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Robust Adaptive Fuzzy Control of Compressor Surge Using Backstepping

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Abstract Active control can expand operation range of compression systems. However, most of existing active controllers depend on known system characteristics and thus have limited engineering application. Robust control methods are used to alleviate the negative effects of uncertainties, but, with conservative results. In order to overcome the weakness of existing active controllers, a second-order MG surge model was presented using a close-coupled valve as actuator. Then, one active surge controller was designed based on backstepping control, in which, fuzzy system was used to model unknown system characteristics. Simulation results showed that the designed controller stabilized compression system outside the surge line, enlarge compressor's working range and additionally showed strong robustness against uncertainties such as unmodelled dynamics, flow and pressure disturbances.

Keywords Compressor · Surge · Active control · Fuzzy system · Adaptive robust control · Backstepping control

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الخلاصة

يمكنُ للتحكم النشط توسيع نطاق تشغيل أنظمة الضغط. ومع ذلك، فإن معظم وحدات التحكم النشط الحالية تعتمد على خصائص النظام المعروفة ، وبالتالي يكون تطبيق الهندسة محدودا. وتستخدم طرق المكافحة القوية للتخفيف من الآثار السلبية لعدم اليقين، ولكن ، مع نتائج متحفظة. ومن أجل التغلب على ضعف وحدات التحكم النشط القائمة تم تقديم نموذج تدفق مج من الدرجة الثانية باستخدام صمام وثيق مقترن كمشغل ميكانيكي. وتم بعد ذلك تصميم جهاز تحكم تدفق واحد نشط على أساس التحكم في التراجع الذي كان يستخدم نظاما غامضا في نمذجة خصائص النظام غير المعروفة. وأظهرت نتائج المحاكاة أن نظام ضغط وحدة تحكم المصمم المستقر خارج خط التدفق يزيد من نطاق عمل الضاغط ويظهر -إضافة إلى ذلك -متانة قوية ضد عدم اليقين مثل الديناميكيات غير المنمذجة، واضطرابات التدفق والضغط.

1 Introduction

Compressor is one of essential elements of gas turbine engine. Given the rotational speed, the maximum flow of compressor is defined by one valve of compression system or of engine. With the gradual decrease of flow, pneumatic instability occurred first inside the compressor, which is followed by surge and/or rotating stall. Surge is a kind of unstable flow regime with flow disruption occurring to the whole compression system which is composed of compressor, inlet/outlet pipes, downstream throttling device, etc. Surge is featured as axial oscillation of flow and pressure. Surge and rotating stall limit operation range of compressor reduce the system efficiency and might bring damage to the whole system, therefore should be avoided during compressor's operation.

Traditional anti-surge control method is passive, that is, consider surge margin requirement under the worst work conditions in the design phase of compressor, so as to avoid the compressor from operating in unstable status, but it costs the opportunity for compressor to work in high pressure ratio and high efficiency zone. Active surge control proposed by Epstein et al. [1] is to use feedback control to allow compressor to work in unstable zones where active control is not adopted, therefore letting the compressor operate at the high pressure ratio and high efficiency points close to surge line. That is a new idea for attacking the surge of compressor.

In the past decades, active surge control has seen significant progress in three aspects. Firstly, Moore and Greitzer together built the compressor's nonlinear state space model (MG model) [2,3], which successfully captured the nonlinear characteristics of compressor's instability and hence won wide application in active control. Secondly, by applying classic nonlinear dynamics, a simpler and easier active control method was raised based on bifurcation theory [4,5]. The third significant progress is the linear control method study accomplished by MIT [1,6].

As active instability control technology develops, many diverse actuators appear and the most representative are swing vanes, high pressure jet, loudspeaker, piston damping mechanism, close-coupled valve (CVV), throttling valve, etc [7]., among which high pressure jet and CCV are taken as better surge control actuators [8].

In most active control algorithms, it is assumed that system model is known. Unmodelled system dynamics and internal/external disturbance probably made active control not function correctly. Therefore, an active control method with all uncertainties considered becomes the new research hot point [9–11]. As a kind of universal approximator [12], fuzzy system is capable of fully utilizing human language information and is widely used to approximate the system's dynamics. Fuzzy system is applied to control system mainly in two ways. One is to approximate control law [13], while the other is to approximate the system's dynamics [12]. By taking CCV as actuator and using fuzzy system to approximate the compressor's dynamics, this paper has designed the robust adaptive active controller of compressor surge using backstepping method.

Experiment research is an important way to study compressor performance, but it is difficult to do this [14–19]. Because it requires high-level experiment facilities, has high cost and is difficult in test. Moreover, there exist risks in the research process of experiment. So, it is significant to do some theoretical research in the early phase of experiment research. It has been proved in reference [20] that MG model can accurately predict the occurrence of surge and rotating stall. So, in this paper, it has important basis to take research based on MG model.

2 Compressor Surge Dynamic Model

Moore–Greitzer's third-order model is expressed by the following three ordinary differential equations:

$$\dot{\Psi} = \frac{1}{4B^2 l_c}(\Phi - \Phi_T(\Psi)) \tag{1}$$

$$\dot{\Phi} = \frac{1}{l_c}\left(\Psi_C(\Phi) - \Psi - \frac{3H}{4}\left(\frac{\Phi}{W} - 1\right)J\right) \tag{2}$$

$$\dot{J} = J\left(1 - \left(\frac{\Phi}{W} -\right)^2 - \frac{J}{4}\right)\delta \tag{3}$$

Ψ is dimensionless pressure rise coefficient, Φ is dimensionless flow coefficient, J is the square of the rotating stall amplitude, Φ_T is throttle valve characteristics and Ψ_C is compressor steady-state characteristics. Differential of state quantity relative to time variable is $\xi = (U/R)t$, of which U is rotor tangential speed and R is the average radius of compressor. B, l_c and δ are constant numbers. The parameter B is defined as:

$$B = \frac{U}{2a}\sqrt{\frac{V_P}{A_C L_C}} \tag{4}$$

Here, a is local velocity of sound; V_P, A_C and L_C are, respectively, the volume of air collector, compressor flow area and the effective length of compressor pipe. Greitzer's study demonstrated that when the parameter B of compression system is more than a critical value B_{cr}, the instability type of compression system is presented as surge; then the parameter B is less than B_{cr}, the instability is presented as rotating stall. Compressor's steady-state characteristics can be approximated by following cubic curve.

$$\Psi_C(\Phi) = \psi_{CO} + H\left(1 + \frac{3}{2}\left(\frac{\Phi}{W} - 1\right) - \frac{1}{2}\left(\frac{\Phi}{W} - 1\right)^3\right) \tag{5}$$

ψ_{CO}, H and W are applicable constant numbers. Throttle characteristic curve is defined as:

$$\Phi_T(\Psi) = \gamma_T\sqrt{\Psi} \Leftrightarrow \Psi_T(\Phi) = \frac{1}{\gamma_T^2}\Phi^2 \tag{6}$$

In models (1)–(3), let $J = 0$ to get pure surge model.

$$\dot{\Psi} = \frac{1}{4B^2 l_c}(\Phi - \Phi_T(\Psi)) \tag{7}$$

$$\dot{\Phi} = \frac{1}{l_c}(\Psi_C(\Phi) - \Psi) \tag{8}$$

Like other physical systems, there is also disturbance existing inside compression system. Disturbance will obstruct the discovery of compression system instability and might cause active surge controller not to function normally. This paper is going to take the surge model of flow quantity and pressure disturbance as control object:

$$\dot{\Psi} = \frac{1}{4B^2 l_c}(\Phi - \Phi_T(\Psi) + d_\Phi(\xi)) \tag{9}$$

$$\dot{\Phi} = \frac{1}{l_c}(\Psi_C(\Phi) - \Psi + d_\Psi(\xi)) \tag{10}$$

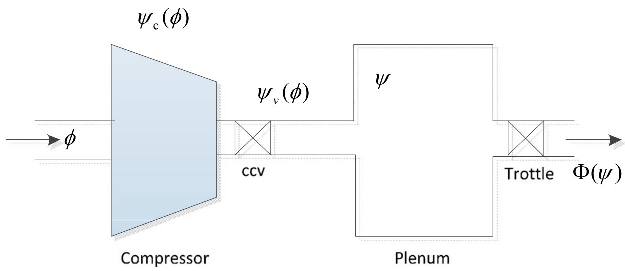


Fig. 1 Diagram of compression system with CCV

3 Actuator

Figure 1 is the diagram of compression system with CCV, in which the distance between CCV and compressor's outlet is small enough that air mass stored in between is negligible.

CCV's influence on the system can be expressed as $\Psi_{\mathrm{V}}(\Phi)$, the pressure drop from air flow passing CCV. The system's state space model is formulated as:

$$\dot{\Psi} = \frac{1}{4B^2 l_{\mathrm{c}}}(\Phi - \Phi_{\mathrm{T}}(\Psi) + d_{\Phi}(\xi)) \tag{11}$$

$$\dot{\Phi} = \frac{1}{l_{\mathrm{c}}}(\Psi_{\mathrm{C}}(\Phi) - \Psi_{\mathrm{V}}(\Phi) - \Psi + d_{\Psi}(\xi)) \tag{12}$$

Taking $\Psi_{\mathrm{V}}(\Phi)$ as system control quantity, it is obtained:

$$\dot{\Psi} = \frac{1}{4B^2 l_{\mathrm{c}}}(\Phi - \Phi_{\mathrm{T}}(\Psi) + d_{\Phi}(\xi)) \tag{13}$$

$$\dot{\Phi} = \frac{1}{l_{\mathrm{c}}}(\Psi_{\mathrm{C}}(\Phi) - u - \Psi + d_{\Psi}(\xi)) \tag{14}$$

4 Robust Adaptive Fuzzy Control on Surge

4.1 Description of Fuzzy System

A fuzzy system is the mapping from input vector to output vector: $\mathbf{x} \to y$, of which $\mathbf{x} = [x_1, \ldots, x_n] \in X_1 \times \cdots \times X_n \subseteq R^n$, $y \in R$. Considering fuzzy system has monolithic fuzzification, Gauss membership function, product inference and central average defuzzification [21], the ith rule of fuzzy logic system has the following form:

Rule i: if x_1 is F_{i1}, x_n is F_{in}, then $y = \omega_i$, where $i = 1, \ldots, m$, m is the number of fuzzy logic rules, ω_i is the statement of the ith fuzzy rule, $F_{ij} (j = 1, \ldots, n)$ represents a fuzzy set in universe of discourse X_i, use Gauss function as membership function:

$$\mu_{F_{ij}}(x_j) = \exp\left(-\left(\frac{x_j - a_{ij}}{b_{ij}}\right)^2\right) \tag{15}$$

where a_{ij} and b_{ij} are design parameters. The final fuzzy system is obtained by integrating individual fuzzy rule, that is,

$$y(\mathbf{x}) = \frac{\sum_{i=1}^{m} \omega_i \left(\prod_{j=1}^{n} \mu_{F_{ij}}(x_j)\right)}{\sum_{i=1}^{m} \left(\prod_{j=1}^{n} \mu_{F_{ij}}(x_j)\right)} \tag{16}$$

Let:

$$\mathbf{W}^T = [\omega_1, \ldots, \omega_m]$$
$$P(\mathbf{x}) = [p_1(\mathbf{x}), p_2(\mathbf{x}), \ldots, p_m(\mathbf{x})]^T$$
$$p_i(\mathbf{x}) = \frac{\prod_{j=1}^{n} \mu_{F_{ij}}(x_j)}{\sum_{i=1}^{m} \left(\prod_{j=1}^{n} \mu_{F_{ij}}(x_j)\right)}$$

Set a fixed value for membership function, which means let a_{ij} and b_{ij} as constant values and defines statement of fuzzy rule ω_i as tunable parameter, then fuzzy system can be formulated as:

$$y(\mathbf{x}) = \mathbf{W}^T \mathbf{P}(\mathbf{x}) \tag{17}$$

where $\mathbf{P}(\mathbf{x})$ is called fuzzy basis function vector, while $\mathbf{W}$ is called parameter vector.

For any real continuous function $y(\mathbf{x})$, in the set $\mathbf{X} \subseteq R^n$ as well as any real number $\varepsilon > 0$, there is a fuzzy system $y^*(\mathbf{x})$ in the form of Formula (17), satisfying $\sup|y^*(\mathbf{x}) - y(\mathbf{x})| < \varepsilon$ [22]. Therefore, adopting fuzzy system to approximate one continuous function $y(\mathbf{x})$ is expressed as:

$$f(\mathbf{x}) = \mathbf{W}^{*T} \mathbf{P}(\mathbf{x}) + \Delta f(\mathbf{x}) \tag{18}$$

where $\Delta f(\mathbf{x})$ satisfies $\Delta f(\mathbf{x}) < \varepsilon$.

4.2 Controller Design

For System (13) and (14), assuming compressor characteristic curve Ψ_{C} is unknown, to adopt backstepping method for controller design, coordinate transformation shall be made first:

$$\begin{cases} e_1 = \Psi - \Psi_{\mathrm{d}} \\ e_2 = \Phi - \alpha \end{cases} \tag{19}$$

Ψ_{d} is the given value of pressure coefficient. Assuming its first-order derivative exists and is bounded, α is virtual control law.

Step 1: design virtual control law for the first subsystem.

$$\begin{aligned} \dot{e}_1 &= \dot{\Psi} - \dot{\Psi}_{\mathrm{d}} \\ &= \frac{1}{4B^2 l_{\mathrm{c}}}(\Phi - \Phi_{\mathrm{T}}(\Psi) + d_{\Phi}(\xi)) - \dot{\Psi}_{\mathrm{d}} \\ &= \frac{1}{4B^2 l_{\mathrm{c}}}(e_2 + \alpha - \Phi_{\mathrm{T}}(\Psi) + d_{\Phi}(\xi)) - \dot{\Psi}_{\mathrm{d}} \end{aligned} \tag{20}$$

Virtual control law α is designed as:

$$\alpha = -k_1 e_1 + \Phi_T(\Psi) + 4B^2 l_c \dot{\Psi}_d + u_{r1} \tag{21}$$

u_{r1} is robustness designed for flow disturbance.

$$u_{r1} = -\hat{B}_1 \tanh\left(\frac{e_1}{\eta_1}\right) \tag{22}$$

η_1 is design parameter, $\hat{B}_1$ is the estimated value of B_1, the unknown upper bound of disturbance, error is defined as:

$$\tilde{B}_1 = B_1 - \hat{B}_1$$

For the first subsystem, choose Lyapunov function V_1:

$$V_1 = 2B^2 l_c e_1^2 + \frac{1}{2}\tilde{B}_1^2 \tag{23}$$

Take the derivative of V_1:

$$\begin{aligned} \dot{V}_1 &= 4B^2 l_c e_1 \dot{e}_1 - \tilde{B}_1 \dot{\hat{B}} \\ &= e_1(e_2 + \alpha - \Phi_T(\Psi) + d_\Phi(\xi)) \\ &\quad -4B^2 l_c e_1 \dot{\Psi}_d - \tilde{B}_1 \dot{\hat{B}} \end{aligned} \tag{24}$$

Substitute Eqs. (21) and (22) in above equation:

$$\begin{aligned} \dot{V}_1 &= -k_1 e_1^2 + e_1 e_2 + e_1 d_\Phi(\xi) - e_1 \hat{B}_1 \tanh\left(\frac{e_1}{\eta_1}\right) \\ &\le -k_1 e_1^2 + e_1 e_2 + |e_1|\, B_1 - e_1 B_1 \tanh\left(\frac{e_1}{\eta_1}\right) \\ &\quad + e_1 \tilde{B}_1 \tanh\left(\frac{e_1}{\eta_1}\right) - \tilde{B}_1 \dot{\hat{B}}_1 \end{aligned} \tag{25}$$

Considering inequation [23] $0 \le |x| - x\tanh(x/\mu) \le 0.2785\,\mu$, the above equation becomes:

$$\begin{aligned} \dot{V}_1 &\le -k_1 e_1^2 + e_1 e_2 + 0.2785\eta_1 B_1 + e_1 \tilde{B}_1 \tanh\left(\frac{e_1}{\eta_1}\right) \\ &\quad -\tilde{B}_1 \dot{\hat{B}}_1 \end{aligned} \tag{26}$$

Adaptive law of $\hat{B}_1$ is defined as:

$$\dot{\hat{B}}_1 = e_1 \tanh\left(\frac{e_1}{\eta_1}\right) \tag{27}$$

Equation (26) becomes:

$$\dot{V}_1 \le -k_1 e_1^2 + e_1 e_2 + 0.2785\eta_1 B_1 \tag{28}$$

and $e_1 e_2$ will be eliminated in next step design.

Step 2: Obtain actual control law.

$$\dot{e}_2 = \dot{\Phi} - \dot{\alpha} = \frac{1}{l_c}(\Psi_C(\Phi) - u - \Psi + d_\Psi(\xi)) - \dot{\alpha} \tag{29}$$

Actual control law is designed as:

$$u = k_2 e_2 + e_1 + \hat{f}_\alpha(\Psi, \Phi) - u_{r2} \tag{30}$$

$\hat{f}_\alpha(\Psi, \Phi) = \hat{\mathbf{W}}^T \mathbf{P}$ is fuzzy system and is used to approximate the function $f_\alpha(\Psi, \Phi) = \Psi_C(\Phi) - \Psi - l_c\dot{\alpha}$. u_{r2} is robustness designed for pressure disturbance.

$$u_{r2} = -\hat{B}_2 \tanh\left(\frac{e_2}{\eta_2}\right) \tag{31}$$

$\hat{B}_2$ is the estimated value of the unknown upper bound of pressure disturbance. Error is defined as: $\tilde{W} = W^* - \hat{W}$, $\tilde{B}_2 = B_2 - \hat{B}_2$

Choose the function Lyapunov for the system:

$$V = V_1 + \frac{l_c}{2}e_2^2 + \frac{1}{2\lambda}\tilde{\mathbf{W}}^T\tilde{\mathbf{W}} + \frac{1}{2}\tilde{B}_2^2 \tag{32}$$

Taking the derivative of V with respect to time and substituting actual control law results in:

$$\begin{aligned} \dot{V} &= \dot{V}_1 + l_c e_2 \dot{e}_2 - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \\ &= \dot{V}_1 + e_2(f_\alpha(\Psi, \Phi) - u + d_\Psi(\xi)) - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \\ &= \dot{V}_1 - k_2 e_2 - e_1 e_2 + e_2 f_\alpha(\Psi, \Phi) - e_2 \hat{f}_\alpha(\Psi, \Phi) \\ &\quad + e_2 d_\Psi(\xi) - e_2\hat{B}_2 \tanh\left(\frac{e_2}{\eta_2}\right) - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \\ &= \dot{V}_1 - k_2 e_2 - e_1 e_2 + e_2\tilde{\mathbf{W}}^T\mathbf{P} + e_2\varepsilon + e_2 d_\Psi(\xi) \\ &\quad - e_2\hat{B}_2 \tanh\left(\frac{e_2}{\eta_2}\right) - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \\ &\le \dot{V}_1 - k_2 e_2 - e_1 e_2 + e_2\tilde{\mathbf{W}}^T\mathbf{P} + \frac{1}{2}e_2^2 + \frac{1}{2}\varepsilon^2 + |e_2|\, B_2 \\ &\quad - e_2 B_2 \tanh\left(\frac{e_2}{\eta_2}\right) + e_2\hat{B}_2 \tanh\left(\frac{e_2}{\eta_2}\right) \\ &\quad - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \\ &\le \dot{V}_1 - k_2 e_2 - e_1 e_2 + \frac{1}{2}e_2^2 + \frac{1}{2}\varepsilon^2 + 0.2785\eta_2 B_2 \\ &\quad + e_2\tilde{\mathbf{W}}^T\mathbf{P} + e_2\tilde{B}_2 \tanh\left(\frac{e_2}{\eta_2}\right) - \frac{1}{\lambda}\tilde{\mathbf{W}}^T\dot{\hat{\mathbf{W}}} - \tilde{B}_2\dot{\hat{B}}_2 \end{aligned} \tag{33}$$

Define parameter adaptive law as:

$$\dot{\hat{\mathbf{W}}} = \lambda e_2 \mathbf{P} \tag{34}$$

$$\dot{\hat{B}}_2 = e_2 \tanh\left(\frac{e_2}{\eta_2}\right) \tag{35}$$

So Eq. (33) is transformed into:

$$\dot{V} \le -k_1 e_1^2 - (k_2 - 0.5)e_2^2 + \sigma \tag{36}$$

where

$$\sigma = \frac{1}{2}\varepsilon^2 + 0.2785\eta_1 B_1 + 0.2785\eta_2 B_2$$

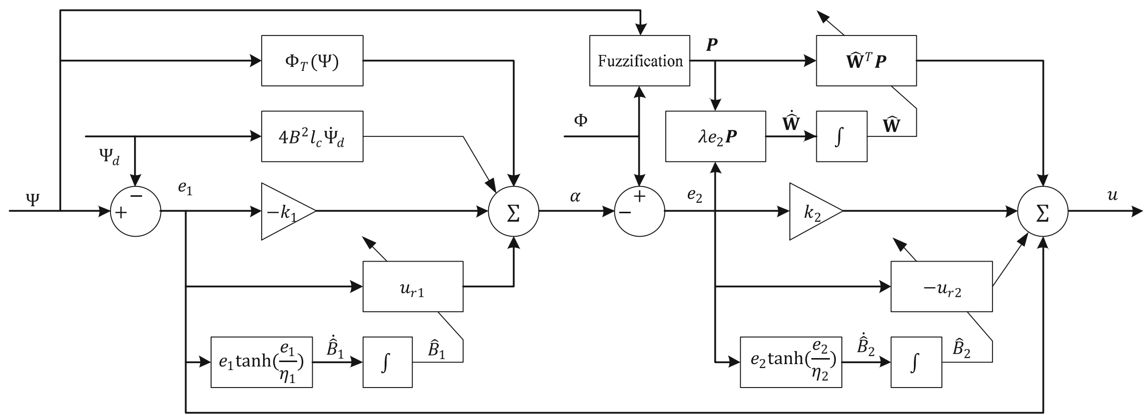


Fig. 2 The principle diagram of the whole adaptive controller

Design parameter $k_2 \geq 0.5$, let $k_0 = \min(k_1, k_2 - 0.5)$, since approximation error of fuzzy is bounded, $\sigma > 0$ and bounded, then Eq. (36) can be adapted into:

$$\dot{V} \leq -k_0(e_1^2 + e_2^2) + \sigma \tag{37}$$

Integrate the above equation on the region $\xi \in [0, T]$.

$$V(T) - V(0) \leq -k_0 \int_0^T (e_1^2 + e_2^2)\mathrm{d}\xi + \int_0^T \sigma \,\mathrm{d}\xi \tag{38}$$

Considering $V(T) \geq 0$, it can be obtained that:

$$\int_0^T (e_1^2 + e_2^2)\mathrm{d}\xi \leq \frac{1}{k_0} V(0) + \frac{1}{k_0} \int_0^T \sigma \,\mathrm{d}\xi \tag{39}$$

From Eq. (39), it is known that for System (13) and (14), by applying control law (30) and adaptive laws (27), (34) and (35), closed-loop system is stable and ultimately bounded, which is in-line with tracking error.

The principle diagram of the whole adaptive controller is given below which is showed in Fig. 2.

The input order Ψ_d of the controller is the set value of pressure rise coefficient in compressor, the feedback signals are pressure rise coefficient Ψ in compressor and flow coefficient Φ of compressor inlet, the output of controller is pressure drop u of CCV. $k_1, k_2, \lambda, \eta_1$ and η_2 in the diagram are controller parameters which need to be designed, $\hat{B}_1$, $\hat{B}_2$ and $\hat{W}$ are adjusted automatically by corresponding adaptive law.

5 Simulation Results and Discussions

To verify the above control algorithm, we build active surge control platform of compressor in MATLAB Simulink which mainly includes throttle valve module, controller module, and compressor module and so on. Controller module and compressor module are realized by S-function. Without close-loop control in compressor, throttle vale module is transitioned from the balanced state to the surge state by decreasing the opening of throttle valve γ_T; compressor model is second-order compressor surge model with CCV; controller is robust adaptive fuzzy controller. To verify the effect of the controller designed in this paper, simulation was taken on the above-mentioned active surge control platform of compressor by MATLAB.

Test I simulated compressor surge without applying control. Parameters are: $B = 1.8, l_\mathrm{c} = 2, H = 0.18, W = 0.25, \psi_\mathrm{CO} = 0.3$. When simulation time is 100, throttling valve parameter γ_T smoothly reduced from 0.65 to 0.6. Simulation results are shown in Fig. 3. The result shows that compressor entered into deep surge after turning down the opening of throttle valve.

Test II simulated active control on compressor surge without disturbance. Controller parameters are: $k_1 = 0.5, k_2 = 1.0, \eta_1 = 0.05, \eta_2 = 0.05, \lambda = 2.0$. Divide flow coefficient and pressure coefficient from 0 to 1.0 into 9 grades. Membership function is:

$$\exp(-(x - 0.1 * i)^2), \quad i = 1, \ldots, 9$$

Simulation results are shown in Figs. 4 and 5. The result shows that without disturbance, compressor was successfully stabilized in the region of low flow by the controller and did not enter into surge which demonstrated the effectiveness of the controller.

Test III simulated active control of compressor surge considering there is flow and pressure disturbance. Control parameters are the same as Test II. Disturbance signals are:

$$d_\Phi(\xi) = 0.02 * \sin(0.1\xi) + 0.02 * \cos(0.4\xi) \tag{40}$$

$$d_\Psi(\xi) = 0.02 * \sin(0.1\xi) + 0.02 * \cos(0.4\xi) \tag{41}$$

Simulation results are shown as Figs. 6 and 7.

The result shows that with disturbance of flow and pressure, controller could still be stabilized in the region of low flow by the controller and did not enter into surge which demonstrated the strong robustness of the controller. At the

Fig. 3 Test I simulation results

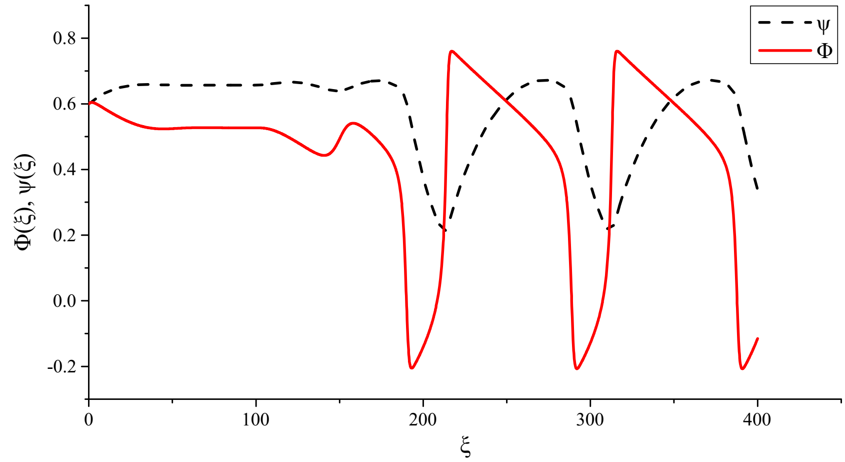


Fig. 4 Test II simulation results

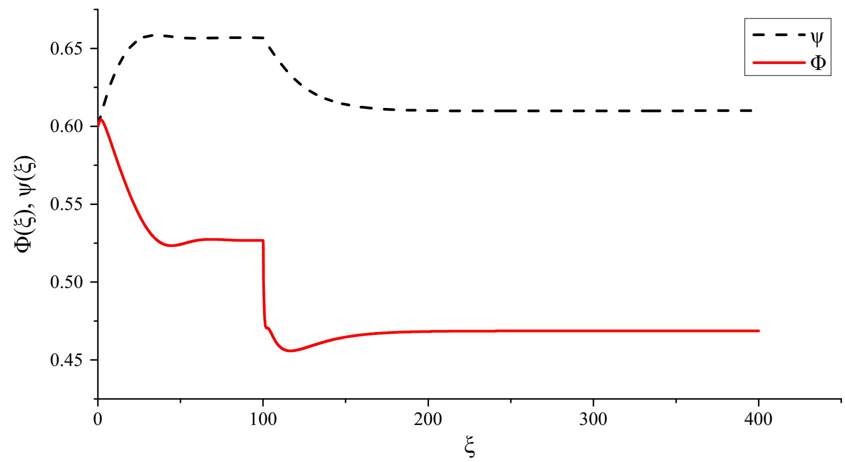
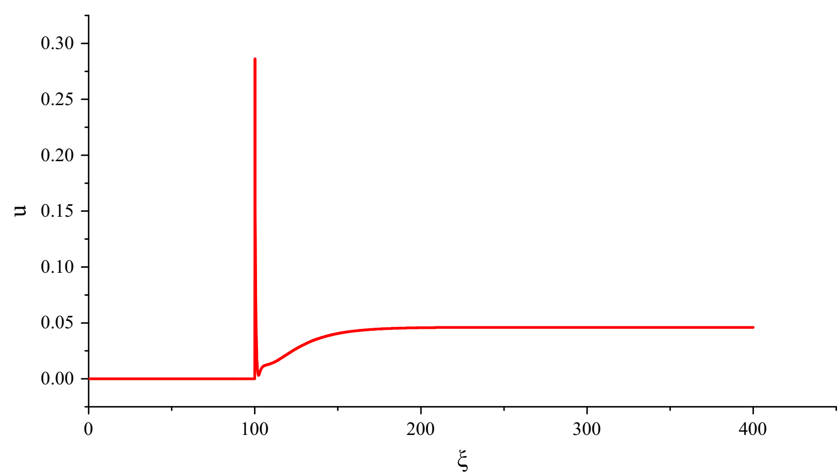


Fig. 5 Test II controller output



beginning of controller being brought in, there is a momentary jump, which is caused by fuzzy system not adapted to adjustment at first. Controlling quantity would have continuous output after fuzzy system adapted and was ready. It can be seen from simulation results that fuzzy system took a very short time to adapt. In Test III, controller output was seen with negative values for a little while, while pressure drop with CCV cannot be negative. To enable CCV to achieve bidirectional adjustment, need to preset an initial offset to CCV.

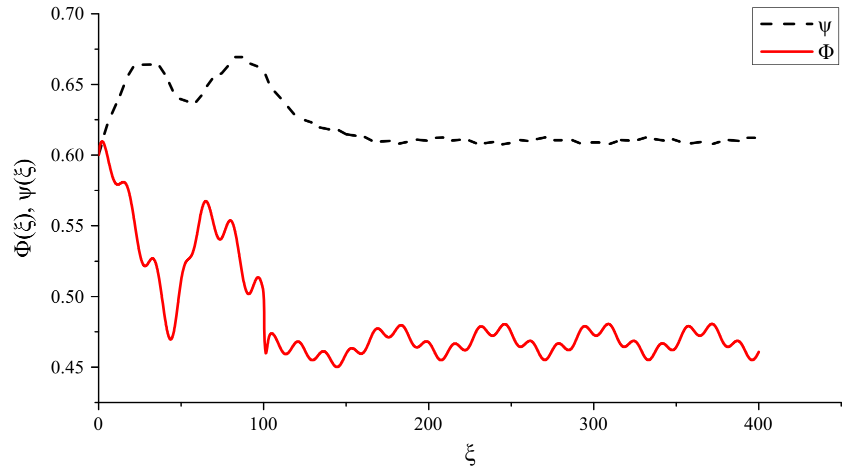


Fig. 6 Test III simulation results

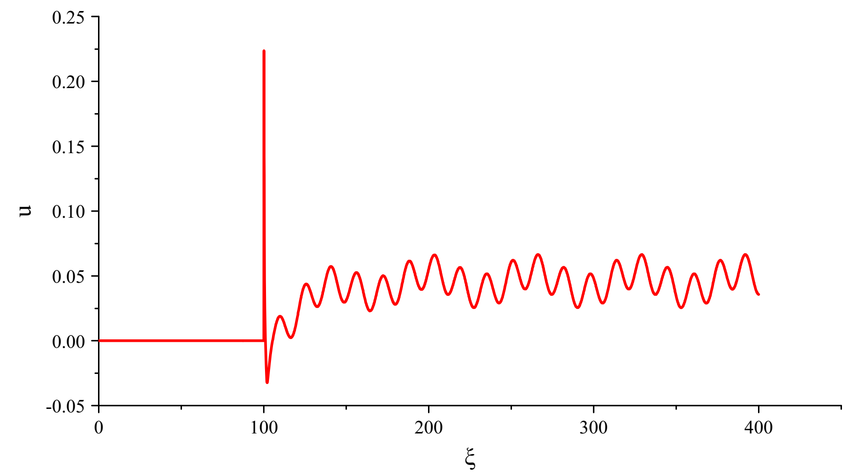


Fig. 7 Test III Controller Output

6 Conclusions

Surge active control is an important way to extend the operating range of compression systems. But most of existing active controllers depend on known system characteristics and thus have limited engineering application. Therefore, following the concept of active control as well as based on second-order MG surge model, using CCV as actuator, active compressor surge controller is designed on the basis of fuzzy system. To verify the above control algorithm, we build active surge control platform of compressor in MATLAB Simulink which mainly includes throttle valve module, controller module, and compressor module and so on. Simulation results demonstrate that the designed controller is effective in active surge control and enables compressor to work steadily in zones beyond the surge line and enlarges compressor's operating range. In the process of designing the controller, compressor's characteristics and the system's parameter B are not needed. The controller has got quite strong robustness against uncertainties like unmodelled dynamics and system disturbances, therefore has shown greater application prospects.

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