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Hybrid Modelling of a High-Pressure Control Valve of a Steam Turbine at Off-Design Modes Using First-Principle Mechanism and Operational Data

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Frequent and long-term operation at off-design modes of thermal power units is becoming a common phenomenon worldwide due to penetration of intermittent renewable power sources. The high-pressure control valve of a steam turbine is a key component for adjusting its power output, and its characteristics are of great significance for quantifying performances of a thermal cycle. However, it still remains a challenge to model and analyse the characteristics of a high-pressure control valve due to lack of post-valve measurement points. In this paper, we propose a hybrid modelling approach, based on first-principle mechanism and operational data, for characteristics of a high-pressure control valve operating at off-design modes. The flow characteristics of a high-pressure control valve are summarized as a function of flow coefficients. Results show that a characteristic function of a high-pressure control valve can be well used for modelling performances of a control stage operated at variable working conditions. This provides opportunities for improving efficiency of a thermal unit by better adjustment of the control stage of a steam turbine according to quantitative representation of its characteristics.

1. Introduction

Thermal power units are more frequently operated at part loads due to penetration of intermittent renewable power sources worldwide. For instance, average utilization hour of coal-fired power plants in China has dropped from 4,739 h in 2014 to 4,165 h in 2016 (Yu et al., 2018). Yet this trend is continuing and projected to continue for a certain time period due to continued development of renewable energy (Sun et al., 2018). This means more thermal power units are expected to undertake peak-shaving tasks, with extended cyclic load ranges (Rummel et al., 2018). However, deviation from design conditions could lead to increase in pollutant emissions on a per kWh basis, drop in thermal efficiency, and operational safety issues (Gu et al., 2016). Performance monitoring and optimization of thermal power units under off-design conditions provides opportunities to address these challenges.
A high-pressure control valve of a steam turbine is a key component for a thermal power unit pressure control and adjusting output. (Łukowicz et al., 2018). For steam turbines with a control stage, there are usually four or six high-pressure control valves, which are opened and closed in a certain order and combination, corresponding to the four or six sets of nozzles. Xu et al. (2014) studied partial opening characteristics of a high-pressure control valve of a 600 MW power unit and established a simplified mathematical model of control stage efficiency. Pondini et al. (2017) established a generic model of steam turbine control valve system and its actuation system, and studied response speed and efficiency of the control valve under different actuation systems, based on which design of a control stage can be optimized. These studies use mechanism models and require obtaining complete internal structural data of a control valve and a control stage, which is usually difficult to obtain. In addition, over a long-term operation period of a thermal power unit, its components may have aging problems, causing characteristics to deviate from their design values, and these changes are usually not represented in a mechanism model. However, an optimal operation strategy should be made based on real-time characteristics of a thermal power unit, and any deviation of

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characteristics could lead to a less optimal operation strategy if not properly accounted for in a decision-making procedure. For these reasons, real-life operational data of thermal power units should be used at the modelling stage. For high-pressure control valves of steam turbines, there are usually no post-valve measurement points, which make it difficult to model high-pressure valve characteristics.

A hybrid modelling method that combines first-principle mechanism with the available operational data is proposed for modelling characteristics of a high-pressure control valve in this paper. The paper is organized as follows. A problem statement and modelling methodology are presented in Section 2, followed by results and discussion of a case study in Section 3. Main conclusions are then summarized in Section 4.

2. Problem statement and modelling methodology

2.1 Steam flow analysis

In a steam turbine, main steam is firstly split and fed to control valves, usually four of them, and each control valve has a corresponding nozzle where main steam expands and accelerates. The main steam then enters a moving blade stage where part of enthalpy of the main steam is converted to mechanical work. Figure 1a shows an illustrative structure of a steam turbine control stage with four high-pressure control valves and measurement points. It should be noted that the four sets of nozzles are separated from each other, but the downstream moving blade stage is shared. Data in the dotted line box (red part) have no measurement points, including the post-valve pressure, the post-nozzle pressure and temperature. Data labelled in blue are measured.

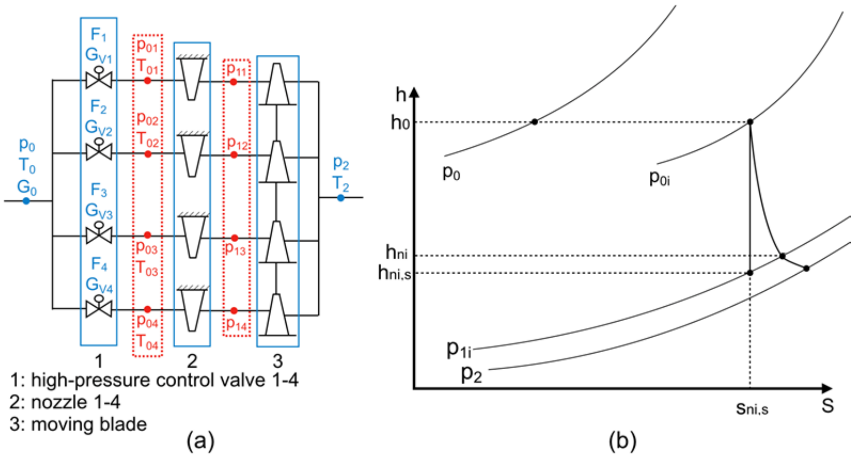

Figure 1: (a) An illustrative structure of a steam turbine control stage with four high-pressure control valves. (b) The thermal process of each steam stream flowing through the valve and control stage

It is not possible to model by separately analysing the high-pressure control valve due to the lack of post-valve data, which means that the corresponding nozzle and moving blade are required to be combined. However, the data behind the nozzle are also unmeasured as shown in Figure1a. Therefore, the conventional method of directly applying measurement data to model is no longer applicable here. The physical process of steam flow needs to be analysed and combined with existing measurement points to solve the missing data and then establish the model.

2.1.1 Ideal flow rate of the high-pressure control valve

Figure 1b shows the thermal process of each steam flowing through the valve and control stage. The flow of steam through the valve is a throttling process with equal enthalpy and pressure drop. It then expands in a down-stream nozzle, with a drop in pressure. Finally, it pushes a moving blade stage to do work, whilst its pressure continues to drop.

The ideal flow rate of the high-pressure control valve can be determined by Eq(1) (Li et al., 2008):

$$G_{Vi,ideal} = 0.648 \cdot \beta_i \cdot A_{Vi} \cdot \sqrt{\frac{p_0}{v_0}} \tag{1}$$

where p_0 is pressure in front of the high-pressure control valve, Pa. v_0 is specific volume in front of the high-pressure control valve, m^3/kg. A_{Vi} is opening area of the valve and determined by the fit diameter and opening degree, m^2.
Eq(2) defines pressure ratio of the valve, and Eq(3) defines coefficient β_i:

$$\varepsilon_{Vi} = \frac{p_{0i}}{p_0} \tag{2}$$

$$\beta_i = 1,\ 0 \le \varepsilon_{Vi} \le \varepsilon_{cr} \text{ or } \beta_i = \sqrt{1-\left(\frac{\varepsilon_{Vi}-\varepsilon_{cr}}{1-\varepsilon_{cr}}\right)^2},\ \varepsilon_{cr} < \varepsilon_{Vi} \le 1 \tag{3}$$

where p_{0i} is pressure after the high-pressure control valve i, Pa. ε_{cr} is critical pressure ratio of the valve.
The specific volume v_0 is obtained by the thermal properties of water vapour:

$$v_0 = f(p_0, T_0) \tag{4}$$

where T_0 is temperature in front of the high-pressure control valve, °C .
In summary, the ideal flow rate of a high-pressure control valve is a function of four variables:

$$G_{Vi,ideal} = f_V(A_{Vi}, p_0, p_{0i}, T_0) \tag{5}$$

2.1.2 Ideal flow rate of the nozzle

The outlet section of the nozzle is analysed. The ideal flow rate of the nozzle is determined by the nozzle outlet area, velocity and specific volume (Xiao et al., 2015):

$$G_{ni,ideal} = A_{ni} \cdot \frac{c_{ni}}{v_{ni}} \tag{6}$$

where A_{ni} is outlet area of nozzle i, m^2. c_{ni} is velocity of nozzle i at outlet, m/s. v_{ni} is specific volume of nozzle i at outlet, m^3/kg.
For a specific nozzle, the outlet area A_{ni} is fixed. Combined with the thermal process shown in Figure 1b, c_{ni} can be obtained from the conservation of energy:

$$c_{ni} = \sqrt{2(h_0 - h_{ni,s})} \tag{7}$$

where h_0 is enthalpy of the main steam, kJ/kg. $h_{ni,s}$ is ideal enthalpy of nozzle i at outlet, kJ/kg.
The h_0 , $h_{ni,s}$, v_{ni} are determined by the thermal properties of water vapour:

$$h_0, h_{ni,s}, v_{ni} = f(p_0, p_{0i}, p_{1i}, T_0) \tag{8}$$

In summary, the ideal flow rate of the nozzle is a function of five variables:

$$G_{ni,ideal} = f_n(A_{ni}, p_0, p_{0i}, p_{1i}, T_0) \tag{9}$$

2.2 Model hypothesis

Since the moving blade of the control stage is an impulsive blade, the reaction degree is small (Xiao et al., 2015), which means that the steam hardly expands in the moving blade and the pressure drop is small. Assume that the reaction degree of the moving blade is zero, that is:

$$p_{1i} = p_2 \tag{10}$$

where p_{1i} is pressure after nozzle i, Pa. p_2 is pressure after the control stage, Pa.

Note that p_2 is a conventional measurement point of power station. Based on the hypothesis of Eq(10), then combining Eq(5), Eq(9) and the mass conservation Eq(11), the post-valve pressure without measurement point can be solved.

$$G_{Vi,ideal} = G_{ni,ideal} \tag{11}$$

2.3 Model form

2.3.1 High-pressure control valve flow coefficient

The actual operating flow rate of the valve is deviated from the ideal, so the model cannot just be built the by the ideal flow rate calculation method. Define the flow coefficient of the high-pressure control valve:

$$\mu_i = \frac{G_{Vi}}{G_{Vi,ideal}} \tag{12}$$

where G_{Vi} is the actual flow rate of the valve i, t/h. The flow coefficient μ_i represents the deviation between the actual flow rate and the ideal flow rate, which is the key parameter reflecting the operating characteristics of the valve. The quantitative of it is the core part of the model.

2.3.2 First-principle mechanism analysis

As described in 2.3.1, the deviation between the actual flow rate and the ideal flow rate is reflected in the flow coefficient. The dominant variables that determine the flow coefficient need to be analysed according to the model hypothesis and the actual physical process. This is the first-principle mechanism analysis.

The reaction degree is one of the mechanisms of the flow coefficient since it is neglected according to the model hypothesis. But it cannot be directly calculated by the measured data. The reaction degree causes the post-control stage pressure to be lower than the post-nozzle pressure, which can be reflected in Eq(13) (Sun, 2015):

$$\varepsilon_0 = \frac{p_2}{p_0} \tag{13}$$

ε_0 is the overall pressure ratio of high-pressure control valve and control stage. It is one of the first-principle mechanisms that determines the operating characteristics of the valve.

Besides, the characteristics of each valve will change with the change of other valves (Xiao et al., 2015) due to the shared moving blade after each nozzle. So the effects of other valves need to be considered. Ma Lin et al. (2013) showed that the flow between the various high-pressure control valves interacted with each other. The inlet parameters of each valve are the same, and the flow is mainly determined by the pressure ratio of the valve according to Eq(1). Therefore, the pressure ratio of other opened valves are also the first-principle mechanisms that determines the operating characteristics of the valve.

In summary, let J_O represents the valve set that has been opened. For high-pressure control valve i, a functional relationship between the flow coefficient and the first-principle mechanisms is necessary:

$$\mu_i = \mathrm{f}_i(\varepsilon_0, \varepsilon_{Vj}),\ i \neq j,\ j \in J_O \tag{14}$$

3. Results and discussions

A 330 MW subcritical steam turbine with four high-pressure control valves is selected and modelled using its actual operating data for one week in this paper. In order to verify the model, the actual operating conditions of another time period are selected for simulation.

3.1 Modelling results

The high-pressure control valves of the modelling unit operate in such a way that valves 1 and 2 are first opened and always open at the same degree, then followed by valve 3. Valve 4 is finally opened. Besides, since the nozzles corresponding to the valves 1 and 2 have the same structure, valves 1 and 2 can be regarded as the same valve. The characteristic functions obtained by modelling are listed in Table 1.

Table 1: The characteristic functions of each high-pressure control valve

Valve	Characteristic function	R^2
1(2)	$\mu_1(\mu_2)=1.7283\cdot\varepsilon_0+0.0633$	0.9952
3	$\mu_3=3.845\cdot\varepsilon_0-3.017\cdot\varepsilon_{V1}(\varepsilon_{V2})+1.205$	0.9940
4	$\mu_4=35.803\cdot\varepsilon_0-1068.3\cdot\varepsilon_{V1}(\varepsilon_{V2})-182.18\cdot\varepsilon_{V3}+1213.4$	0.7268

R^2 is the goodness of fit of function. It can be seen from the results that the characteristic function of each valve has a significant linear correlation with the analysed first-principle mechanisms, which explains the rationality of this method to some extent. The linear correlation of valve 4 has decreased. The cause is when valve 4 is opened, all valves are opened, and there are more factors affecting the flow coefficient of valve 4, resulting in a decrease in the linear correlation of the valve 4 model. In addition, since the operating points of the valve 4 are small, the measured data for modelling is limited, which is also the cause of the decrease in the goodness of fit of the valve 4.

3.2 Simulation results

The characteristic function obtained by the model is used to simulate the high-pressure control valve flow rate under each working condition and compared with the measured value. The absolute values of the relative errors of the simulations of the four valves are 0.4 %, 0.4 %, 0.9 %, and 1.9 %. Valve 4 has fewer operating points and greater relative error, which corresponds to the modelling result of lower goodness of fit. But the relative error of most operating conditions remains within ±2.0 %, which is an acceptable range in actual engineering simulation.

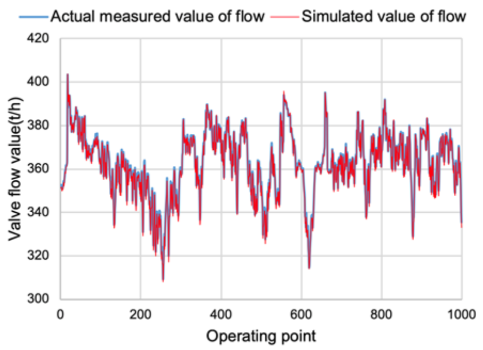

Figure 2: Simulation results and measured data of high-pressure control valve 1(2)

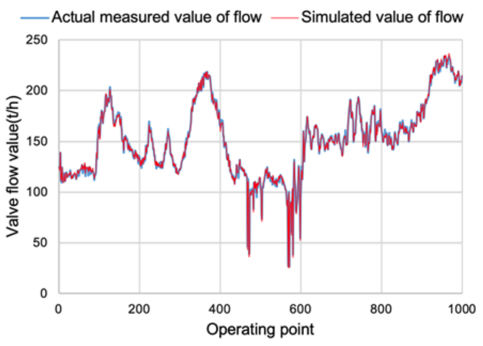

Figure 3: Simulation results and measured data of high-pressure control valve 3

In order to visualise the simulation results, the simulation results and measured values of valve 1 (valve 2), valve 3 and valve 4 are plotted in Figures 2, 3, and 4. Compared with valve 1(2) and valve 3, valve 4 has fewer operating conditions, which are only 150 points. The blue indicates operating measured data and the red indicates simulated values obtained by applying the model. As can be seen from the figures, the four valves have a wide range of operating conditions, especially valves 3 and 4. At the same time, the simulation results of each valve can well match the actual operation of the unit, which prove the accuracy and reliability of the model at off-design modes.

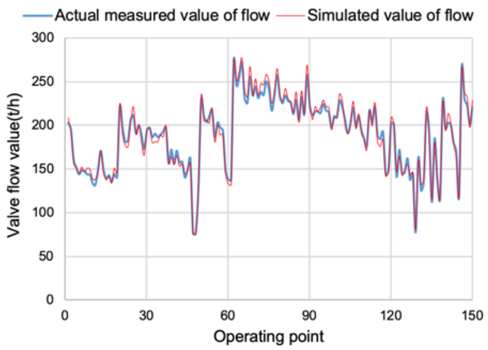

Figure 4: Simulation results and measured data of high-pressure control valve 4

4. Conclusions

The results show that the hybrid modelling method based on operational data and first-principle mechanism can be well used to model the high-pressure control valve of steam turbine, which solves the modelling difficulties due to lack of post-valve measurement points. The characteristic parameter of the model is the defined flow coefficient of the high-pressure control valve. The first-principle mechanisms are the overall pressure ratio of high-pressure control valve and control stage and the pressure ratio of other opened valves. The model can reflect the variable operating characteristics of the actual operation of the valve and is used to simulate the valve at off-design modes. The simulation results have high precision, and the average absolute value of relative error is less than 2 %, which proves the reliability of the modelling method and the accuracy of the model. It also shows that this method can be used for performance monitoring of steam turbine high-pressure control valves and lays a foundation for improving efficiency of a thermal unit.

Acknowledgments

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