speed pump and low-speed pump, while the opening mode of circulating water pump optimized jointly changes from low load to high load, and the opening mode changes from a single high-speed pump to two low-speed parallel operation and then to a high-speed pump and a low-speed pump.

At the same time, according to the analysis in figure 4, the value of the sliding pressure main steam pressure of the combined optimization is lower than that of the single optimization, which shows that the combined optimization of the sliding pressure main steam pressure can improve the economy and has a positive effect on the safe operation of the unit. It can be seen that the joint optimization has a positive impact on improving the economic performance of units, and can play a guiding role in the sliding pressure operation of units with different loads and different seasons.

References

[1] Fan X, Qin J M, Fu C P 2011 Experimental study on optimal sliding pressure operation of 600 MW supercritical steam turbines Steam Turbine Technology 53 143-4+150
[2] Guo H G 2018 Parameter optimization of large nuclear power thermal system based on genetic algorithm Steam turbine technology 60 295-8
[3] Zhou L X, Hua M, Wang W 2011 Calculating method of off-design condition for unit initial parameters and heat consumption correction curve J. Power Engin. 31 387-90+396
[4] Zhang J 2018 Design of MRAC optimal control system for main steam pressure of 600 MW units Boiler Technol. 49 6-9
[5] Zhao N, Zhang W B, Wang H J 2009 An on-line calculation method for exhaust enthalpy and optimal initial pressure of steam turbines. *J. North China Electr. Power Univ.* 36 38-43+53

[6] Fang F, Zhang J X 2011 Modeling and Simulation of power plant thermal system based on system dynamics. *China J. Electr. Engin.* 31 96-103

[7] Liu J Z, Wang W 2011 Operation optimization of constant speed circulating water pump for thermal power units. *J. Power Engin.* 31 682-8

[8] Zhao B, Liu L 2007 Study on Optimization of cold end of steam turbine. *Therm. Turb.* 36 19-23

[9] Li X Y, Yan J J 2001 Study on evaluation index and diagnosis method of cold end system of thermal power plant. *Chinese J. Electr. Engin.* 21 94-8

[10] Wang W 2011 Modeling and energy-saving optimization of thermal power unit cold end system. *North China Electr. Power Univ. (Beijing, China)*

[11] Dong L J, Zhang C F 2007 Optimal dispatch of circulating water pumps. *Power Sci. Engin.* 23 73-8

[12] Huang X Y 2004 Discrete optimization model of unit circulating water system in thermal power plant and its application. *Therm. Power Engin.* 19 302-5

[13] You Q H 2012 Optimum design of cold end system for large coal-fired generating units. *North China Electric Power University (Beijing, China)*

[14] Zhang T T 2011 Study on exhaustive algorithm for unit commitment optimization. *J. Qingdao Univ. Technol.* 32 81-6

[15] Zheng T 2018 Main steam pressure slip curve optimization of 300 MW air-cooled units. *Electr. Power Autom.* 7 112-4