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MODELING AND CONTROL SYSTEM DESIGN OF A MARINE CONDENSING SYSTEM

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ABSTRACT

Rapid and frequent load changes bring a number of challenges to the control system of a marine condenser. However, few studies have been published in this area. In this paper, a whole condensing system, which includes condenser, ejector, cooling water pump and its driving turbine, was modeled based on three conservation laws. Propulsion steam turbine was also modeled to simulate the load changes. A proportional integral (PI) controller was developed to regulate the condenser pressure. Opening signal of the governing valve of the propulsion turbine was added to the controller as a feed forward signal, and to improve the performance further, fuzzy algorithm was adopted to tune the gains of PI controller. Numerical experiments were conducted to study the dynamic behavior and the control performance of the condensing system. The simulation results show that employing the valve opening signal of main turbine as feed forward signal and tuning the gains of conventional PI controller by fuzzy logic are both effectual approaches to enhance the control performance. The former is good at reducing maximum overshooting while the latter is good at decreasing settling time. The combination of these two methods can improve the performance of simple PI controller further.

NOMENCLATURE

A	area(m^2)
B	discharge coefficient of valve ((kg·s)/(MPa·kg/m^3)$^{1/2}$),
C	specific heat capacity(kJ/(kg·K))
d	diameter(m)
f	hydraulic resistance coefficient (MPa/(kg/s)^2)
F	flow rate(kg/s)
h	enthalpy(kJ/kg)
J	moment of inertia(ton·m^2)
k	coefficient
K	heat transfer coefficient(kW/(m^2·K))
L	length(m)
m	molar mass(kg/mol)
M	mass(kg),moment of force(kN·m)
n	rotating speed(r/min)
p	pressure(MPa)
P	power(kW)
P_r	prandtl number
Q	heat flux(kW)
R	gas constant(J/kg·K)
R_e	reynolds number
t	time(s); temperature(℃)
T	Kelvin temperature(K)
V	volume(m^3)
W	mass ratio
Greek symbol	
β	coefficient
η	efficiency
λ	coefficient of thermal conductivity (kW/(m·K))

ρ	density(kg/m^3)
ω	angular speed(rad/s)
Subscript	
0	design value
a	air
c	condenser
cw	cooling water
ej	ejector
g	saturated gas
i	in, integral
l	saturated liquid; cooling
m	mixture; metal tube; motive steam
max	maximum
o	out
p	proportion
P	Pump; propeller
s	steam; suction
T	turbine
V	valve
Superscript	
*	normalized value

1 INTRODUCTION

Condenser is one of the most important devices in steam power system. It condenses the exhaust steam to liquid water, make the Rankine cycle accomplished.

Mathematical models are indispensable for studying the transient behavior and designing control system. Several investigations have been conducted to model the condenser. Zhang developed a relative simple model, and step responses to cooling water temperature and exhaust steam flow rate was conducted to study the dynamic characteristics of condenser[1]. A more complex model was carried out by Botsch et al, the transient behavior of the system was examined by five different experiments, each corresponding to a step change in one of the five key loads. The results were also validated by experimental data.[2] Evaluations were carried out on the effects of different closure correlations on the numerical simulation of condenser by Hu et al, and all of the numerical results were compared with the experimental data. Suggestions and recommendations were also given on the closure correlations based on the evaluations[3]. Malin established a mathematical model of condenser by using the semi-empirical correlation of Nusselt[4]. Cui adopting empirical correlation of Берман[5]. Comparison between the Nusselt correlation and the Берман correlation was also conducted by Cui, and the Берман correlation was recommended due to its better agreement with experimental data.

Little attention has been focused on the control system design of condenser. There are two reasons account for it. Firstly, load variation in land-based power plant is not frequent, so a continuous control of condenser pressure is unnecessary. Secondly, most marine systems are powered by diesel, which means there is no condenser. However, in large scale marine systems, steam power system is preferred since it can generate much more power per unit. And as the development of nuclear power, steam power has a more wide application in systems that demand for long-time persistence. Because of rapid and frequent load changes, condenser in a marine steam power system serves under more severe conditions. Thus control system design of marine condensing system has great potentials.

In this paper, a non-linear mathematical model for condenser is developed, and to control the condenser pressure, the main (propulsion) turbine, the cooling water pump and its driving turbine are also modeled. A conventional PI controller is designed for the condensing system, and to improve the control performance, fuzzy algorithm and feed forward loop are added.

The organization of the paper is as follows. Section 2 deals with the nonlinear modeling of the condensing system. In Section 3, we describe the control strategy for the system. Simulation results and discussion are provided in Section 4. Conclusions appear in Section 5.

2 MATHEMATICAL MODEL

Figure 1 illustrates a schematic diagram of the condensing system. Exhaust steam from the main turbine and the driving turbine enters the condenser and then condenses to water. The latent heat of exhaust steam is absorbed by cooling water which is pumped from sea by cooling water pump(CWP). The CWP is driven by a steam turbine, and the governing valve of the cooling water pump turbine(CWPT) is used to regulate the flow rate of cooling water through the condenser. To maintain a vacuum in condenser, a steam driven ejector is employed to extract the mixture of steam and air from condenser. When the mixture goes through the cooler of the ejector, steam condenses to water and then return to the condenser, while air, which can not condense, is discharged to the environment.

For a cooling water pump, the flow rate of cooling water has a minimum limitation. It is detrimental and not economical for the pump and its driving turbine to operate below this point. In this system, when the flow rate of cooling water drops near to the limit, the prime mover of pump will be switched from steam turbine to a motor, which is not concerned in this paper.

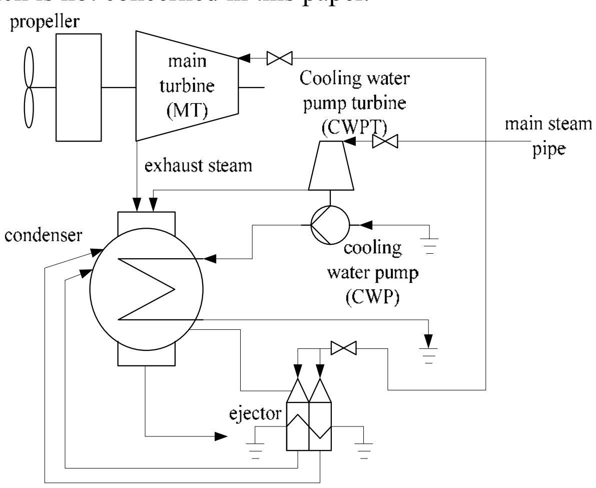

Figure 1. Schematic diagram of the marine condensing system

The condensing system has been modeled based on the mass, momentum and the energy conservations. To simplify the theoretical model, the pressure drops in steam tubes are neglected.

2.1 CONDENSER

A shell-and-tube condenser is modeled as shown in Fig. 2. There are two flow regions in this surface condenser: the shell side and the tube side. In the shell side, there are also two distinct zones: the gas zone and the liquid zone. Condensation takes place in the gas zone, and then condensate is collected at the bottom of the shell which is considered as the liquid zone.

The geometric volume of the liquid zone may vary with time but is very small compared to the volume of the gas. So, the liquid zone has little impact on the pressure of condenser. Hence the conservation equations of the liquid zone are omitted in this paper.

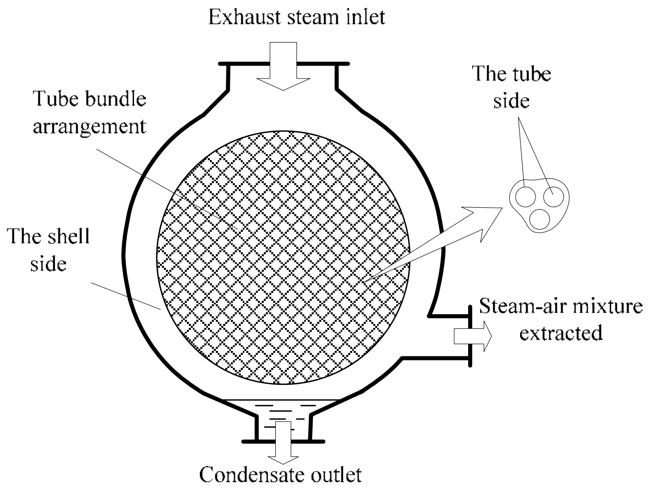

Figure 2. Schematic presentation of a shell-and-tube condenser

To simplify the condenser model, some assumptions are made as follows:

1) It is assumed that the gas is fully mixed in the gas zone.
2) The pressure drop in the shell side is neglected.
3) The air in the condenser is perfect gas.
4) The geometric volume of the gas is assumed to be constant

2.1.1 THE SHELL SIDE

As the liquid zone is not in consideration, only the gas zone is studied in the shell side. The gas zone is a mixture of steam, non-condensable gases and liquid condensate. The condensation pressure is considered as the sum of steam partial pressure and non-condensable gases partial pressure.

(1) Steam partial pressure

Applying the mass conservation equation to steam component, we have:

$$\frac{dM_s}{dt} = F_{si} - F_c - F_{ejs} \qquad (1)$$

the left of Eq.(1) can be written as:

$$\frac{dM_s}{dt} = \frac{d(\rho_s V)}{dt} = V\frac{d\rho_s}{dt} = V\frac{\partial \rho}{\partial p}\frac{dp_s}{dt} \qquad (2)$$

then the steam partial pressure can be obtained by:

$$\frac{dp_s}{dt} = \frac{F_{si} + F_c - F_{ejs}}{V\dfrac{\partial \rho}{\partial p}} \qquad (3)$$

1) Extracted steam

$$F_{ejs} = F_m\left(1 - W_a\right) \qquad (4)$$

where F_m is the flow rate of mixture extracted by ejector, and W_a is calculated by:

$$W_a = \frac{p_a m_a}{p_a m_a + p_s m_s} \qquad (5)$$

2) Condensate liquid

The flow rate of condensate liquid is computed from the heat transfer equation:

$$Q_c = K_c \Delta t_c A = F_c\left(h_{si} - h_{sl}\right) \qquad (6)$$

then we have:

$$F_c = \frac{K_c \Delta t_c A}{\left(h_{sg} - h_{sl}\right)} \qquad (7)$$

the temperature difference is given by:

$$\Delta t_c = \frac{t_{cw2} - t_{cw1}}{\ln\dfrac{t_s - t_{cw1}}{t_s - t_{cw2}}} \qquad (8)$$

The heat-transfer coefficient in the shell side has great influence on the heat transfer in the condenser[3]. The semi- empirical correlation of Nusselt is adopted in literatures[3-5]. Cui modeled the condenser employing an empirical correlation of Берман, and comparison between the Nusselt correlation and the Берман correlation was also conducted by Cui. Comparing the simulation results with experimental data, the Берман correlation was recommended due to its better agreement with experimental data. Therefore the Берман correlation is used in this study.

$$K_c = 4.07\beta_1\beta_2\beta_3\beta_4\beta_5\beta_6\beta_7\beta_8 \qquad (9)$$

where β s are the correction coefficients inclusion of material, clean condition of the surface, velocity and temperature of cooling water, arrangement of tube bundle, steam load and non-condensable gases. More details about the correction coefficients can be found in literature[5].

(2) Non-condensable gases partial pressure

The non-condensable gases, which inevitably leak into the condenser, are mainly air. Applying the mass conservation equation to the air component, we have:

$$\frac{dM_a}{dt} = F_{ai} - F_{ao} \qquad (10)$$

considering air as perfect gas, the equation of state for perfect gas can be written as:

$$10^6 \times V\frac{dp_a}{dt} + 10^6 \times p_a\frac{dV}{dt} = R_a T_a\frac{dM_a}{dt} + R_a M_a\frac{dT_a}{dt} \qquad (11)$$

compared with the change of pressure, the change of temperature is much slower. Thus neglecting the changes of temperature and volume, the air partial pressure can be obtained by:

$$\frac{dp_a}{dt} = \frac{R_a T_a}{10^6 \times V} \frac{dM_a}{dt} \qquad (12)$$

(3) Condensation pressure

The condensation pressure is the sum of steam partial pressure and air partial pressure.

$$p_c = p_s + p_a \qquad (13)$$

2.1.2 THE TURB SIDE

The tube-side fluid flow is a pipe flow and has been well studied. The tube wall energy balance is included in this side.

(1) Tube wall

During unsteady conditions, the heat accumulation in the metal tubes can not be neglected. Based on energy balance, it is derived as:

$$\frac{dT_m}{dt} = \frac{Q_c - Q_l}{M_m C_m} \qquad (14)$$

where Q_l is the heat absorbed by cooling water, which is computed from:

$$Q_l = K_l (T_m - T_{cw}) A \qquad (15)$$

Dittus-Boelter equation is one of the most widely used correlations for computing heat transfer coefficients in pipes, it is written as:

$$K_l = 0.023 R_e^{0.8} P_r^{0.4} \frac{\lambda_{cw}}{d} \qquad (16)$$

(2) Cooling water

The outlet temperature of cooling water is derived from energy balance equation:

$$M_{cw} C_{cw} \frac{d\left(\frac{T_{cwi} + T_{cwo}}{2}\right)}{dt} = F_{cw} C_{cw} (T_{cwi} - T_{cwo}) + Q_l \qquad (17)$$

neglecting the change of the inlet temperature of cooling water[5], we have:

$$\frac{dT_{cwo}}{dt} = 2 \times \frac{F_{cw} C_{cw} (T_{cwi} - T_{cwo}) + Q_l}{M_{cw} C_{cw}} \qquad (18)$$

2.2 STEAM TURBINE

The cooling water pump turbine and the main turbine share the same mathematical model, while they have different parameter values. The process in the turbine can be classified into three parts: the governing valve, the working converting part and the volume between them.

2.2.1 GOVERNING VALVE

The mass flow through the governing valve is given as:

$$F_V = B_{max} f\left(\frac{L}{L_{max}}\right) \sqrt{\rho \Delta p} \qquad (19)$$

where f is the valve characteristics function, representing the relationship between the flow cross area and the stroke of valve stem.

2.2.2 VOLUME

Applying the mass conservation equation between volume inlet and exit, the following equation is obtained:

$$V \frac{d\rho}{dt} = F_V - F_T \qquad (20)$$

The pressure in the volume is gained by:

$$\frac{dP}{dt} = \frac{F_V - F_T}{V \frac{\partial \rho}{\partial p}} \qquad (21)$$

2.2.3 WORK CONVERTING PART

Steam with high pressure and temperature passes through this part, and converts the thermal energy into kinetic energy to drive the rotor.

The mass flow rate through the work converting part is:

$$\frac{F_T}{F_0} = \frac{p}{p_0} \left(1 - k_1 \frac{\Delta n}{n_0}\right) \sqrt{\frac{T_0}{T}} \qquad (22)$$

The turbine generating power can be expressed as:

$$P_T = F_T (h_i - h_{os}) \eta \qquad (23)$$

where h_{os} is the flow enthalpy at the turbine exit for isentropic expansion.

η is the turbine efficiency which can be written as:

$$\frac{\eta}{\eta_0} = \left[k_2 - (k_2 - 1)\frac{n}{n_0}\right] \frac{n}{n_0} \qquad (24)$$

2.3 ROTOR

Ignoring the shaft friction, a torque balance equation for rotor can be expressed as:

$$J \frac{d\omega}{dt} = M_T - M_P \qquad (25)$$

The corresponding power balance equation is:

$$J\omega \frac{d\omega}{dt} = P_T - P_P \qquad (26)$$

where P_P is the propeller consumption power or the pump consumption power, which is calculated in the following steps respectively.

2.4 PROPELLER

The propeller characteristics and loss effects presented so far are all quasi-static.

The propeller consumption power is commonly given as[6]:

$$\frac{P_P}{P_{P0}} = k_3 \left(\frac{n}{n_0}\right)^3 \qquad (27)$$

k is a coefficient which can be calculated from the power-speed curve provided by the manufacturer.

2.5 COOLING WATER PUMP

The pressure rise of a pump is generally given in a normalized equation:

$$p^* = k_4 n^{*2} + k_5 n^* F^* + k_6 F^{*2} \qquad (28)$$

as well as the pump efficiency:

$$\eta_P = k_7 \left(\frac{F^*}{n^*}\right)^2 + k_8 \frac{F^*}{n^*} + k_9 \qquad (29)$$

Then the pump consumption power can be obtained by:

$$P_P = \frac{Fp \times 10^3}{\rho \eta_P} \qquad (30)$$

2.6 COOLING WATER PIPE

Consider the pipes that the cooling water passes through as one pipe with constant cross-section area, then apply the momentum conservation equation to this pipe, hence we have:

$$\frac{dF}{dt} = \frac{A}{L}\left[\left(p_1 + p - p_2 - fF^2\right) \times 10^6\right] \qquad (31)$$

where the term of fF^2 is the pressure drop caused by hydraulic friction.

2.7 EJECTOR

The air, which inevitably leaks into a condenser under vacuum, reduces the heat transfer efficiency, increases the condensation temperature and enhances corrosion reactions. So ejector is indispensable in a condensing system.

Since ejectors are widely used in a range of applications such as conditioning, aerospace, air removal of non-condensable gases and refrigeration, a lot of work has been done to investigate the performance of ejectors [7-9]. However, the models published care more about the internal characteristic, such as the pressure and velocity distribution, a lot of factors are taken into account. It makes the mathematical models too complicated for condensing system study. In this paper, an ejector model based on external characteristic is developed.

The external characteristic curves of a two-stage ejector are shown in Fig.3. The curves have two parts, the part with small slope is called working region, the other part with large slope is called overload region. Working in the overload region will cause a sharp increase of suction pressure. But it rarely occurs until the mixture flow rate increases very greatly. So only the working region of the characteristic curves is considered in this paper.

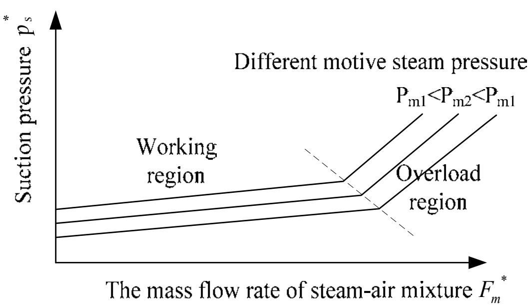

Figure 3. External characteristic curve of a two-stage ejector

The external characteristic of a two-stage ejector can be expressed as a series of straight lines with a same slope inclination and different intercepts, which can be written in a normalized form as:

$$p_s^* = k_{10} F_m^* + k_{11} \frac{p_{m0}}{p_m} + k_{12} \qquad (32)$$

There are two variables within only one equation, so another equation is needed:

$$p_c^* - p_s^* = k_{13} F_m^{*2} \qquad (33)$$

The equation above describes the flow of steam-air mixture in the pipe between the condenser and the suction chamber of ejector.

Figure 4 is a block diagram of the entire condensing system. The component of controller will be discussed in the following section.

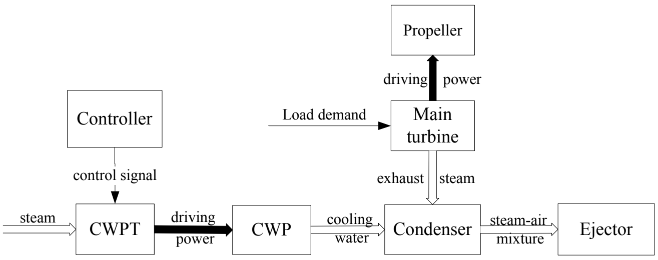

Figure 4. Block diagram of the marine condensing system

3 CONTROL STRATEGY

As higher or lower condenser pressure will threaten the safety, stability and economic operation of steam turbine and condenser, even the entire power system. Rapid and frequent load changes cause great variations of condenser pressure. An automatic control system is designed to regulate the pressure.

3.1 CONVENTIONAL PI CONTROLLER

The proportional integral derivative (PID) controller is one of the most widely used controllers in industrial processes. It has many advantages such as easy comprehension of its principle, simplicity in structure and robustness in operation [10]. PI controller is the most popular form of PID controller as it can achieve no steady state error with two gains. Thus, firstly, a simple PI controller is designed to regulate the condenser pressure. When the condenser pressure changes, the difference between current value and set value of condenser pressure is delivered into the PI controller, then control signal is generated by the PI algorithm to adjust the valve position of CWPT. The power change of CWPT leads to variation of cooling water flow rate, after the heat transfer process, the condenser pressure changes and returns to the set value eventually.

3.2 PI CONTROLLER WITH FEED FORWARD

The PI controller mentioned above is a feedback control system. It only begins to regulate when there is a change of desired value. A feed forward(FF) controller can be very beneficial in achieving a satisfactory performance in the rejection task. Since the adjustment of a feed forward controller starts as long as the disturbance occurs. The position signal of the governing valve of the propulsion turbine is used as feed forward signal to the simple PI controller.

Since steam turbine valve does not behave linearly, the position signal is not a reliable indicator of turbine load. Hence nonlinear correction is employed to transfer the nonlinear position signal to an approximately linear opening signal. The corrected opening signal shows an approximately linear relationship with the steam flow rate (Fig. 5). Because of the linear relationship between steam flow rate and turbine power, the corrected opening signal of governing valve can nearly indicate the turbine load.

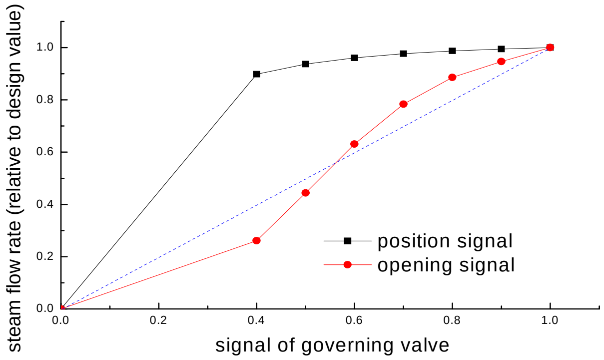

Figure 5. The nonlinear correction of position signal of governing valve

When the load demand varies, the position of governing valve of main turbine changes, then the corrected opening signal drives the CWPT to regulate the flow rate of cooling water before the condenser pressure changes.

3.3 FUZZY PI CONTROLLER

Since the PID controller does not perform as well in a non-linear system as it does in a linear system, great effect has been made in the past and more recently to enhance the PID control performance by tuning the PID gains while the system is in operation [10-13]. Fuzzy logic is a theory of logic which simulates human thinking by taking account of imprecision inherent in physical systems. Consequently, fuzzy logic could be used to control non-linear industrial processes [10]. Hence, a fuzzy PI controller is designed to regulate the condensing system.

The fuzzy algorithm uses the error and the change in error as its inputs, and the outputs are the corrections of PI controller gains. Hence, the gains turn out to be:

$$\begin{aligned} k_p &= k_{p0} + \Delta k_p \\ k_i &= k_{i0} + \Delta k_i \end{aligned} \quad (34)$$

Firstly, we should determine the membership functions for inputs(error and change in error) and outputs(kp and ki), which are shown in Fig.6 respectively. This step is called fuzzification.

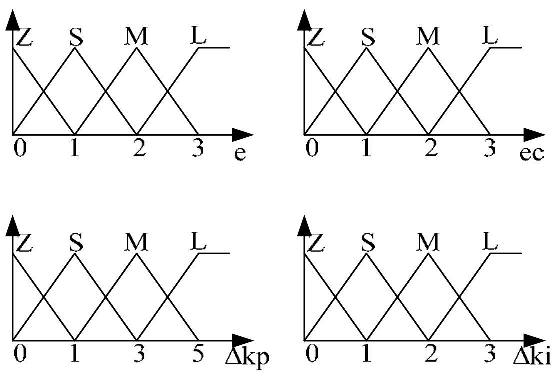

Figure 6. The membership functions for e, ec, kp, ki

Secondly, the fuzzy rule bases for Δk_p and Δk_i are constructed as shown in Tab. 1 and Tab. 2.

Table 1. Rule base for Δkp

Δk_p		e			
		Z	S	M	L
ec	Z	Z	S	M	L
	S	L	M	S	L
	M	M	M	S	M
	L	S	M	S	S

Table 2. Rule base for Δki

Δk_i		e			
		Z	S	M	L
ec	Z	Z	L	L	L
	S	L	L	Z	Z
	M	L	L	S	Z
	L	L	M	S	Z

Finally, the linguistic statements of Δk_p and Δk_i obtained in the second step should be defuzzified to get the outputs of fuzzy logic. In this step, "center-average" method is adopted[14].

3.4 FUZZY PI CONTROLLER WITH FEED FORWARD

To move a further step towards smaller overshooting and shorter settling time, fuzzy algorithm and feed forward are both included in the control system, as shown in Fig. 7.

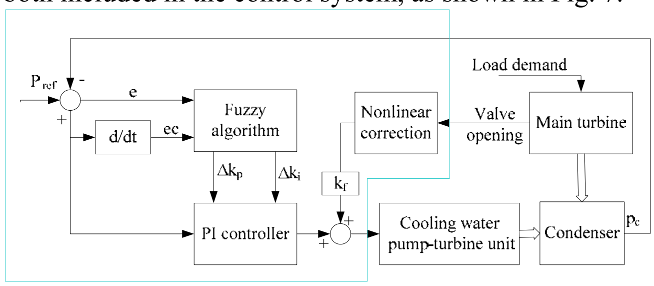

Figure 7. Block diagram of fuzzy PID controller with feed forward

4 RESULTS AND DISCUSSION

4.1 DYNAMIC CHARACTERISTIC

A decrease of load is simulated by shutting down the governing valve of main turbine from full load opening to 60% of full load opening. Figures 8-11 show the dynamic characteristics of the condensing system, and all the parameters are normalized by dividing corresponding values under full load condition.

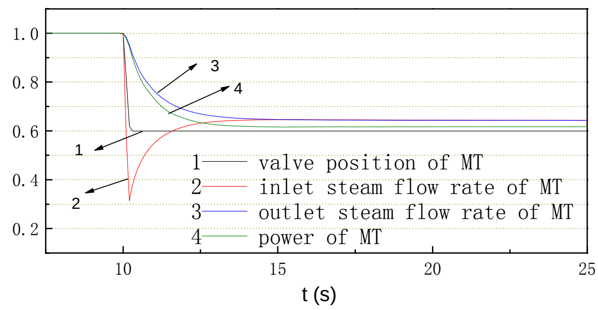

Figure 8. Dynamic characteristics of main turbine

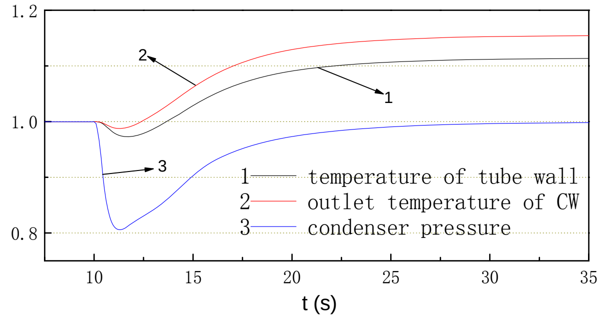

Figure 9. Dynamic characteristics of condenser

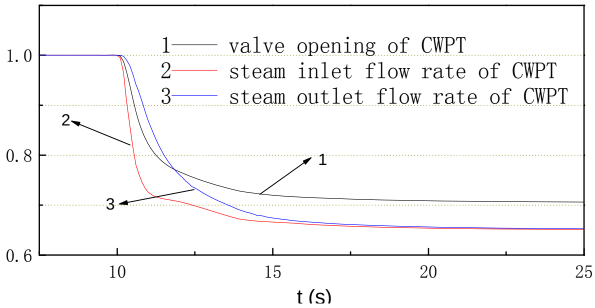

Figure 10. Dynamic characteristics of cooling water pump turbine

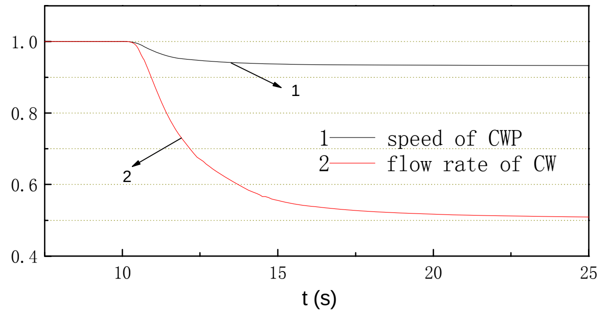

Figure 11. Dynamic characteristics of cooling water pump

Figure 8 shows the dynamic characteristics of main turbine. As the power demand decreases, the governing valve of main turbine is shut down quickly, which results in a reduction in the flow rate through the main turbine, so that the decrease of power generated by the turbine can meet the load change.

Since the exhaust steam flow rate of main turbine decreases, the condenser pressure begins to drop (Fig.9). Hence the control system starts to close the valve of CWPT, which causes a decrease in steam flow rate through the CWPT(Fig.10). Then the speed of turbine-pump unit runs down, so fewer cooling water is pumped to the condenser (Fig.11). Thus, the condenser pressure finally returns to the set point. As we can see in Fig.8 and Fig.11, the decrease of flow rate is larger in cooling water(about 50%) than that in exhaust steam of main turbine(about 65%), per unit cooling water absorbs more latent heat, so the outlet temperature of cooling water increases (Fig.9).

4.2 CONTROL SYSTEM PERFORMANCE

Because of the requirement of maneuverability for marine system, rapid load decreases or increases occur occasionally, which can be simulated by the step shut down or open up of the governing valve of main turbine. Numerical experiments of step shut down from full load opening to 85%, 75%, 65% and 50% of full load opening are conducted in this paper. Shut down experiments below 50% opening are not studied because of the minimum flow rate limitation of cooling water pump. All the four control systems mentioned above are simulated respectively.

The gains of PI controller are:k_p=0.25, k_i=0.06. The basic gains of fuzzy PI controller are the same with PI controller, but they are corrected by the fuzzy algorithm while the system is in operation. The gain of feed forward loop is k_f=0.27.

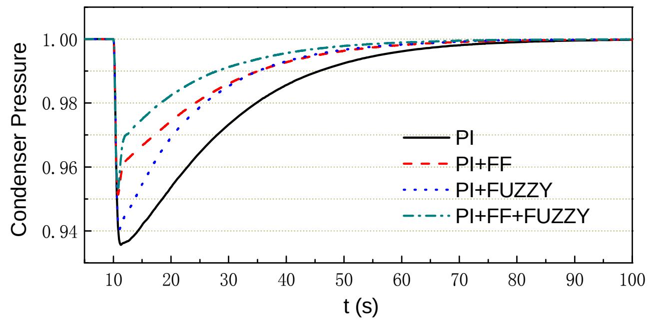

Figure 12. 85% down of the governing valve of main turbine

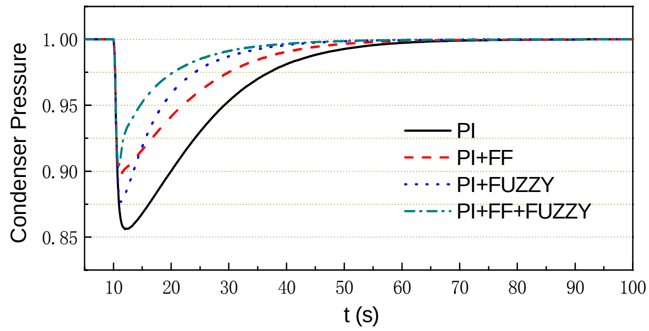

Figure 13. 75% down of the governing valve of main turbine

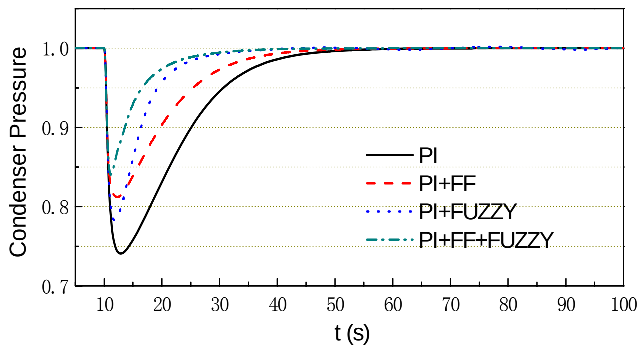

Figure 14. 65% down of the governing valve of main turbine

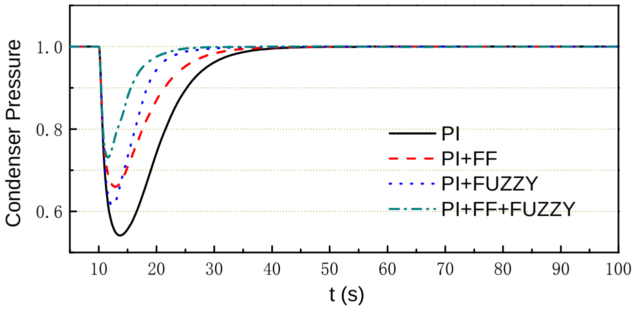

Figure 15. 50% down of the governing valve of main turbine

Table 3. Maximum overshooting $(p_{\min} - p_{\infty})/p_{\infty}$

	PI	PI+FF	PI+FUZZY	PI+FF+FUZZY
85%	−6.4%	−4.3%	−5.9%	−3.7%
75%	−14.4%	−9.8%	−11.8%	−7.1%
65%	−25.9%	−18.8%	−21.7%	−15.9%
50%	−45.9%	−34.0%	−38.6%	−26.9%

Table 4. Settling time $t_{|p-p_{\infty}|=0.02p_{\infty}}$

	PI	PI+FF	PI+FUZZY	PI+FF+FUZZY
85%	21.2	10.5	12.9	5.1
	100.0%	49.5%	60.8%	24.1%
75%	27.1	20	14.1	10.4
	100.0%	73.8%	52.0%	38.4%
65%	25.8	20.6	12.6	10.3
	100.0%	79.8%	48.8%	39.9%
50%	22.1	18.1	12.5	10
	100.0%	81.9%	56.6%	45.2%

Figures 12-15 show the responses of the condenser pressure. The results of step experiments for the four control systems demonstrate that the fuzzy PI controller with feed forward performs best, while the conventional PI controller is the worst. The maximum overshooting and 2% settling time of every experiment are presented in Tab. 3 and 4 respectively.

As we can see from Figs. 12-15 or Tabs. 3-4, both fuzzy algorithm and feed forward can reduce the maximum overshooting and the settling time. When the load change is more rapid(75%-50% down), feed forward path performs better in decreasing maximum overshooting while fuzzy algorithm makes greater improvement in shortening settling time. When the valve shuts down with a relative small degree (85% down of full load opening), adding a feed forward path is more effectual than adding a fuzzy algorithm, since feed forward path results in smaller maximum overshooting as well as shorter settling time.

5 CONCLUSIONS

A non-linear marine condensing system is modeled in this paper. The system dynamic behaviors demonstrate the validation of mathematical models. Four controllers are developed to control the condenser pressure. The results of load change experiments show that tuning the gains of conventional PI controller by fuzzy algorithm and employing the corrected valve opening signal of main turbine as the feed forward signal are both effectual approaches to improve the control performance. The addition of either fuzzy algorithm or feed forward path can both reduce the maximum overshooting and settling time. The feed forward path can make the system perform much better in maximum overshooting, while tuning the gains of PI controller by fuzzy algorithm makes greater improvement in settling time. The combination of them is the most effective method to enhance the control performance.

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