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Full-scale Mathematical Model and Simulation of Marine Natural Recirculation Drum-boiler

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Abstract

Through some reasonable hypothesis and simplification of conflagation, heat and mass transfer, a full-scale dynamic mathematic model of marine natural recirculation drum-boiler is established by modular modeling method via the basic working elements of objects. The dynamic feature of a certain type natural recirculation drum-boiler was studied by this model as an instance, by contrasting the simulative results and real values, the precision and real-time capability of the developed model was testified. This study was helpful for the performance research, working optimize and control strategy building of various natural recirculation drum-boilers.
Keywords: natural recirculation drum-boiler, simulation, mathematic model, dynamic feature of thermal-power system

1. Introduction

Drum-boiler is the important part of marine steam power plant[1]. It provide steam which possesses the needed pressure and temperature for every equipments in power plant, its working circs directly influence the safety, stability and economization. As a effective method that study the dynamic feature of object, the full-scale mathematical model of marine drum-boilers have basilic significance and applied value.

Scholars have conducted extensive research in the aspects of modeling and simulation of marine boiler. Out of mass conservation law, Astrom[2] Established a model for water level of drum boiler on the assumption that the mass share of steam of evaporation tubes is linear distribution. The operation result of this model was good agreement with experimental data in the trends, but the static simulation accuracy is not satisfied. Then Adam and Kim established the distrbuted parameter models for the evaporation tubes respectively used the homogeneous models of steam-water mixture[3] and vapor and liquid two-phase flow theory[4], the models were both described the flow of steam-water mixture in drum-boiler more detailed, and reflect the characteristics of boiler water recirculation more reasonable. Tian L established a centrain nonlinear general model[5,6] for drum-boiler through the analysis of emergy

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balabces that exist in the heat storage of boiler and steam pipes. Those moedls have simple form, and were successfully applied in the analysis of boiler control system and the study of boiler load-pressure characteristics, but they are not the full-scale model of objects, can not reflect the dynamic characteristics changes in the processes of boiler startup, boiler stop or load change widely

For a more comprehensive study of the boiler characteristics, scholars usually requires the dervations of the thermal process of boilers by mechanism modeling [7~9]. This paper adopted modular modeling method, and divided the object into several independent modulars by temporarily cut off the interrelations between every parts of marine natural recirculation drum-boiler. Then the models of every modulars were established according to the basic working principle of object. Finally, through the module bonding, the whole mathematic model of marine natural recirculation drum-boiler was structured.

2. Modeling process

2.1. module division

Out of the comprehensive consideration of volum, reliability and other aspects, most marine steam power plant adopt natural water recirculation boiler[10]. This paper adopt modular modeling method to study the modeling and simulation of marine natural water recirculation boiler, divided the whole boiler into six parts: drum, combustor, water wall, convective evaporation pipe bundles, superheater and decline pipes. Among them, combustor is the starting point of the mathematical moding of object, the heat flux of thermal evaporation and superheat system can be derived by calculate the gas temperature distributions outflow combustor, then the temperature and heat flux distribution of water wall, convective evaporation pipe bundles and superheater can be calculated, that provide the boundary conditions for the calculation of flow parameters and distribution. Drum modular was mainly used to calculate the dynamic changes of steam pressure, water level and other parameters. The relationships between each modules showen in Fig.1.

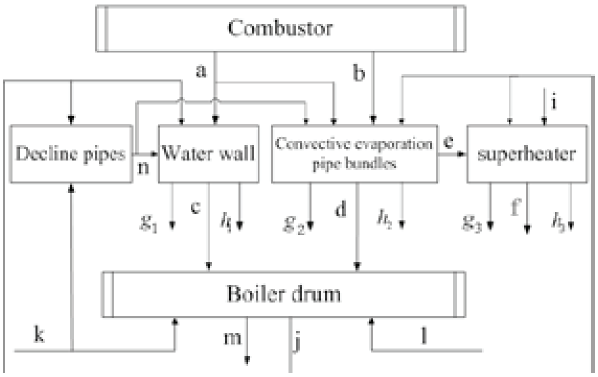


Fig.1. relationship of model modules of natural recirculation drum-boiler

In Fig.1: a refers to the outlet gas temperature of combustor, b refers to the flue gas themperature distribution at the enter of convective evaporation pipe bundles, c refers to the parameters of the steam-water mixture that outflow water wall, d refers to the parameters of the steam-water mixture that outflow convective evaporation pipe bundles, e refers to the flue gas themperature distribution at the enter of superheater, f refers to the parameters of superheated steam, $g_{1\sim3}$ respectively refer to the outer temperatures of water wall, convective evaporation pipe and superheater, $h_{1\sim3}$ respectively refer to the heat fluxs of water wall, convective evaporation pipe and superheater, i refers to the mass flow rate of superheated steam, j refers to the drum pressure, boiler water enthalpy and density, k refers to the mass flux and enthalpy of supply water, l refers to the outlet steam flow rate of drum, m refers to the boiler water level, n refers to the parameters of the water that outflow decline pipes.

2.2. modeling principles

Boiler plant is a nonlinear distributed parameter object which possesses typical characters such as pure lag, large inertia and so on, its working process is very complex. In this paper, the modeling assumptions are as follows:

(1) steam and water separate completely in drum
(2) refrigerant mixed uniformly, can be treat as lumped parameter
(3) steam and water uniformly mixed in water wall and convective evaporation pipe bundles, and ignore fluid inertia of the mixture
(4) do not consider the radiation heat transfer of convective evaporation pipe bundles and superheater, the radiation of combustor is completely absorbed by water wall
(5) ignores the axial heat conduction of tubes
(6) the circumferential heat of convective evaporation pipe bundles and superheater are uniform
(7) when the model involves the structure sizes of object, using the average
(8) treat combustor as a gray body that composed by gray wall and gray gas
(9) treat the combustion process as heating process, treat gas as ideal gas

3. Modular models

3.1. drum

In order to correctly reflect the dynamics of steam pressure, water level and temperature of drum, paper divided the internal volum of boiler into liquid and vapor two regions. Because the water storage capacity of marine drum-bouler is usually larger, and the water supply method usually adopts direct injection mode, so when boiler stable running, the boiler water in drum can be approximatly considered to be saturated[2,7,11].

(1) liquid region

The flows enter and exit the liquid region of drum are supply water flow W_{gsh}, decline pipes inlet flow W_d, the moisture content of drum inlet steam-water mixture flow $W_{sl}(1-\chi_{sl2})+W_{dl}(1-\chi_{dl2})$, dynamic evaporation and condensation flow W_{dv}, and boiler sewage flow W_{ex}.

According to assumption (2), established the mass conservation equation as follows:

$$dV_{bw}/dt = \upsilon_{bw}[W_{gsh} + W_{sl}(1-\chi_{sl2}) + W_{dl}(1-\chi_{dl2}) - W_d - W_{ex} - W_{dv}] \tag{1}$$

Where: V_{bw} is boiler water volume in drum, χ_{sl2} and χ_{dl2} respectively refer to the dryness of outflow steam-water mixture of water wall and convective evaporation pipe bundles, υ_{bw} is specific volume of saurated water under drum pressure p_b.

Only adopt formula (1) to calculate boiler water level may cause a larger errors, there are two main reasons: (a)when steam-water mixture flow through steam-water separator, there would be some steam was entrained into drum liquid region by water droplets, (b) the dynamic evaporation and condensation of boiler water is not only occurring in the vapor and liquid two-phase surface, but appeared in the entire liquid region, that make the liquid region expansion thereby bring the "falsehood water level". These two factors should be taken into account when calculating the actual volume of drum liquid region:

$$V_{bw}^* = V_{bw} + (W_{sl} + W_{dl})\chi_{bw}\upsilon_{bs}\Delta t_{bs}/\phi + W_{dv}(\upsilon_{bs} - \upsilon_{bw}) \tag{2}$$

Where: V_{bw}^* is the actual volume of drum liquid region, υ_{bs} is specific volume of saurated steam under drum pressure p_b, χ_{bw} is the mass ratio of steam entrained by water droplets, ϕ is boiler Recycling Ratio, Δt_{bs} is the average residence time that steam remain in liquid region, can be determined by experience or test.

When the accuracy is not critical, we can ignore the internal structure and external protrusions of drum, and approximate it as a lying cylinder, then the boiler water level can be calculated by the following formula:

$$l_b = r_b(1-\cos\alpha), \qquad \alpha = \begin{cases} \pi\left(4-\pi+\sqrt{(4-\pi)^2 + 32V_{bw}^*/L_b r_b^2}\right)/16, & V_{bw}^* \le \pi r_b^2 L_b/2 \\ \pi\left(12+\pi-\sqrt{\pi^2+24\pi+16+32V_{bw}^*/L_b r_b^2}\right)/16, & V_{bw}^* > \pi r_b^2 L_b/2 \end{cases} \tag{3}$$

Where: L_b and r_b respectively refer to the equivalent length and radius of drum, α is water lever angle.

(2) vapor region

The flows enter and exit the liquid region of drum are saturated steam flow W_{bs2}, superheater inlet steam flow W_{bgr}, drum inlet steam flow $W_{sl}\chi_{sl2} + W_{dl}\chi_{dl2}$, dynamic evaporation and condensation flow W_{dv}, and steam flow of discharge valve W_{exh}. According to mass conservation, we have the following formula:

$$(V_{b0} - V_{bw}^{*})K_{bps}\,\partial p_b \big/ \partial t = W_{dv} + W_{sl}\chi_{sl2} + W_{dl}\chi_{dl2} - W_{bs2} - W_{bgr} - W_{exh} \tag{4}$$

Where: $K_{bps} = \partial \rho_{bs} \big/ \partial p_b$ is the compress coefficient of saturated steam, can be determined by corresponding water and water vapor function, V_{b0} is the gross volume of drum.

Boiler's dynamic evaporation and condensation have a great relationship with the enthalpy of boiler water, can be calculated by the follow formula:

$$W_{dv} = K_{dv}(h_w - h_{bw}) \tag{5}$$

In fromula (5), the specific enthalpy of saturated water can be expressed as the single-valued function of drum pressure:

$$h_{bw} = h_{phw}(p_b) \approx h_{bw}^{'} + (p_b - p_b^{'})\, dh_{bw}^{'} \big/ dp_b^{'} \tag{6}$$

Where: h_w is the specific enthalpy of boiler water in drum, superscript ' refer to the previous time.

When boiler load changes, drum pressure changes is certainly faster than the enthalpy of boiler water. Therefore, we can thus believe that the specific enthalpy of boiler water h_w is the saturated water specific enthalpy under the pressure of the previous time, there are: $h_w = h_{bw}^{'}$. Then fromula (5) can be rewritten as:

$$W_{dv} = -K_{dv}(p_b - p_b^{'})\, dh_{bw}^{'} \big/ dp_b^{'} \tag{7}$$

Substituting fromula (7) into fromula (4), and processed by Euler method:

$$p_b = \Delta\tau\,(W_{sl}\chi_{sl2} + W_{dl}\chi_{dl2} - W_{bs2} - W_{bgr} - W_{exh}) \big/ \left[(V_{b0} - V_{bw}^{*})K_{bps} + K_{dv}\Delta\tau\, dh_{bw}^{'} \big/ dp_b^{'}\right] + p_b^{'} \tag{8}$$

Where: $\Delta\tau$ is calculation time step.

3.2. combustor

The combustors of marine boiler are extremely compact, the distribution characteristics of internal temperature field are not obvious when working, so the combustion flue gas can be treated as a lumped parameter object. According to assumption (9), established the energy conservation equation as follows:

$$0.5 m_g C_{pg}\, d\overline{T}_g \big/ dt = W_o H_o (1 - k_b) - Q_{bra} - (W_o + W_a)(\overline{T}_g - T_0) \tag{9}$$

Where: $\overline{T}_g$ is the average temperature of gas in combustor, W_o and W_a are the fuel and air feed flows of boiler, H_o is the chemical exergy of fuel, k_b is the thermal coefficient of boiler combustor, m_g and C_{pg} are gas mass and specific heat capacity at constant pressure.

According to assumption (4), radiant heat transfer rate can be calculated by the following formula:

$$Q_{bra} = \sigma A_{sl}(\overline{T}_g^{4} - \varepsilon_{sl} T_{sl1}^{4}) \tag{10}$$

Where: σ is boltzmann coefficient, A_{sl} and T_{sl1} respectively are the effective radiation area and average wall temperature of water wall, ε_{sl} is the grayscale of water wall, usually taken as constant.

Limited by testing technologies, the temperature distribution of outflow gas is difficult to obtain by experiment, if adopt theoretical calculations, the problem will be very complicated for the complexity of internal structure and combustion process. Therefore, paper adopted parabolic empirical formula[12] that commonly used in engineering when calculated the outflow gas temperature of combustor. Assuming that the gas temperature presented parabolic distribution along the width of flue, the highest and lowest temperatures were respectively appeared in the middle and both sides of the flue:

$$T_{gx} = \overline{T}_g + x_{yd}(1 - \varepsilon_x) \big/ 6 - \varepsilon_x x_{yd}^{2} + 1 \big/ 12 \tag{11}$$

Where: T_{gx} is the average gas temperature in the width of the flue, ε_x is horizontal flue gas temperature deviation coefficient, usually take the empirical value, $x_{yd} \in [0, 0.5]$ is the horizontal relative position between calculated point and the midpoint of the flue.

The changes of gas temperature on the height of the flue has some relationship with the arrangement of combustor shape and evaporative pipe bundles, but in general there is a decreasing trend with the increase of relative height, and it is essentially a linear relationship[11,13]. So in this paper, a linear model was adopted to approximately simulate the changes of gas temperature along the height of flue:

$$T_{gxy} = T_{gx}(1 + \varepsilon_y / 2 - \varepsilon_y y_{yd}) \tag{12}$$

Where: T_{gxy} is the flue gas temperature, ε_x is vertical flue gas temperature deviation coefficient, usually take the empirical value, $y_{yd} \in [0, 1]$ is the relative height of calculated point.

3.3. water wall

According to assumption (3)and(4), took the outlet parameters as lumped parameters, established the mass, momentum and energy conservation equation as follows:

$$V_{sl}\frac{d\upsilon_{sl}}{dt} = \upsilon_{sl}^2(W_{sl1} - W_{sl2}),\ \ \xi_{sl}\frac{W_{sl2}^2}{2} = (p_d - p_b)\upsilon_{sl} - \Delta H_{sl}g,\ \ V_{sl}\frac{dh_{sl}}{dt} = \upsilon_{sl}[(W_{sl1} + W_{sl2})\frac{h_{sl} - h_{wd}}{2} + Q_{sl}] \tag{13}$$

Where: $V_{sl}, \upsilon_{sl}, h_{sl}, Q_{sl}$ respectively are the volume, specific volume, specific enthalpy and heat adsoration of steam-water mixture in water wall, W_{sl1} and W_{sl2} are the fluid flow that inflow and outflow water wall, p_d and h_{wd} are the outlet pressure and enthalpy of decline pipes, ΔH_{sl} and ξ_{sl} are the relative height and pressure loss coefficient of water wall.

Pressure loss coefficient can be taken as constant when accuracy is not critical. Yet the heat adsoration of steam-water mixture can be calculated by the following empirical formula[14]:

$$Q_{sl} = K_{sl}A_{sl}p_{sl}^{4/3}(\overline{T_{slm2}} - T_{sl})^3 \tag{14}$$

Where: T_{sl} is the saurated temperature of steam-water mixture under drum pressure p_b, $\overline{T_{slm2}}$ and K_{sl} are the aversge temperature and convection heat transfer coefficient of the inner surface of water wall.

The dryness of the outflow steam of water wall can be calcultaed by the enthalpy of steam:

$$\chi_{sl2} = (h_{sl2} - h_{bw}) / (h_{bs} - h_{bw}) \tag{15}$$

Where: h_{bs} and h_{bw} are the specific enthalpy of saurated steam and water under drum pressure p_b.

The thermal storage equation can be derived by energy balance:

$$m_{sl}C_{msl}\, d\overline{T_{slm2}} / dt = Q_{bra} - Q_{sl} \tag{16}$$

Where: m_{sl} and C_{msl} are the mass and specific heat capacity of metal heat-transfer surface of water wall.

According to assumption (5), the temperature of water wall were two-dimensional distribution along the radial direction of pipe walls. Since the pipe wall of marine boiler is not thick, and the heat conductivity of metal surface is extremely fast, so we adopted the two-dimensional steady thermal conductivity formula, take fromula (14) and (16) as the boundary conditions to calculate the temperature distribution of pipe walls:

$$r^2\,\partial^2 T_{slm} / \partial r^2 + r\,\partial T_{slm} / \partial r + \partial^2 T_{slm} / \partial \varphi^2 = 0 \tag{17}$$

3.4. convective evaporation pipe bundles

According to the assumption (4) of this paper, there is only convective heat transfer between convective evaporation pipe bundles and gas, the heat flux of outer surface of pipe bundles is:

$$q_{dlxy} = \alpha_{dlxy}(T_{gxy} - T_{dlxy1}) \tag{18}$$

Where: α_{dlxy} is flue gas convective heat transfer coefficient at the ralative position $[x, y]$, T_{dlxy1} is the outer surface temperature in the appropriate position of pipe wall.

flue gas convective heat transfer coefficient can be calculated by the following formula[10]:

$$\alpha_{dlxy} = 0.2 C_s C_z Re_{xy}^{0.65} Pr_{xy}^{0.33} \lambda_{gas} / d_{dl} \tag{19}$$

Where: Re_{xy} and Pr_{xy} respectively are the Reynolds numbers and Prater numbers of the flue gas at the ralative position $[x, y]$, λ_{gas} is thermal conductivity coefficient of the flue gas, d_{dl} is the outer diameter of convective evaporation pipes, C_s and C_z respectively are the bundles arrangement correction coefficients and pipe row correction coefficients.

Because the fluid parameters of flue gas is quite difficult to calculate by theoretical formulas, so paper adopted avergae convective heat transfer coefficient to calculate the distribution of heat flux.

$$\alpha_{dlxy} \approx \overline{\alpha}_{dl} = \kappa_{dl} (W_o + W_a)^m \tag{20}$$

Where: κ_{dl} is the flue gas heat transfer coefficient of convective evaporation bundles, m is a coefficient[15], $m = 0.65$ for lateral washed and regularly arranged bundles, $m = 0.6$ for the lateral washed and staggered bundles, $m = 0.8$ for the lognitudinal washed bundles and large metal flat.

Dividing control bodies along the flow of fluid in convective evaporation pipe bundles, and took the outlet parameters of control bodies as lumped parameters, the mass, momentum and energy conservation equations of convective evaporation pipes were established as follows:

$$\frac{V_{dlxy} d\upsilon_{dlxy}}{dt} = \upsilon_{dlxy}^2 (W_{dlxy1} - W_{dlxy2}), \quad \frac{\xi_{dlxy} W_{dlxy2}^2}{2} = (p_{dlxy1} - p_{dlxy2}) \upsilon_{dlxy} - \Delta H_{dlxy} g, \quad \frac{V_{dlxy} dh_{dlxy2}}{dt} = \upsilon_{dlxy} [(W_{dlxy1} + W_{dlxy2}) \frac{h_{dlxy2} - h_{dlxy1}}{2} + Q_{dlxy}] \tag{21}$$

Where: $V_{dlxy}, \upsilon_{dlxy}, W_{dlxy}, h_{dlxy}, p_{dlxy}$ and Q_{dlxy} respectively are the volume, specific volume, mass flow, specific enthalpy, pressure and heat adsoration of steam-water mixture at the ralative position $[x, y]$, subscript 1 and 2 respectively refer to inflow and outflow, $\xi_{dl}, \Delta H_{dlxy}$ are the pressure loss coefficient and relative height of convective evaporation pipes at the ralative position $[x, y]$.

The calculation of heat adsoration Q_{dlxy}, dryness of outflow steam χ_{dl2} and inner surface temperature distribution T_{dlmxy2} are similar to water wall:

$$Q_{dlxy} = K_{dlxy} A_{dlxy} p_{dlxy2}^{4/3} (T_{dlmxy2} - T_{dlxy})^3, \quad \chi_{dl2} = (h_{dl2} - h_{bw}) / (h_{bs} - h_{bw}), \quad m_{dlxy} C_{mdl} \, dT_{dlmxy2} / dt = q_{dlxy} A_{dlxy} - Q_{dlxy} \tag{22}$$

According to assumption (5) and (6), we can take the inner surface temperature distribution and outer surface heat flux distribution as the boundary conditions to calculate the temperature distribution of pipe walls by one-dimensional thermal conductivity equation as follows:

$$\rho_{mdl} C_{mdl} \, r dT_{dlmxy} / dt = \lambda_{mdl} [dT_{dlmxy} / dt + r \, d^2 T_{dlmxy} / dt^2] \tag{23}$$

Where: $K_{dlxy}, T_{dlmxy2}, A_{dlxy}$ and m_{dlxy} respectively are the heat transfer coefficient, inner surface temperature, heat transfer area and metal wall mass of convective evaporation pipes at the ralative position $[x, y]$, T_{dlxy} is the saurated temperature of steam-water mixture under pressure p_{dlxy2}, ρ_{mdl}, C_{mdl} and λ_{mdl} are the density, specific heat capacity and heat transfer coefficient of pipe walls.

3.5. decline pipes

According to assumption (1), decline pipes is a single-phase flow region of drum-boiler, the outlet pressure of decline pipes can be calculated by the momentum conservation equation as follows:

$$p_d = p_b + \Delta H_{bw} g / \upsilon_{bw} - \xi_d W_d^2 / 2 \tag{24}$$

Taking the heat dissipation effect into account, the outlet water enthalpy of decline pipes is:

$$h_{wd} = h_{wjq}(1 - k_d) \tag{25}$$

Where: h_{wd} and h_{wjq} are the outlet water enthalpy of decline pipes and economizer, ξ_d, k_d and ΔH_{bw} are the pressure loss coefficient, thermal coefficient and elevation of decline pipes.

3.6. superheater

The working conditions of superheater is the worst in all heating surfaces of marine tube boiler. The practice has proved that the inhomogeneity of steam flow and differences of flue gas temperature between the bundles are main reason that cause the local overheat. By the analyzing the heat transfer and steam flow distribution, the distributed parameter model of superheater module has been getten as follows.

In trems of marine natural recirculation drum-boiler, there is usually arranged many superheaters along the flue gas flow direction. According to assumption (9), the flue gas temperature that enter the first stage superheater was:

$$T_{grgxy11} = T_{gxy} - q_{dlxy1} A_{dlxyi} \big/ [(W_o + W_a)C_{pg}] \tag{26}$$

The outlet gas temperature of every stage superheaters can be sequentially derived from formula(26):

$$T_{grgxyi2} = T_{grgxyi1} - q_{grxyi1} A_{grxyi} \big/ [(W_o + W_a)C_{pg}] \tag{27}$$

The convective heat transfer flux of superheater was:

$$q_{grxyi} = \kappa_{gri}(W_o + W_a)^{0.6}[(T_{grgxyi1} + T_{grgxyi2})\big/2 - T_{grxyi1}] \tag{28}$$

Where: T_{grgxyi} is the flue gas temperature distribution of superheater, subscript 1 and 2 respectively refer to inflow and outflow, A_{grxyi}, q_{grxyi} and T_{grxy1} respectively are the heat transfer area, heat transfer flux heat and outer surface temperature of superheater at the ralative position $[x, y]$, κ_{gri} is the flue gas heat transfer coefficient of superheater, C_{pg} is the specific heat capacity at constant pressure of flue gas.

Influenced by the superheater heat load, superheater structures and superheater inlet static pressure distribution, there is a deviation among the steam flow of each branch pipes. Assumed the outlet and inlet pressure of each pranch pipes is the same, the steam flow distribution can be calculated by the momentum conservation equation as follows:

$$W_{grij} = \sqrt{2[(p_b - p_{gr})\overline{\upsilon}_{grij} - \Delta H_{grij} g]\big/\xi_{grij}} \tag{29}$$

Where: W_{grij}, ξ_{grij}, $\overline{\upsilon}_{grij}$ and ΔH_{grij} respectively are steam flow, pressure loss coefficient, average specific volume and relative height of branch pipe, p_{gr} is the pressure of outlet superheated steam.

Dividing control bodies along the flow of fluid in superheater, and assumed the control body at ralative position $[x, y]$ located on the j branch pipe in the i superheater, the energy conservation equation of superheater was established as follows:

$$V_{grxy}\, dh_{grxy2}\big/dt = \overline{\upsilon}_{grij}\left[W_{grij}(h_{grxy2} - h_{grxy1}) + Q_{grxy}\right] \tag{30}$$

Formula (30) shows that the steam pressure in superheater approximately presents a linear variation along the fluid flow direction:

$$p_{grxy} = p_{gr} + (p_b - p_{gr})\Delta H_{grxy}\big/\Delta H_{gr} \tag{31}$$

Where: V_{grxy}, p_{grxy}, h_{grxy}, ΔH_{grxy} and Q_{dlxy} respectively are the volume, pressure, specific enthalpy, relative height and heat adsoration of superheater at ralative position $[x, y]$.

Heat adsoration of superheater and wall temperature distribution can be calculated as follows[17]:

$$Q_{grxy} = K_{grxy} W_{grxy}^{0.8}(T_{grmxy2} - T_{grxy}),\quad m_{grxy} C_{mgr}\frac{dT_{grmxy2}}{dt} = q_{grxyi} A_{grxy} - Q_{grxy},\quad \rho_{mgr} C_{mgr}\frac{dT_{grmxy}}{dt} = \frac{\lambda_{mgr}}{r}\frac{d}{dr}\left(r\frac{dT_{grmxy}}{dr}\right) \tag{32}$$

Where: K_{grxy}, A_{grxy}, m_{grxy}, T_{grmxy2} and T_{grxy} respectively are the heat transfer coefficient, heat transfer area, metal wall mass, inner surface temperature and superheated steam temperature, ρ_{mgr}, C_{mgr} and T_{grxy} are the density, specific heat capacity and heat transfer coefficient of pipe walls.

4. Experiment and result analysis

4.1. object description

Take a certain type marine natural ecirculation boiler as an instance, and adopted the modle in this paper to do the simulation research on its dynamic characteristics, the simulation program was writeen by Visual Fortran 6.5. The main parameters of this type boiler is shown in Table 1.

Tab. 1 main structure and capability parameters of simulative object

Parameter names	Parameter values	Parameter names	Parameter values
Superheated steam production	$80\ t/h$	Boiler water mass	5.5 tons
Saturated steam production	$5\ t/h$	Drum inner dimensions	Diameter 1.1 m， length 3.0 m
Superheated steam pressure	6.0 MPa	Water wall	ϕ 38×3 mm
Superheated steam temperature	450±10℃	Convective evaporation pipe bundles	9 rows in all: the 1st row ϕ 38×3mm, the 2nd ~ 9th rows ϕ 30×3mm, 1st~3th rows staggered, 4st~9th rows regularly arranged
Drum pressure	6.175 MPa		
Supply water temperature	104℃		
Startup time	30min	Decline pipes	4 rows in all: ϕ 38×3mm, staggered
Fuel consumption	$6.95\ t/h$		
gross volume of drum	2.89 m^3	Superheater	10 rows in all: ϕ 20×2.5mm, staggered
Excess air coefficient	5.9 m^3		

4.2. Results and Analysis

According to the parameters in Section 4.1, we carried out simulation experiment in the support of Minis simulation platform, the results were shown in Figure 2-5. The curves in figures are model output, and the marked points are actual measured values.

Fig.2 is the changes of drum pressure, superheated steam temperature, water temperature and water level over time during the process of boiler startup. Figure shows that, the initial statues were temperature 60℃, drum pressure 0.2 MPa , water level 220mm. Boiler water expansion after heated, water level began to rise, and water temperature began to rise at the same time. About one minute later, the heating process was completed, boiler water began to vaporize, drum pressure rise. About 5 minute later, drum pressure rose to 0.8 MPa , we increased the fuel injection and slightly opened the saturated and superheated globe valves to warm up auxiliaries. Then the water level began to decline, while the drum pressure, water temperature and superheated steam temperature continues to rise. The rising speed is much faster than before because fuel flow increased. About 6 minute later, boiler began to supply saturated steam, then water level continues to drop, while the rising trend of drum pressure and water temperature slowed down. About 14 minute later, drum pressure rose to 2 MPa and superheated steam temperature reached 360±10℃, boiler began to supply superheated steam and automatic adjustments of boiler worked at the same time. The whole startup process is about 30 minutes, that match the real performance of this type boiler.

Fig.3 and Fig.4 respectively are the response curves of drum pressure, water level, superheated steam temperature and water temperature when fuel regulative valve and when boiler load stepup 10% under operating conditions. Figures show that the simulation results and measured values agreed well with each other, it proved that the model in this paper has a satisfactory accuracy, can correctly reflect object's static and dynamic characteristics under normal working conditions.

One of the purposes of model-based simulation study is to conduct the fault simulation experiments in ensuring the safety and economy of experiments, and then get the more complete fault characteristics of objects by reproduced various fault symptoms and fault phenomenas, thus provides the experimental basis for fault diagnosis and treatment. Paper took the boiler exhausting valve leakage as an example, the fault simulation results is shown in Fig.5. Figure shows that the drum pressure, boiler water level and temperature were all decreased after failure occurred. Recause the decline of drum pressure, the steam flow througt the superheater will reduce, therefore the superheated steam temperature will rise, and then slowly drop and finally tends to stable along with the redistribution of steam flow. With te boiler load re-balancing, drum pressure and water temperature will stabilize at a new balance point. As for the water level, because it doesn't have the self-balancing capabilityes, will continue to plummet. The phenomenons that reflected in the curves of Fig.5 was consistent with the theoretical derivations, it proved the correctness of this model from another side.

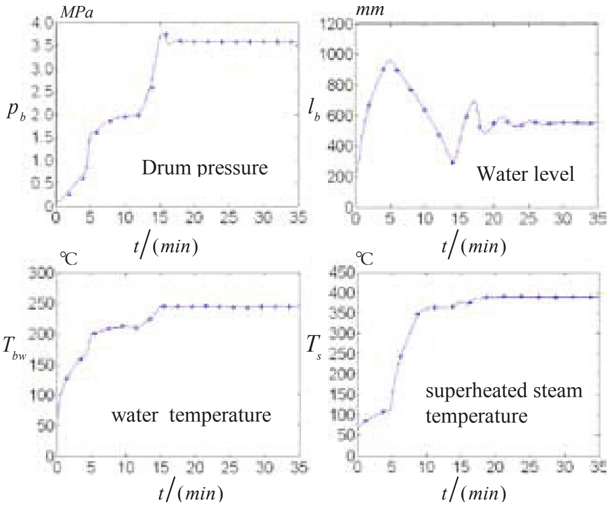


Fig. 2. curves of main parameters when boiler cold-startup

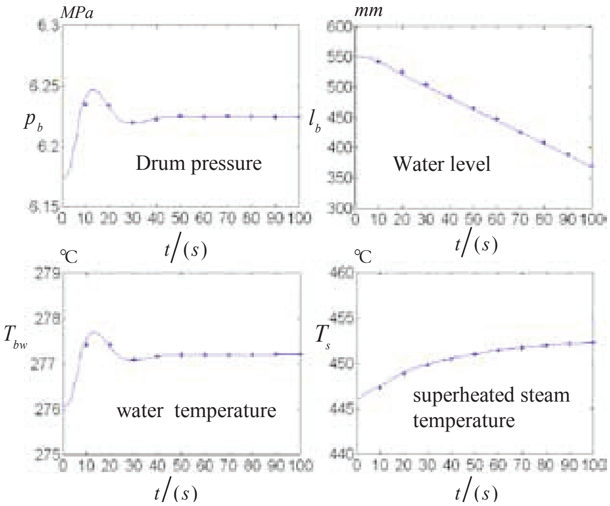
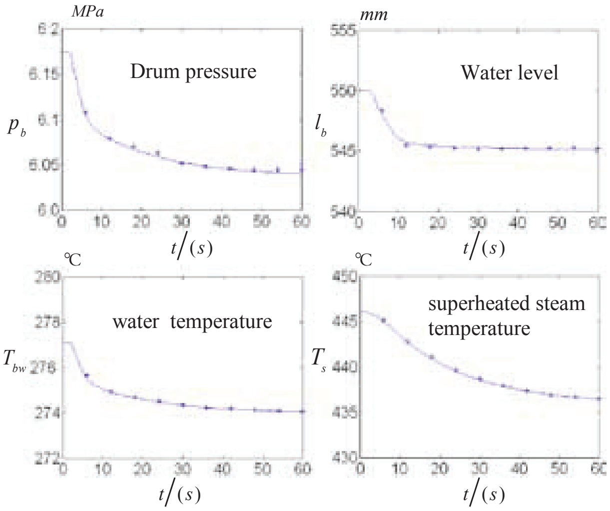


Fig. 3. the dynamic response of boiler when fuel stepup 10% under rating running situations



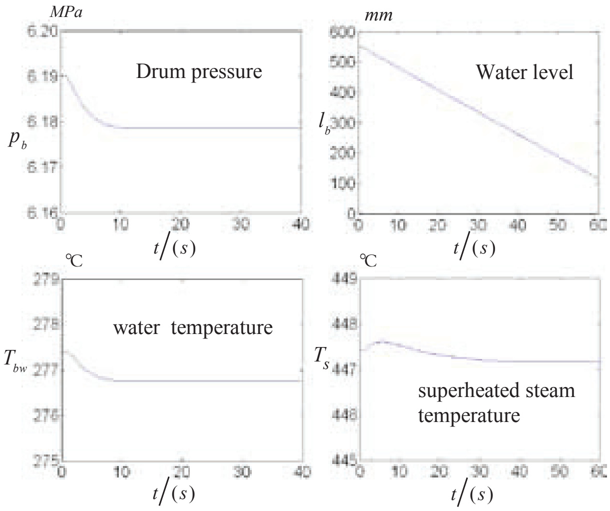


Fig. 4. the dynamic response of boiler when load stepup 10% under rating running situations

Fig. 5. the dynamic response of boiler when steam exhaust valve leak under rating running situations

5. Conclusion

From the perspective of improving the model accuracy and real-time capabilities, paper adopted modular modeling method to establish the mathematical model of marine natural recirculation drum-boiler. Paper focused on the modular decomposition of the marine boiler plant based on the analysis of the interactions between every components of object, then detailed expounding the construction of every module models, and validated the model with specific object. The model present in this paper is a kind of full-scale model for marine natural recirculation drum-boiler, that have a good simulation accuracy and real-time capabilities, can play some reference value on the performance analysis, operation optimization, fault diagnosis and control strategy development of various natural recirculation drum-boilers.

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