Gas Temperature Control of a Supersonic Heat-Airflow Simulated Test System

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ABSTRACT In order to accurately control the gas temperature of a supersonic heat-airflow simulated test system, this paper introduces the working principle of the system, including the fuel supply subsystem and the air supply subsystem, establishes the mathematical model of the system, including the fuel supply subsystem, the air supply subsystem and the gas temperature, and analyses the characteristics of the system. Based on characteristics such as non-linearity, parameter time variations, disturbance and pure time delay exist in the system; a new cascade control scheme that can control both the fuel supply subsystem and the air supply subsystem is proposed to achieve an accurate control of the gas temperature. On this basis, the simulation and experimental studies are carried out by the proposed control algorithm. The results show that the proposed gas temperature cascade control algorithm can achieve a fast, stable and no overshoot control of the gas temperature in a wide operating range, and the control accuracy can reach at ±5°C.

INDEX TERMS Temperature control, supersonic, cascade control, time delay, fuzzy sliding mode, fuzzy-PID.

I. INTRODUCTION

Supersonic heat-airflow simulated test system (SHSTS) is an important ground test support equipment in the field of aerospace. It is mainly used to simulate the high temperature and high heat flux density environment encountered by high-end aviation equipment such as high-speed aircraft and aero engine, so as to provide thermal performance test conditions for high-end equipment in the field of aerospace and ensure that the performance of the equipment meets the design requirements [1], [2]. Therefore, the SHSTS plays a very important role in the equipment development and performance test in the aerospace field [3]. It is of great significance to study its temperature control performance characteristics and improve its overall performance test level for improving the manufacturing level of important equipment in the national aerospace field.

At present, the research on the SHSTS mainly focuses on the structural design, flow field analysis and combustion mechanism research, and numerical simulation of the test system [4]–[6]. However, the theoretical studies on the centralized parameter modeling of the combustion process, fuel flow rate control, air flow rate control and gas temperature control, which are closely related to promote the temperature control performance of the test system, are not thorough and systematic. This situation leads to the performance of the test system cannot meet the current development requirements of aero engine and high-speed aircraft in China. Therefore, it is particularly urgent to study the gas temperature control methods of the test system to improve the overall performance level of the test system.

In terms of temperature control of the combustion system, the current researches mainly focus on utility boilers, gas turbines and aero-engines. In recent years, for the combustion system with large time delay, multivariable and frequent disturbance, the research of control strategy mainly focuses on three aspects. Firstly, traditional Proportion Integration Differentiation (PID) control combines with intelligent control to optimize the effect of PID control [7]–[10]; secondly, predictive control combines with other control algorithms to form a new predictive control algorithm [11]–[14]; thirdly, study the decoupling control algorithm to solve the strong coupling problem of the combustion control system [15]–[18]. However, the main steam pressure and combustion efficiency
are often more concerned in the control of the boiler combustion system, and the temperature is seldom directly controlled. In addition, the fuel of the boiler combustion system is mainly coal, which belongs to the combustion process of solid fuel. The object of this study is kerosene, that is to say, the combustion process of the SHSTS belongs to the combustion process of liquid fuel, which is different from that of the boiler combustion system. In the temperature control of gas turbines and aero engines, because the total temperature of the turbine inlet cannot be measured, the method of controlling the exhaust temperature of the turbine is basically used to control the inlet temperature indirectly, which is quite different from the method of establishing combustion model directly and controlling the gas temperature in this paper. In addition, the main goal of temperature control for gas turbines and aero engines is to make the temperature not exceed the maximum value set by operating conditions. The temperature control requirement of the SHSTS is to achieve a high quality of the gas temperature in a wide temperature range (300-1800°C), which requires higher control speed and accuracy. Therefore, the existing theory and method cannot be simply applied to the gas temperature control of the SHSTS.

In order to achieve an accurate gas temperature control of the SHSTS, the author realized the gas temperature control by adjusting the fuel flow rate with a fixed air flow rate in the previous study. However, it is found that the proposed gas temperature control method is very sensitive to changes in air flow rate, which affects the effect of the gas temperature control, and fuel rich combustion often occurs. In addition, the disturbance of fuel flow rate and air flow rate will affect the performance of the gas temperature control during the experiment. Because the cascade control has the advantages of improving the dynamic characteristics of the process, overcoming the system disturbance and improving the quality of the system control [19], so this paper puts forward a new cascade control strategy based on the original control strategy, that is, combining the original independent fuel supply subsystem and air supply subsystem into a cascade control system to realize the comprehensive control of fuel flow rate and air flow rate, eliminate the impact of air flow rate changes on gas temperature control, and improve gas temperature control performance. In addition, the mathematical model of the air supply subsystem is established, and according to the characteristics of the subsystem, a new sliding mode predictive control strategy is proposed.

II. SYSTEM DESCRIPTION

The structure and composition of The SHSTS can be found in Fig.1. It can be seen from the figure that it is a large complex system with multiple subsystems, such as fuel supply subsystem, air supply subsystem, exhaust subsystem, gas generator, specimen support device, and measurement control subsystem, etc. Aviation kerosene supplied by the fuel supply subsystem is combusted in the combustor of the gas generator to form central flame and annular flame to heat the high-speed airflow provided by the air supply subsystem, and produces a high-temperature gas at 300-1800°C acting on the specimen and supporting device, so the performance test of high-end equipment in aerospace field can be realized, for example, the thermal performance test of key components of supersonic aircraft and aero engine. The whole system is monitored and controlled by the measurement control subsystem. The measurement and control subsystem controls the normal operation of each subsystem according to the workflow and steps of the test system, monitors the working status of the system, and obtains important data of the system.

The fuel supply subsystem and the air supply subsystem are two important subsystems of the SHSTS. They provide fuel flow rate and air flow rate for the combustor of the test system to ensure normal combustion. Therefore, the gas temperature generated by the combustor mainly depends on the fuel flow rate and air flow rate into the system combustor. The fuel supply subsystem is a hydraulic system which combines frequency conversion hydraulic technology with proportional throttle valve control technology. It consists of frequency converters, three-phase asynchronous motors, plunger pumps, electro-hydraulic proportional control valves, flow rate sensors, etc. The air supply subsystem is a combination of hydraulic and pneumatic systems. The air pressure part is composed of an air compressor and a regulating valve, which is used to generate air flow meeting the pressure conditions and control the air flow rate into the combustor; the hydraulic part is mainly composed of a hydraulic cylinder and a servo valve, which is used to control the regulating valve on the air path.

The control subsystem is a two-stage computer control system based on industrial computer and Programmable Logic Controller (PLC). PLC is used to realize the field control of the fuel supply subsystem and air supply subsystem, and collect the important data of the system, such as gas temperature, fuel pressure, fuel flow rate, air flow rate, etc. Industrial computer has two main functions, one is to realize the remote control of the whole control system, to eliminate the harm of environmental noise to the staff, and the other is to compensate the lack of PLC’s computing power, to realize some complex control algorithms.

III. SYSTEM MODELING

A. FUEL SUPPLY SUBSYSTEM MODEL

According to the different fuel flow rate requirements of the combustor, the fuel supply subsystem has two working modes: pump control and valve control. The mathematical
model is different for different working modes, so the mathematical model of the fuel supply subsystem should be established under different working modes.

The fuel supply subsystem is mainly composed of frequency converters, three-phase asynchronous motors, gear pumps and pipelines in the pump control mode, their mathematical model can be obtained from [9].

\[ U_A = K_f K_{in} u \]  
\[ T_e = \frac{3m_p}{2\pi R_{pe}} K_f U_A - \frac{m_p^2}{40\pi R_{pe}} K_2 n_p = K_1 U_A - K_2 n_p \]  
\[ T_e = J_f \frac{\pi}{30} \frac{d\eta}{dt} + B_T \eta + D_p \eta_p \]  
\[ q_1 = \frac{1}{60} D_p \eta_p - C_p \eta_p \]  
\[ q_1 = \frac{p_p}{L_s + R_L + R_N} \]

where

\[ K_1 = \frac{3m_p}{2\pi R_{pe}} K_f; \quad K_2 = \frac{m_p^2}{40\pi R_{pe}}; \]
\[ L = \frac{4\rho l}{\pi d^2}; \quad R_L = \frac{128\mu l}{\pi d^4}; \quad R_N = \frac{\sqrt{2\rho_{r}p_s^2}}{6C_dA_0}. \]

The transfer function from fuel flow rate \( q_1 \) to control voltage \( u \) of the converter can be obtained by combining equation (1) to equation (5).

\[ G_{f1}(s) = \frac{q_1(s)}{u(s)} = \frac{b_0}{a_2 s^2 + a_1 s + a_0} \]  
\[ b_0 = \frac{1}{60} K_f K_{in} D_p; \quad a_2 = \frac{\pi}{30} J_f C_p L; \]
\[ a_1 = \frac{\pi}{30} J_T + K_3 L + \frac{\pi}{30} J_T C_p R + C_p L (K_2 + \frac{\pi}{30} B_T); \]
\[ a_0 = (1 + C_p R) (K_2 + \frac{\pi}{30} B_T) + K_3 R; \]
\[ K_3 = \frac{D_p^2}{120\pi \eta_m}; \quad R = R_L + R_N. \]

In the valve control mode, the output flow rate of the pump is adjusted to a certain constant, and the fuel flow rate into the combustor is controlled by controlling the opening of the proportional valve installed on the bypass branch fuel circuit. The transfer function from the fuel flow rate \( q_1 \) to the input voltage \( U_{pv} \) of the proportional amplifier can be obtained from [20].

\[ G_{f2}(s) = \frac{q_1(s)}{U_{pv}(s)} = - \frac{b_{v0}}{a_2 s^2 + a_1 s + a_0} \]  
\[ b_{v0} = K_q K_{pv} K_{v1}; \quad a_0 = K_c (R_1 + R_2) + 1; \]
\[ a_1 = T_{pv} K_c (R_1 + R_2) + T_{pv} + K_c L; \quad a_2 = T_{pv} K_c L; \]
\[ a_2 = C_{vd} W \sqrt{\frac{2P_v}{\rho f}}; \quad K_c = -C_{vd} W X_v \sqrt{\frac{1}{2\rho_f P_v}}. \]

The reason for the negative sign in equation (7) is as follows. In this control mode, the proportional valve is installed on the bypass branch fuel circuit, and the fuel flow rate into the combustor is equal to the output fuel flow rate of the proportional valve; in addition, in this mode, the output flow rate of the pump is a constant, so the term is zero when the transfer function is converted.

The comprehensive mathematical model under two working modes can be obtained by summing equation (6) and equation (7).

\[ q_f(s) = \frac{b_0 u(s)}{a_2 s^2 + a_1 s + a_0} - \frac{b_{v0} U_{pv}(s)}{a_2 s^2 + a_1 s + a_0} \]

Considering that the gear flowmeter used in the actual system has a pure time delay characteristic, the actual fuel flow rate \( q_{fd} \) obtained by the gear flowmeter lags behind the theoretical fuel flow rate. Assume that the pure time delay is \( \tau_s \), the following equation can be obtained.

\[ q_{fd}(s) = q_f(s) e^{-\tau_s} \]

B. AIR SUPPLY SUBSYSTEM MODEL

The air supply subsystem is a combined pneumatic and hydraulic system. Among them, the air flow rate regulating valve and a hydraulic servo system driving the regulating valve are used to control the air flow rate into the combustor. Therefore, the establishment of the mathematical model of the air flow rate regulating valve and the hydraulic servo system is the prerequisite for establishing the mathematical model of the entire air supply system.

1) MATHEMATICAL MODEL OF THE ELECTRO-HYDRAULIC SERVO VALVE

The servo amplifier is the electric signal conversion part of the electro-hydraulic servo valve, which converts the control voltage input by the control system into the drive current of the servo valve. Because the response time of the electrical system is much faster than that of the hydraulic system, a proportional link can be used to describe the mathematical model of the servo amplifier.

\[ I = K_{s1} u_s \]

where, \( u_s \) is servo amplifier’s input voltage; \( K_{s1} \) is amplifier’s gain; \( I \) is servo valve’s driving current.

Considering that the frequency width of servo valve is about 3-5 times of the working frequency of the system, when the mathematical model of the servo valve is established, it is approximated as a first-order inertia link, and so the following equation can be obtained.

\[ G_{a1}(s) = \frac{x_v(s)}{I(s)} = \frac{K_{s2}}{T_v s + 1} \]  
\[ where, \ x_v \ is \ servo \ valve’s \ displacement; \ K_{s2} \ is \ servo \ valve’s \ displacement \ driving \ coefficient; \ T_v \ is \ servo \ valve’s \ time \ constant. \]
2) MATHEMATICAL MODEL OF THE HYDRAULIC CYLINDER
Since the servo valve used in this hydraulic servo system is a zero-open four-way spool valve and the hydraulic cylinder used in this hydraulic servo system is a symmetric cylinder, the following equation can be obtained from [21, 22].

\[ Q_L = K_{ap}x_p - K_{plp} \] (12)
\[ Q_L = A_p x_p + \left( \frac{V_i}{4\beta_p} + C_I \right) p_{l} \] (13)
\[ A_p p_{l} = (M_l s^2 + B_p s + k_1 x_p) + F_L \] (14)

where, \( Q_L \) is the load flow rate; \( K_{ap} \) is the flow coefficient; \( K_{plp} \) is the flow pressure coefficient; \( p_{l} \) is the load pressure; \( A_p \) is hydraulic cylinder’s cross-section area; \( x_p \) is hydraulic cylinder piston’s displacement; \( V_i \) is cylinder’s volume; \( C_i \) is the total leakage coefficient; \( \beta_p \) is the bulk elastic modulus; \( M_l \) is cylinder’s mass; \( B_p \) is the viscous damping coefficient; \( k_1 \) is the spring stiffness; \( F_L \) is the external load force.

When only considering the inertial load of the system, the mathematical model of the hydraulic cylinder can be obtained from equation (12) to equation (14).

\[ G_{a2}(s) = \frac{x_p(s)}{x_i(s)} = \frac{K_{xc}}{s + \frac{2\alpha_a}{\omega_h} + \frac{2\alpha_c}{\omega_h} + 1} \] (15)

where, \( \omega_h = \sqrt{\frac{4\beta_p A_p^2}{M_l V_i}} \); \( \xi_h = \frac{K_{xc}}{A_p} \); \( \omega_c = \frac{K_{xc}}{C_I} \).

3) MODEL OF THE AIR FLOW RATE REGULATING VALVE
The air flow rate regulating valve is installed on the main air intake and is mainly used to control the air flow rate into the combustor of the system. Since the hydraulic cylinder directly drives the air flow rate regulating valve, the displacement of the air flow rate regulating valve is equal to the displacement of the hydraulic cylinder piston. The air flow through the valve port of the air flow rate regulating valve can be approximately regarded as the one-dimensional isentropic flow of ideal gas through the nozzle, so from the Sanville flow formula, the air mass flow rate into the combustor can be obtained as

\[ q_a = C_{d1} w p \sqrt{\frac{2}{R T} p_f \left( \frac{p}{p_s} \right)} \] (16)

where,

\[ f\left( \frac{p}{p_s} \right) = \begin{cases} \sqrt{\frac{k}{k-1} \left( \frac{p}{p_s} \right)^{\frac{k-1}{k}} - \left( \frac{p}{p_s} \right)^{\frac{k+1}{k}}} & 0.528 \leq \frac{p}{p_s} \leq 1 \\ \frac{2}{k+1} \sqrt{k} & 0 \leq \frac{p}{p_s} \leq 0.528 \end{cases} \]

\( C_{d1} \) is valve’s flow coefficient; \( w \) is valve’s opening area gradient; \( R \) is the air constant; \( T \) is gas’s absolute temperature of the throttle; \( p \) is the combustor pressure; \( p_s \) is the air source pressure; \( k \) is the isentropic index.

After the air enters the combustor of the system through the flow rate regulating valve and directly burns with the fuel entering the combustor, it can be considered that the system has no obvious load, that is, the pressure of the air source is much greater than the pressure in the combustor. Therefore, equation (16) can be simplified as follows

\[ q_2 = K_{33} x_p \] (17)

where \( K_{33} = \frac{1}{2} \frac{2}{R T + 1} \left( \frac{2}{k+1} \right)^{\frac{1}{k+1}} \). 

4) MODEL OF THE WHOLE AIR SUPPLY SUBSYSTEM
The mathematical model of the whole air supply system can be obtained by combining equation (10), equation (11), equation (15) and equation (17).

\[ G_{a}(s) = \frac{q_a(s)}{u_a(s)} = \frac{K_s}{s(T_s s + 1)(\frac{s^2}{\omega_h^2} + \frac{2\alpha_a}{\omega_h} s + 1)} \] (18)

where \( K_s = \frac{K_{a1} K_{c2} K_{a3} K_{m1}}{A_p} \).

In the control process of the actual air supply system, because the signal transmission, the response of the components and the measurement of the signal all require time, the air supply system inevitably has a pure delay characteristic. Given a delay time of \( t_1 \), the mathematical model of the air supply system can be expressed by the following equation.

\[ G_{a}(s) = \frac{K_s}{s(T_s s + 1)(\frac{s^2}{\omega_h^2} + \frac{2\alpha_a}{\omega_h} s + 1)} e^{-t_1 s} \] (19)

C. GAS TEMPERATURE MODEL

The combustor of the SHSTS can be regarded as a lumped parameter link from [9]. According to the law of conservation of energy and mass, the gas temperature model of the system can be obtained.

\[ V p V T + \rho_q q_a c_v T + K A_1 a T + \rho_{water} q_{water} c_{water} T \]

\[ = H p f q f + \rho_q q_a c_v T_{in} + K A_1 T_{in} + \rho_{water} q_{water} c_{water} T_{w0} \] (20)

where \( V \), \( \rho_p \), \( c_p \), \( T \) are combustor’s volume, gas density, gas specific heat capacity, and gas temperature; \( \rho_q \) is input air density; \( q_a \) is input air flow rate; \( K \) is combustor wall’s heat transfer coefficient; \( \alpha \) and \( \beta \) are the scale factors; \( A_1 \) is combustor wall’s heat transfer area; \( \rho_{water} \) is cooling water’s density; \( q_{water} \) is cooling water’s flow rate; \( c_{water} \) is cooling water’s specific heat capacity; \( H \) is the fuel calorific value; \( \rho_f \) is the fuel density; \( q_f \) is the fuel flow rate; \( T_{in} \) is input air temperature; \( T_{\infty} \) is the ambient temperature, \( T_{w0} \) is cooling water’s inlet temperature.

When the experimental conditions are fixed, the inlet air temperature \( T_{in} \), the initial temperature of the cooling water \( T_{w0} \) and the environment temperature \( T_{\infty} \) are basically unchanged, which can be regarded as constant.

IV. CONTROL STRATEGY

A. OVERALL CONTROL SCHEME

Because the gas temperature of the SHSTS depends not only on the fuel flow rate into the combustor, but also on the air flow rate into the combustor, and there are many factors such
as complex working conditions and disturbances, this paper presents a new cascade control scheme that can control both the fuel supply subsystem and the air supply subsystem, it is shown in Fig.2. The inner loop control system consists of a fuel flow rate control system and an air flow rate control system, which control the fuel flow rate and air flow rate respectively. The outer loop control system takes the gas temperature as the control target, the system combustor as the control object, and the temperature sensor as the feedback. In addition, in order to keep the combustion state of the combustor in the optimum state, i.e. high-efficiency and low-emission combustion of fuel, and to avoid the phenomenon of fuel rich combustion, a flow distributor is installed after the output of the gas temperature controller.

B. FUEL FLOW RATE CONTROL LAW
The actual fuel supply subsystem has some unfavorable factors, such as pure time delay, multi-interference and parameter time variations, which bring difficulties to the accurate control of the fuel flow rate. In the previous work, the author had done a lot of research work on the accurate control of the fuel flow rate, and had obtained a lot of research results and accumulated a lot of valuable experiences [1], [2], [20]. Therefore, on the issue of fuel flow rate control, this paper directly adopts a fuzzy-PID predictive control method proposed in the previous study. This method had been verified by a large number of experiments and is an effective fuel flow rate control method.

C. AIR FLOW RATE CONTROL LAW
1) CONTROLLER STRUCTURE AND PRINCIPLE
It can be seen from the modeling process of the air supply system that the mathematical model of the air supply system established in this paper not only has nonlinear characteristics, but also has a pure time delay link in the feedback channel. In addition, the operation of the entire system is complicated, and there is inevitably interference in the actual work. To overcome these disadvantages existing in the system, achieving high-quality control of air flow rate is not an easy task, and existing control algorithms cannot be directly applied. Therefore, based on fuzzy control and sliding mode control, combined with Smith predictor, a new type of fuzzy sliding mode predictive control algorithm is proposed, its structural schematic diagram is shown in Fig.4. It can be seen from the figure that its main principle is to use the Smith predictor to compensate the pure time delay problem in the system, and use fuzzy sliding mode control to overcome the instability and control performance degradation of the Smith predictor due to the model mismatch in the system. In addition, it can also be seen from the figure that the basic structure of the fuzzy sliding mode controller here is to use the sliding mode function as the input of the fuzzy controller and the output of the fuzzy controller as the control variable. The advantage of this kind of fuzzy sliding mode controller is to use fuzzy control to reduce the chattering of sliding mode control, and to reduce the dimensionality of the system and simplify the structure of the fuzzy controller by sliding mode control.
2) SMITH PREDICTOR

Smith predictor is an effective method to compensate the pure time delay problem of the control system, and it has been widely used since it was proposed. The basic principle of the conventional Smith predictor is to introduce a compensator in parallel with the controlled object to weaken and eliminate the pure time delay. It can be seen from the modeling of the air supply system that the pure time delay of the air supply system is on the feedback channel of the system. Therefore, this article slightly modifies the conventional Smith predictor and sets the compensator on the feedback channel of the controller to realize pure time delay compensation of the feedback channel; its structure and principle schematic diagram can be found in Fig.5.

![FIGURE 5. The structure and principle of Smith predictor.](image)

The transfer function from output \( Y \) to input \( R \) can be obtained by simplifying Fig.5:

\[
Y(s) = \frac{G_c(s)G_s(s)}{1 + G_c(s)G_m(s) + G_c(s)G_s(s)e^{-\tau s}} \tag{21}
\]

It can be seen from equation (21) that in order to compensate the effect of the pure time delay on the system, only the denominator of equation (21) does not contain the pure time delay link, that is to say, it is required to meet \( G_m(s) = G_s(s)(1 - e^{-\tau s}) \).

3) FUZZY SLIDING MODE CONTROLLER

The basic principle of the fuzzy sliding mode controller designed in this paper is to use the sliding mode function as the input of the fuzzy controller, and use the fuzzy control rules to adjust the control variables so that the condition \( \dot{s} < 0 \) is satisfied [23]. Suppose \( r(k) \) is the instruction input, \( y(k) \) is the system output, and \( e(k) = r(k) - y(k) \) is the error, the following sliding mode control surface can be obtained:

\[
s(k) = ce(k) + de(k) \tag{22}
\]

where, \( c \) is a constant and \( de(k) \) is the variation of error.

The input variables of the fuzzy controller are \( s(k) \) and \( ds(k) \), respectively, and the output variables \( \Delta U \) of the sliding mode controller are designed using two-dimensional fuzzy control rules. Let \( s \), \( \dot{s} \) and \( \Delta U \) denote fuzzification variables of input and output variables, respectively. When the input and output variables are fuzzified, the seven-level fuzzy sets [NB, NM, NS, ZO, PS, PM, PB] are used to represent the fuzzified variables, the domain is \([-3, -2, -1.0, 1, +2, +3]\), and the normal distribution function is the membership function.

The core of the fuzzy controller is to design the fuzzy control rules, because the fuzzy sliding mode controller in this paper must meet the sliding mode reachability conditions \( (\dot{s}s < 0) \). Therefore, this paper designed the fuzzy control rules under the sliding mode reachability conditions, and the results are shown in Table 1. It can be seen from Table 1 that when both \( s \) and \( \dot{s} \) are PB, then \( \dot{s}s \) is PB, and \( \Delta U \) is also PB at this time, so that \( \dot{s}s \) can be quickly reduced; when \( \dot{s}s < 0 \), this satisfies the sliding mode reachability condition, so \( \Delta U \) is ZO; when \( s \) and \( \dot{s} \) are NB, then \( \dot{s}s \) is PB, and \( \Delta U \) is NB at this time, so that \( \dot{s}s \) can also be quickly reduced. Since the fuzzy control rules in Table 1 are designed under the premise of meeting the sliding mode reachability conditions, the fuzzy sliding mode controller designed in this paper is stable.

| \( s \) | NB | NM | NS | ZO | PS | PM | PB |
|---|---|---|---|---|---|---|---|
| \( \dot{s} \) | PB | ZO | PS | PM | PB | PB | PB |
| | PM | NS | ZO | PS | PM | PB | PB |
| | PS | NM | NS | ZO | PS | PM | PB |
| | ZO | NB | NM | NS | ZO | PS | PM |
| | NS | NB | NB | NB | NM | NS | ZO |
| | NM | NB | NB | NB | NM | NS | ZO |

Because the output variable of fuzzy controller is fuzzy variable, it cannot be directly used in the control object, so it needs to be defuzzified. The center of gravity method described by the following equation is used to realize the defuzzification.

\[
uf(k) = \frac{\sum_{i=1}^{n} x_i \mu(i)}{\sum_{i=1}^{n} \mu(i)} \tag{23}
\]

D. GAS TEMPERATURE CONTROL LAW

In theory, the output of the temperature controller in Fig.2 is the sum of the fuel flow rate and air flow rate, and the flow rate distributor distributes the flow rate according to the optimal air-fuel ratio. In practice, the general work flow of the system is as follows: firstly, the working conditions should be determined, namely, the working temperature and Mach number (air flow rate) should be determined, then Mach number should be adjusted to reach the predetermined value, and finally the gas temperature will reach the preset value by adjusting the fuel flow rate. Therefore, in the actual control, the output of the temperature controller is the fuel flow rate,
and the flow rate distributor has two functions, one is to give the set value of the air flow rate, the other is to limit the fuel flow rate, so as to prevent the phenomenon of fuel rich combustion caused by the low air-fuel ratio. On the basis of guaranteeing high quality control of the fuel flow rate and air flow rate in inner loop, the main work of the temperature controller in outer loop is to determine the set value of the fuel flow rate according to the error between the set value and the actual value of the gas temperature, and the working process is relatively simple. Therefore, the conventional incremental PID is used as the controller of the gas temperature in the outer loop. The output of conventional incremental PID controller can be expressed as

\[ u(k) = \Delta u(k) + u(k-1) \]  
\[ \Delta u(k) = K_p[e(k) - e(k-1)] + K_i e(k) + K_d [e(k) - 2e(k-1) + e(k-2)] \]

where \(K_p\), \(K_i\), and \(K_d\) are proportional coefficient, integral coefficient and differential coefficient, respectively.

V. SIMULATION AND EXPERIMENT

A. SIMULATION

1) SIMULATION PARAMETERS

In order to verify the validity of the gas temperature control strategy of the SHSTS proposed in this paper, the simulation model was established by using Matlab. In order to simplify the simulation, only the mathematical model of the pump control mode was considered for the fuel supply subsystem. The model parameters of the fuel supply subsystem, air supply subsystem and gas temperature system were obtained by looking up the tables and calculating them, respectively, they are shown in Table 2, Table 3 and Table 4.

2) SIMULATION RESULTS

Based on the parameters listed in Table 2, Table 3 and Table 4, the control method proposed in this paper was simulated in Matlab. In order to prove the advancement and effectiveness of the method proposed in this paper, the control strategy proposed in this paper was compared with the sliding mode predictive control (SPDC) method proposed in our previous work (see [3]). In the simulation, 1000 °C was taken as the target gas temperature, and the simulation results under different air mass flow rate (different working conditions) were obtained, it is shown in Fig.6. From the Fig.6, it can be seen that the proposed gas temperature control algorithm can achieve a better control effect in the range of the air mass flow rate from 0.2 kg/s to 1.0 kg/s (basically including all the experimental conditions), that is, to achieve a rapid and non-overshoot control of the gas temperature. This shows that the control algorithm proposed in this paper can overcome the influence of working conditions (air mass flow rate changes) on gas temperature control, and get better gas temperature control results.
control performance in a larger operating range. It can also be seen from the simulation comparison results that the control method proposed in this paper is slightly better than SPDC. This is because when the air mass flow rate is 0.5 kg/s, we adjusted the control effect of the two methods to be almost the same, but when the air mass flow rate becomes 0.2 kg/s, the overshoot of the system when using SPDC is greater than that using the control algorithm proposed in this paper, and when the air mass flow rate becomes 1.0 kg/s, the system response speed when using SPDC is significantly slower than that using the control algorithm proposed in this paper.

The model parameters of the fuel supply subsystem will change due to the changes of motor parameters, temperature and load. In order to verify the ability of the designed controller to overcome the variation of parameters of the fuel supply subsystem, the simulation study was carried out using the proposed control method when the parameters of the fuel supply subsystem model changed while other parameters and control parameters remained unchanged. The gas temperature control curve was obtained under the condition that the parameter of the fuel supply subsystem model increases from 680 to 1200, while other parameters and control parameters remain unchanged, it is shown in Fig. 7. It can be seen from the figure that when the parameters of the fuel supply subsystem have changed, better control performance can still be obtained. That is to say, the control strategy proposed in this paper has the ability to overcome the parameters change of the fuel supply subsystem. In addition, it can also be seen from the simulation comparison results that the control algorithm proposed in this paper is significantly better than the SPDC method in the ability to overcome the inner loop interference of the system, when the interference occurs in the inner loop fuel supply system, the gas temperature fluctuation is relatively small and the recovery time is fast when the control algorithm proposed in this paper is adopted.

In addition to parameter time variations, there are also some disturbances caused by the flow field fluctuation in the combustor, the non-linear factors such as motor, pump and so on in the fuel supply subsystem. Fig. 8 is the simulation result under the condition that adding a voltage disturbance of 0.1 V to the fuel flow rate control subsystem in the 50s of the system response while the other parameters of the controller remain unchanged. It can be seen from the figure that adding disturbance to the fuel supply subsystem will cause the fluctuation of the gas temperature, but due to the role of the controller, in a short time, the system overcomes the influence of interference and restores to its original state. This shows that the control strategy designed in this paper can overcome the interference of the fuel supply subsystem. In addition, it can also be seen from the simulation comparison results that the control algorithm proposed in this paper is significantly better than the SPDC method in the ability to overcome the inner loop interference of the system, when the interference occurs in the inner loop fuel supply system, the gas temperature fluctuation is relatively small and the recovery time is fast when the control algorithm proposed in this paper is adopted.

In addition, the pure time delay of the fuel flow rate transfer function in the inner loop of the system may also change with different operating conditions. When the pure time delay of the fuel flow rate model increases from 3s to 4s and other simulation conditions remain unchanged, the simulation results are shown in Fig. 9. It can be seen from the figure that the
control performance of the system will be changed when the pure time delay of the fuel supply subsystem model increases, that is, the adjustment time of the system increases and the dynamic response of the system becomes worse. However, from the overall control effect, the system has no overshoot, and the response time can also meet the requirements. Therefore, the control method proposed in this paper has the ability to overcome the pure time delay of the fuel supply subsystem. At the same time, it can also be seen from the figure that although the response time of the system is slightly faster than the algorithm proposed in this paper when using SPDC, the system has a small overshoot. Overall, both methods can overcome the impact of pure delay uncertainty on the system, and the control effect can meet the requirements of use.

B. EXPERIMENTS

1) EXPERIMENT SETUPS

The experimental setups of the gas temperature control of the SHSTS are shown in Figure 10. From the figure, it can be seen that the experimental setups of the system mainly include a supersonic heat-airflow test platform and its automatic measurement control system. The supersonic heat-airflow test platform is the main equipment of the experiment, which provides the basic conditions for the experiment; the measurement control system is mainly used to realize the automatic control of the test platform. The measurement control system is a set of two-level computer control system based on PLC and industrial computer. Its composition, principle and function have been described in detail in the section 2, and will not be repeated here.

C. EXPERIMENTAL RESULTS

In order to verify the effectiveness of the control algorithm proposed in this paper, the gas temperature control of the SHSTS was studied experimentally on the basis of simulation. The gas temperature range of this test system is 900-1800°C, which belongs to the high temperature combustion system. In practice, in order to avoid burning down the equipment, the adjustment range of the gas temperature cannot be too large, usually 100-300°C. For this reason, in the experiment, firstly the gas temperature was stabilized at 1100°C, and then was set to 1400°C, and the step response curve of gas temperature from 1100°C to 1400°C can be obtained. The gas temperature response curve was carried out under the condition \( q_a = 0.5\text{kg/s} \), it is shown in Fig.11. It can be seen from the figure that the gas temperature raised steadily from 1100°C to 1400°C, without overshoot, and the rising time is about 20 seconds. This shows that the control algorithm proposed achieves a perfect control performance and meets the requirements for the gas temperature control. In addition, the step response experiment of the gas temperature drop from 1400°C to 1100°C was completed under the same working conditions and control conditions. The experimental result is shown in Fig.12. It can be seen from the figure that under the condition of temperature drop, the gas temperature drops steadily from 1400°C to 1100°C without overshoot, which also achieves a perfect control performance.

In addition, in order to verify the ability of the control algorithm to adapt to different working conditions, i.e. different air mass flow rate, the step response experiments of gas temperature from 1100°C to 1400°C were carried out under the conditions \( q_a = 0.3\text{kg/s} \) and \( q_a = 0.7\text{kg/s} \),
the control accuracy is namely, the rising process is fast, stable, no overshoot, and perfect control performance under three working conditions. It can be seen from the figure that, like the simulation results, the experimental results were compared on the same figure. The experimental results are shown in Fig. 13. respectively, and the experimental results were compared on the same figure. The experimental results are shown in Fig. 13. It can be seen from the figure that, like the simulation results, the control algorithm designed in this paper can achieve a perfect control performance under three working conditions, namely, the rising process is fast, stable, no overshoot, and the control accuracy is ±5°C.

VI. CONCLUSION

(1) The supersonic heat-airflow simulated test system has characteristics of non-linearity, parameter time variations, pure time delay and multiple disturbances, which have a negative impact on the accurate control of the gas temperature.

(2) The simulation results show that the control algorithm designed in this paper can achieve a fast, stable and no overshoot control of gas temperature in a wide range of operating conditions, and has the ability to overcome the parameter time variations, pure time delay and interference in the fuel supply subsystem. In addition, the control algorithm proposed in this paper is better than the SPDC method proposed in our previous work.

(3) The experimental results partially validate the simulation results, that is, the control algorithm designed in this paper can achieve a fast, stable and no overshoot control of the gas temperature in a wide operating range, and the control accuracy can reach at ±5°C.

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