Fuzzy Gain Scheduling of an I-P-D Controller for Oscillation Compensation in a Sticky Pneumatic Control Valve

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Abstract — **In the control loop, the pneumatic control valve is a highly nonlinear component with nonlinearities such as stiction, which induces the limit cycle and oscillations in the steady-state response. This paper successfully shows the elimination of oscillations from the process variable (PV) and controller output (OP) due to the sticky pneumatic control valve using a proposed control methodology, namely, fuzzy gain scheduling of an integral minus proportional minus derivative (FGS I-P-D) controller. The uniqueness of the proposed control method is that it is a standalone solution and it does not require any additional compensating component in the closed loop, as reported in the literature. In the I-P-D controller, integral action is performed on the error signal, whereas proportional and derivative actions are realized using the PV. The gains of the I-P-D controller were computed at runtime using a Mamdani-type fuzzy inference mechanism. The performance of the FGS I-P-D controller was compared with that of the conventional I-P-D controller for setpoint tracking capability and external disturbance rejection at different operating points on a laboratory scale pressure control unit. The experimental results clearly show that the FGS I-P-D controller outperformed the classical I-P-D controller in every aspect of investigation performed and suppressed efficiently any stiction-induced oscillations in PV and OP.**

Index Terms — **Fuzzy gain scheduling, I-P-D, stiction, pneumatic control valve, hysteresis, pressure control system.**

I. Introduction

Control loop performance is a vital idea attributable to the fact that only 33% of all industrial control loops give a pleasant execution [1]. These execution issues can have an unfavorable impact on the efficiency and benefit of any process industry. The poor execution of any feedback loop of any typical process industry may show itself as an oscillation and poor setpoint following at the operating point. Oscillation is an imperative issue for the degradation of plant performance. There can be an assortment of various sources of oscillation e.g., poor controller tuning, external disturbances, and the presence of nonlinearities [2]. Of all wellsprings of oscillation in the control loop, valve nonlinearities, such as stiction are considered to be one of the significant reasons for oscillations. The detection and diagnosis of oscillation are an important aspect of plant performance. Most of the research revolves around the nonlinearity in the process due to stiction in the pneumatic control valve [3]. Data-driven methods to deal with distinguished nonlinearities in the control loop are more useful as they do not require an exact model of the process dynamic, which is rarely accessible [4]. Choudhury et al. described stiction on the basis of real data as as follows: "stiction is a property of an element such that its smooth movement in response to a varying input is preceded by a static part (dead-band + stick-band), followed by a sudden abrupt jump called slip-jump." Slip-jump is expressed as a percentage of the output span. Its origin in a mechanical system is static friction, which exceeds the dynamic friction.

The input–output behavior of a sticky pneumatic control valve is shown in Fig. 1. It consists of three parts; stick-band + dead-band (S_1 and S_2), slip-jump and moving phase. When the valve is at rest (S_2), it sticks because it cannot overcome the static friction. After the controller output overcomes S_2, the valve jumps to a new position and continues to move (J_3). Due to the very low or zero velocity, the valve may stick again (J_3), while traveling in the same direction. This can be overcome, when the controller output signal is larger than S_2. S_1 and S_2 represent the behavior of the valve when it is not moving although the input to the valve keeps changing.

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The slip-jump phenomenon represents the abrupt conversion of potential energy stored in the actuator due to high static friction into kinetic energy as the valve starts to move. The magnitude of the slip-jump is very crucial in determining the limit cyclic behavior introduced by stiction. Once the valve jumps or slips, it continues to move until it sticks again. In this moving phase, dynamic friction is present, which may be much lower than the static friction. This is repeated with the reversal of valve direction.

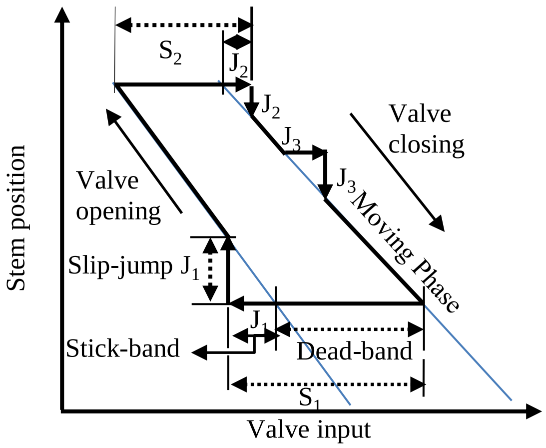

Figure 1. Input–output behavior of a typical sticky pneumatic control valve [13].

II. LITERATURE REVIEW

There are some techniques reported in the literature to mitigate stiction-based problems in a closed-loop system. These techniques can be classified as, knocker, constant reinforcement, two-move method and optimization approaches, and the rest of the literature survey is organized accordingly.

One of these techniques is the knocker approach, which was first introduced by Hägglund [5]. In this method, stiction is overcome by knocking on the valve by adding short pulses to the control signal. The knocker signal has a sequence of pulses, which is continuously applied in the direction of the rate of change of the control signal to prevent instabilities in the process output. This method detracts the oscillation in the control loop, but at the cost of quicker and more extensive valve stem movement. This shows an invasive approach to compensate the stiction-based oscillation in the control loop. Invasive approaches are not desirable for routine maintenance of control valves.

Further, Srinivasan and Rengaswamy [6] provided a non-invasive technique for controlling oscillations on the basis of routine operating data of the process variable and controller output. They showed that the knocker performance was influenced by the pulse parameters, and they proposed a procedure to automate the knocker parameters based on the basis of the stiction measure. They reduced the variability in OP and PV six to seven times, but at the cost of aggressive stem movement. Such an aggressive stem movement may not be preferred as it can wear out the valve quickly. Furthermore, Cuadros et al. [7] modified the knocker approach by adding a supervision layer to the control error and added the pulses when there was a requirement of reduction of process error only. This technique reduced the integrated absolute error (IAE) and valve stem movements in the closed-loop control system.

Ivan and Lakshminarayanan [8] suggested a constant reinforcement technique in which a constant amplitude signal is added to the controller output signal, similar to a knocker method. The value of the constant signal is calculated as; $\alpha(t) = a \times sign(\Delta u)$, where, a is the amplitude and u is the controller output. The new compensator gets rid of the periodicity of the knocker approach and executes lower variability in the PV and OP signal. However, this method does not vanquish the valve aggressiveness.

Further, Srinivasan and Rengaswamy [9] developed an effective stiction compensation technique, namely, two-move approach, which used a distinct compensator. At each instant, the compensator output was derived on the basis of instantaneous values of OP, the derivative of the OP, and the stiction band. Thus, the compensation method was highly reliant on the precise measurement of stiction measure. In addition, it was also assumed that there should be no model mismatch, which is not possible in a real-world scenario. In the continuation of this, Farenzena and Trierweiler [10] utilized the two-move approach to modify the PI controller block to compensate the stiction. They believed that the presented technique could produce a faster closed-loop performance than the open-loop process. In addition, it can track the changes in the reference setpoint efficiently and handle external disturbances at the cost of a steady-state error in the PV. Cuadros et al. [11] revisited the famous two-move approach and proposed an enhanced compensator-based stiction handling technique in pneumatic control valves. They assumed an unrealistic assumption while setting the parameters of the compensator: that the pneumatic control valve dynamics and the process dynamics are similar. The proposed compensating technique handles external disturbances well, but failed to provide a satisfactory performance in a cascade loop. Further, Wang [12] proposed a closed-loop compensation technique to eliminate oscillations caused by pneumatic control valve stiction. In this method, a short-time rectangular wave is added to the reference in order to incorporate two movements for the pneumatic control valve to reach a desired position.

Furthermore, in order to bring the process quickly to a steady state, an optimization-based approach was proposed by Srinivasan and Rengaswamy. They performed simulation studies to demonstrate the optimal approach. Recently, some work on the effective control of oscillation in the PV due to the sticky pneumatic valve has been reported in the literature. Mishra et al. [13, 14] proposed a Stiction Combating Intelligent Controller (SCIC), a variable gain fuzzy PI controller. They evaluated the performances of the proposed SCIC controller and conventional PI controller on a pilot plant for efficient control of a flow process having a sticky control valve. Experimental results revealed that the SCIC controller outperforms the traditional PI controller for changes in the reference setpoint and disturbance rejection. Further, Mishra et al. [15] assessed the performance of the SCIC controller experimentally by effective ratio control in the presence of a control valve with stiction nonlinearity. Again, the SCIC controller showed a superior performance over the classical PI controller. Furthermore, Mishra et al. [16] proposed a novel nonlinear PI controller (NPIC) for efficient control of a flow process having a sticky control valve on a laboratory-scale plant. The viability of the proposed NPIC was assessed by comparing its execution with a classical PI controller. The NPIC exhibited better execution as compared to the PI controller.

Soft computing techniques were further adopted by Rohilla et al. [17] using a fuzzy PI controller to control a pressure process having a sticky pneumatic control valve. They developed a fuzzy PI controller using triangular and Gaussian membership functions (MFs). Experimental studies revealed that fuzzy PI controllers with a Gaussian controller give a better performance. Further, Rohilla et al. [18] used the fuzzy I+PD controller to control a pressure system having a sticky pneumatic control valve. This was a combination of a fuzzy I controller and a conventional PD controller. Again, experimental investigations showed that the fuzzy I+PD controller demonstrates a much superior performance as compared to the conventional PID controller.

It is clear from the literature review presented above that all methods show a great capability to minimize oscillations in the control loop. However, they have a couple of impediments. For example, the knocker method provides a faster stem movement of the control valve, whereas the two-move method requires an exact model of the plant. Besides, an extra part is required to remunerate the stiction-based oscillation. Moreover, these methods may not provide the best setpoint following and disturbance rejection. In addition, it has been reported in the literature that 95% of closed loops in the process industries are governed by conventional PID controllers [19]. Any solution around the conventional PID controller is acceptable for process industries. Various modified versions of conventional and intelligent PID controllers, such as PI-D and I-P-D, are reported in the literature [20, 21].

To overcome these shortcomings, a straightforward fuzzy gain scheduling of an integral minus proportional minus derivative (FGS I-P-D) controller is proposed in this paper. This method is based on the method proposed by Zhao et al. [22]. Here, the integral action is performed on the error signal, whereas the proportional and derivative action is performed on the PV. The gains of the I-P-D controller are obtained via fuzzy logic at runtime. The proposed control scheme is a standalone adaptive solution for processes having a sticky pneumatic control valve that does not require any additional element in the closed loop, such as a compensator. The performance of FGS I-P-D and conventional I-P-D controllers was evaluated on a laboratory-scale pressure control unit for changes in the reference setpoint and external disturbance. Experimental studies that were carried out stated that the FGS I-P-D controller outperforms the I-P-D controller and very conveniently mitigates the control valve stiction-based oscillations from the PV and OP.

The main contributions of this work can be summarized as follows.

- In this paper, an FGS of a modified conventional PID controller (i.e., I-P-D controller) is proposed, namely, the FGS I-P-D controller.
- The FGS I-P-D controller successfully demonstrates the suppression of oscillations from the PV and OP due to the sticky pneumatic control valve on a laboratory-scale pressure control unit.
- The proposed control scheme is a standalone solution and no other compensating device is required in the feedback control loop, as reported in the literature.

III. EXPERIMENTAL SETUP AND PROBLEM FORMULATION

In this section, detection and quantification of nonlinearities (such as stiction, hysteresis, and stick band) in the pneumatic control valve and details of equipment used in the experimental setup are discussed.

A. *Experimental Setup*

The pressure control unit available in the lab consists of following parts: a process tank, an equal percentage characteristic pneumatic control valve (actuator), a pressure sensor (SX30DN), a current-to-pressure (I/P) converter, a voltage-to-current (V/I) converter, and an I/O DAQ card (NI USB-6008). A snapshot of the experimental setup is shown in Fig. 2. The input to the system is a control signal generated by the controller. This determines the opening rate of the pneumatic control valve. These signals are transmitted to the plant through the I/O card. The output of the system is pressure, which is measured in terms of voltage by a piezo-resistive pressure sensor. Thus, the pressure signal (PV) is fed back to a PC with a control algorithm through the DAQ card.

The process tank is equipped with an inlet valve (V-3) at the bottom, through which compressed air is fed, and an outlet valve (V-5) at the top to provide leakage of air. Air is compressed by an electrically driven single-stage reciprocating air compressor with the following characteristics: 0.5-inch outlet, 1440 RPM, 1.5 kW, 2 HP, 220 V, and 200 psi.

The pneumatic control valve plays the role of the actuator, which controls the airflow coming from the air supply unit and maintains the user-specified pressure in the plant. The specifications of the pneumatic control valve are as follows: 10-square-inch diaphragm, 0.5-inch port size, valve flow coefficient $C_V = 0.44$, stroke length = 0.551 inches, air supply = 3 –15 psi, and control action = air-to-close. The plant has a piezo-resistive transducer that converts pressure (0–2 bar) into voltage (0–2.5 V). The V/I converter converts voltage (0–2.5 V) into current (4–20 mA), and the I/P converter converts current (4–20 mA) into pressure (3–15 psi). The National Instruments™ I/O card is used to create a connection between the controller and the plant. The DAQ card has 12 digital inputs/outputs with 32-bit resolution, eight analog inputs, and two static analog outputs with 12-bit resolution (range: −10 V to 10 V). The controller was designed in Matlab/Simulink® (2011b) software on a personal computer equipped with an Intel Core™ i7 processor (2.5 GHz) and 4 GB of RAM. The fourth-order Runge–Kutta ODE solver with a sampling rate of 0.01 s was utilized for all the experimental work.

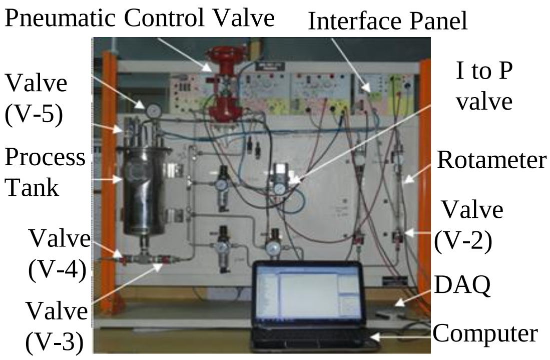

Figure 2. Experimental setup of the pressure control system.

B. Problem Formulation

Pneumatic control valves are widely used as an actuator in process control systems. Friction in valves cause nonlinear phenomena, such as hysteresis, stick-slip motion, and oscillation. The typical variation of PV and OP with respect to time of the pressure control unit is shown in Fig. 3. Both PV and OP show an oscillatory behavior. These signals oscillate because of the static friction in the pneumatic control valve. The corresponding error signal is plotted in Fig. 4, which shows the asymmetric distribution of error. This shows that nonlinearity (stiction) is present in the pneumatic control valve. It is notable that stiction in the control valve produces limit cycles in the PV and OP. Plotting PV versus OP produces elliptical cyclic patterns as shown in Fig. 5. If an ellipse is fitted between the PV-OP plots, it will show that the valve experiences stiction problems.

Stiction can be quantified from the maximum width of the ellipse fitted in the PV-OP plot measured in the direction of OP [4]. The amount of apparent stiction can be obtained using the following expression: $\Delta x = 2mn/\sqrt{(m^2 \sin^2\alpha + n^2 \cos^2\alpha}$, where m and n are the lengths of the major and minor axes of the fitted ellipse, respectively, and α is the angle of rotation. In the experimental setup used, these values are as follows: m = 11, n = 2, and α = 30°. The data used to plot the stiction is measured in a closed loop of a single-input, single-output (SISO) system with a conventional I-P-D controller having the following gain parameters: $K_i = 1.253$, $K_p = 2.4$, and $K_d = 1.14$. It shows 6% apparent stiction in the pneumatic control valve, but even 1% of the stiction in the control valve can degrade the performance of the plant [9].

In the pressure control unit, the pneumatic control valve is totally open at 0 V and is totally shut at 2.5 V. Hysteresis was measured by forward (valve closed) and reverse (valve open) movement of the valve stem on applying the input voltage signal (0–2.5 V) from 0% to 100% and afterwards from 100% to 0%, as shown in Fig. 6.

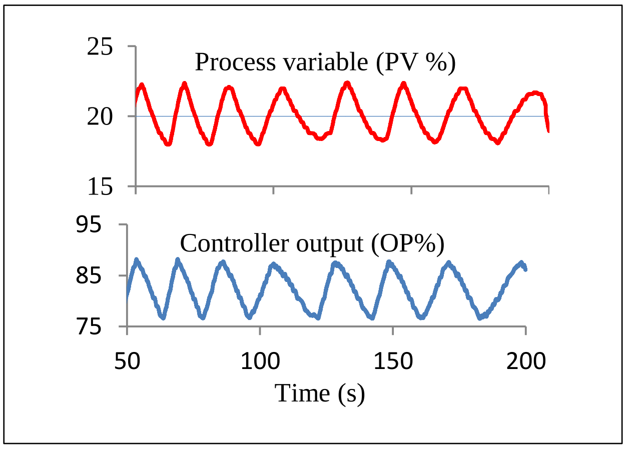

Figure 3. Typical oscillation in PV and OP due to sticky pneumatic control valve stiction.

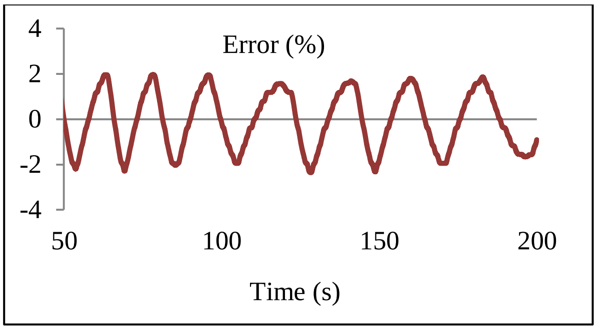

Figure 4. Typical oscillation in error due to the sticky pneumatic control valve.

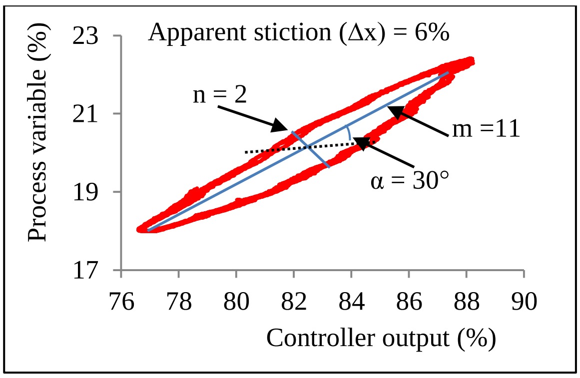

Figure 5. Stiction quantification by the fitted ellipse in PV-OP.

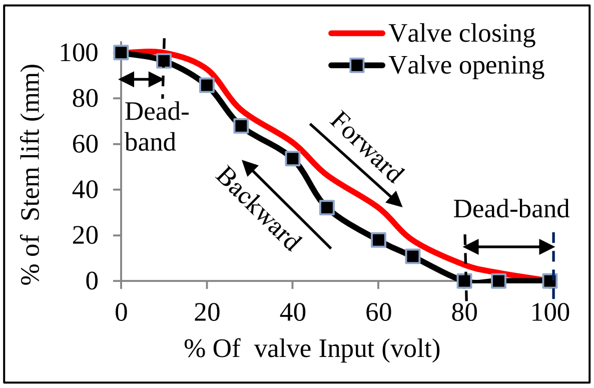

Figure 6. Hysteresis and dead band in the sticky pneumatic control valve.

IV. DESIGN OF AN FGS I-P-D CONTROLLER

The main objective of this investigation is to design an adaptive controller, to limit the inconstancy in PV-OP and mitigate stiction impact. For this purpose, an intelligent control algorithm, namely fuzzy gain scheduling of I-P-D control strategy is proposed. Normally, stiction introduce fluctuations in PV and OP. These oscillations can be controlled by changing the gains of I-P-D controller at run time using fuzzy logic. The FGS I-P-D controller improves the PV-OP response by smoothing motions at steady state. The range of gains of the FGS I-P-D controller is obtained by Zeigler-Nichols (ZN) tuning method and the same gains are used in the conventional I-P-D controller.

A. *ZN Method*

For the current study, the ZN closed loop tuning method was used to tune the gains of the conventional PID controller. The controller was tuned for setpoint following for 40% operating point. The ultimate gain K_U and period P_U are obtained as 4 and 3.83 s, respectively. Thus, the gains of the conventional PID controller are computed. The tuned values of K_p, K_i and K_d are 2.4, 1.253 and 1.149 respectively. The same gains are used for the conventional I-P-D controller throughout the experimental study.

B. *Conventional I-P-D Control Structure*

The conventional I-P-D controller is a modified version of the classical PID controller; in the time domain, it is defined as:

$$U_{I-P-D}\,(t) = K_i \int e(t)dt - K_p y(t) - K_d \frac{dy}{dt} \qquad (1)$$

where $U_{I-P-D}\,(t)$ is the controller output, $e(t)$ is the error signal, $r(t)$ is the referenced setpoint, $y(t)$ is the PV, K_i is the integral gain, K_p is the proportional gain, and K_d is the derivative gain. The structure of the conventional I-P-D control system is shown in Fig. 7. With this structure, only the integral component deals with the error signal, whereas proportional and derivative components act on the PV [i.e., $y(t)$]. The controller parameters are kept the same as obtained by the ZN tuning method as mentioned above. The control system is implemented in a SISO mode, and the experiments are carried out on a pressure control system with an equal percentage pneumatic control valve. The step response of a closed-loop control system shows an oscillatory response after the transient response. So, in order to achieve a smooth, steady-state response, it is essential to determine a new set of gains of the I-P-D controller. Therefore, an intelligent technique is required to compute the optimal value of the gains of the I-P-D controller at runtime. Therefore, an FGS algorithm is utilized to overcome this issue.

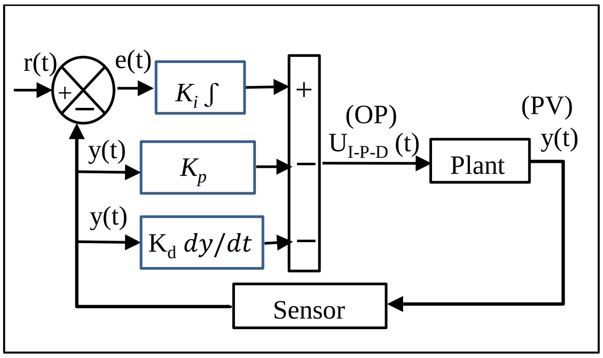

Figure 7. Conventional I-P-D control system structure.

C. *FGS I-P-D Control Structure*

In the present work, an FGS I-P-D controller is proposed, which updates the gains of the I-P-D controller, namely, K_i, K_p, and K_d, in real time in a closed-loop system as shown in Fig. 8, where $r(t)$, $y(t)$, and f_1 and f_2 are the reference input, plant output, and fuzzy gains, respectively. The fuzzy gains ($f_1 = 0.8$ and $f_2 = 1.8$) were tuned manually. The inputs to the FGS are error $e(t)$ and rate of change of error de/dt.

The gains of the I-P-D controller parameters are updated according to the error and rate of change of error at each sample time constant by the Mamdani fuzzy algorithm [23, 24]. In this control scheme, it is assumed that the controller gains are in prescribed ranges. $K_i \in [K_{i,min}\ K_{i,max}]$, $K_p \in [K_{p,min}\ K_{p,max}]$, and $K_d \in [K_{d,min}\ K_{d,max}]$. The ranges of the controller gains are determined experimentally by the ZN oscillation method before applying the FGS I-P-D algorithm (i.e., $K_i \in [0.05\ 0.2]$, $K_p \in [0.5\ 2.5]$, and $K_d \in [0.5\ 1.5]$). For convenience, K_i, K_p, and K_d are normalized into the range between 0 and 1 by considering the following linear transformations [25]:

$$K_i^* = (K_i - K_{i,min}) / (K_{i,max} - K_{i,min}), \qquad (2)$$

$$K_p^* = (K_p - K_{p,min}) / (K_{p,max} - K_{p,min}), \qquad (3)$$

$$K_d^* = (K_d - K_{d,min}) / (K_{d,max} - K_{d,min}). \qquad (4)$$

For the current study, Gaussian MFs are considered for input variables, namely, the error and rate of change of error, and output variables, namely, K_i, K_p, and K_d, are shown in Figs. 9 and 10, respectively. The input linguistic variables are defined over five MFs, namely, Negative Big (NB), Negative (N), Zero (Z), Positive (P), and Positive Big (PB), and output linguistic variables are described over five MFs, namely, Small (S), Medium Small (MS), Medium (M), Large Medium (LM), and Large (L).The centroid method is used to find the crisp the fuzzy logic controller. The ranges of the MFs of inputs ($e(t), de/dt$) and outputs (K_i, K_p, and K_d) are determined as -1 to 1 and 0 to 1, respectively. Table I shows the fuzzy logic based rules for K_i^*, K_p^*, and K_d^*. The rule base for all parameters is obtained on the basis of the system response and characteristic. Once the parameters K_i^*, K_p^*, and K_d^* are obtained, the controller gains K_i, K_p, and K_d are calculated using Eqs. (2), (3), and (4).

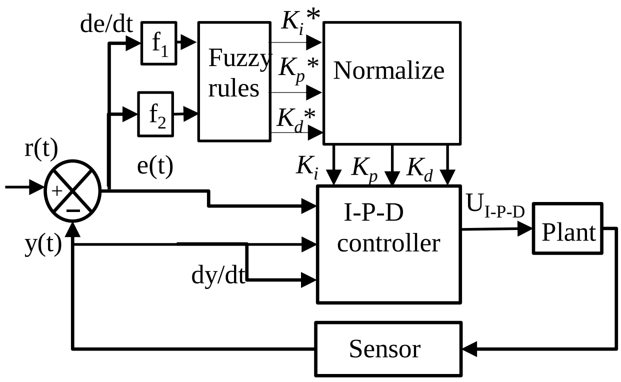

Figure 8. FGS I-P-D control system structure.

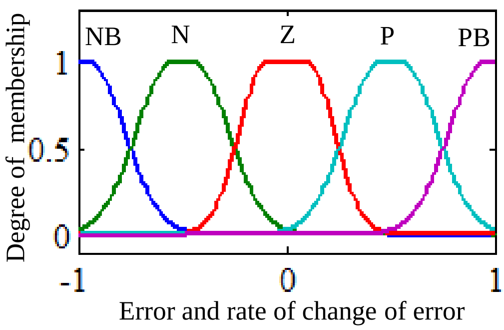

Figure 9. Input MFs.

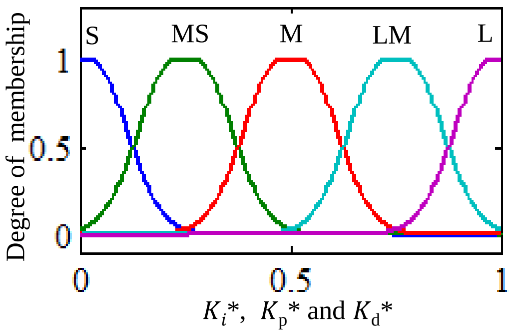

Figure 10. Output MFs.

TABLE I. FUZZY RULE FOR CONTROLLER OUTPUT K_i^*, K_p^*, AND K_d^*.

Error	Rate of change of error				
	NB	N	Z	P	PB
NB	L\S\LM	L\S\MS	LM\MS\S	M\M\MS	M\M\LM
N	LM\S\M	LM\MS\S	LM\MS\MS	M\M\MS	MS\M\M
Z	LM\MS\M	LM\MS\MS	M\M\MS	MS\LM\MS	MS\LM\M
P	LM\MS\M	MS\LS\LM	MS\LM\M	MS\LM\M	MS\L\L
PB	M\M\L	MS\LM\L	MS\LM\LM	S\L\LM	L\L\L

V. EXPERIMENTAL RESULTS AND DISCUSSION

The conventional I-P-D and FGS I-P-D controllers were tested in a SISO laboratory-scale pressure control unit for setpoint tracking, disturbance rejection, and robustness. In the experimental setup, compressed air is circulated through the pipes. The output of the system is pressure, which is converted into an electrical signal by a pressure transducer. The input was supplied in the form of a step input signal. The conventional I-P-D controller was tuned by the ZN tuning method, and the gains were set as follows: K_i = 1.125, K_p = 2.4, and K_d = 1.149, respectively. To evaluate the controllers' performance for setpoint following and disturbance rejection, three distinctive operating points were chosen, that is, 20%, 40%, and 60% of the maximum pressure. The controllers were designed in Matlab/Simulink® environment, with a fourth-order Runge–Kutta solver. The sampling time was set to 0.01 s throughout the experiments. In order to compare the performances of the conventional I-P-D versus the FGS I-P-D controller, the following four integral performance criteria were used to characterize the setpoint following responses:

- Integrated absolute error (IAE)
- Integrated time absolute error (ITAE)
- Integrated squared error (ISE)
- Integrated time squared error (ITSE)

Figures 11, 12, 13, 14, 15, and 16 show the variation of PV and OP with respect to the time of the closed-loop control system for conventional I-P-D (dashed line) and FGS I-P-D (continuous line) controllers for a setpoint at 20%, 40%, and 60% of the maximum pressure. As can be clearly seen in these figures, the conventional that the I-P-D controller makes large fluctuations in the PV-OP and retains continuous oscillations at the steady-state region. On the contrary, the FGS I-P-D controller gets to the setpoint much faster than the conventional I-P-D controller, and it suppresses the stiction-based oscillations by fast updating the controller gains.

The PV-OP mapping of the two controllers at 20%, 40%, and 60% setpoints is shown in Figs. 17, 18, and 19, respectively. These mappings show that the conventional I-P-D controller retains an elliptical pattern (apparent stiction), whereas the FGS I-P-D controller completely mitigates the elliptical pattern at all the considered operating points. Data in Figs. 17, 18, and 19 were collected after the transient response died out. A total of 5,000 samples were used to show the elliptical loop. The proposed FGS I-P-D controller shows a significant improvement in setpoint tracking over that of the conventional I-P-D controller.

Integral performance indices are commonly used to assess the performance of some closed-loop control systems as computable measures of the performance of that plant. The integral performance indices of the FGS I-P-D and conventional I-P-D controllers at 20%, 40%, and 60% setpoints are compared with the help of bar charts and presented in Figs. 20, 21, and 22, respectively. These indices have been calculated for a period of 0–200 s. The FGS I-P-D controller has considerably decreased the setpoint tracking error performance indices at all the operating points. Moreover, setpoint following and control activity exhibit an attractive behavior for stiction compensation at all the operating points, as the control targets are met without rambling.

The experimental investigations clearly demonstrated that the FGS I-P-D controller shows a much superior performance as compared to the I-P-D controller and is able to suppress stiction-based oscillations significantly from the PV and OP. The reduction of oscillations in PV and OP is obtained mainly by the update of the I-P-D controller gains by the fuzzy logic algorithm in the FGS I-P-D controller at runtime.

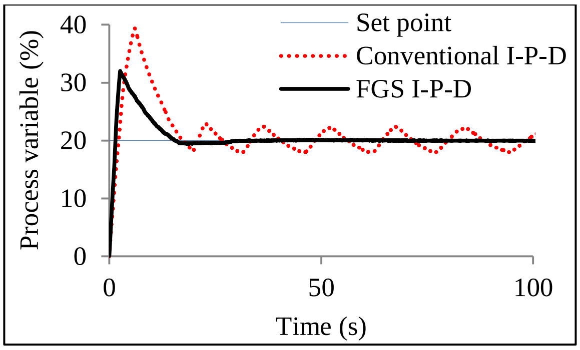

Figure 11. Response of the PV at the level of 20% of maximum pressure.

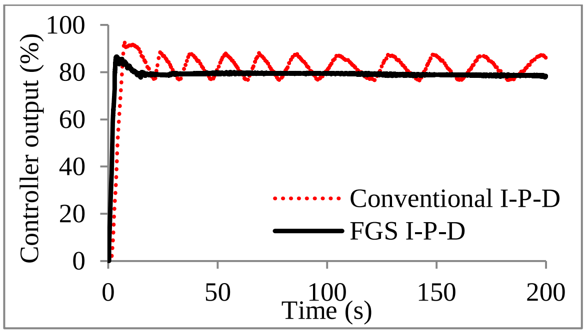

Figure 12. Response of the controller output at the level of 20% of maximum pressure.

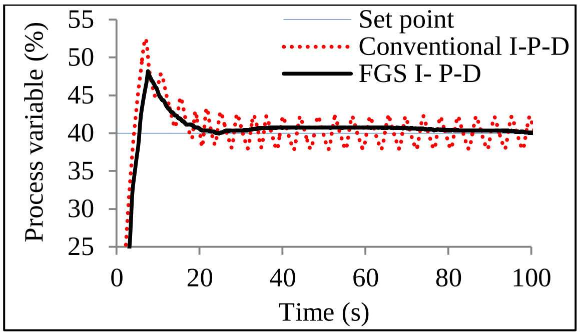

Figure 13. Response of the PV at the level of 40% of maximum pressure.

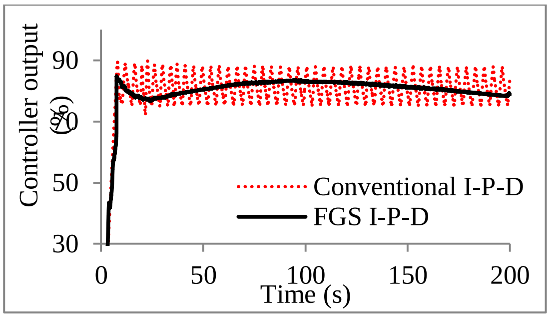

Figure 14. Response of the controller output at the level of 40% of maximum pressure.

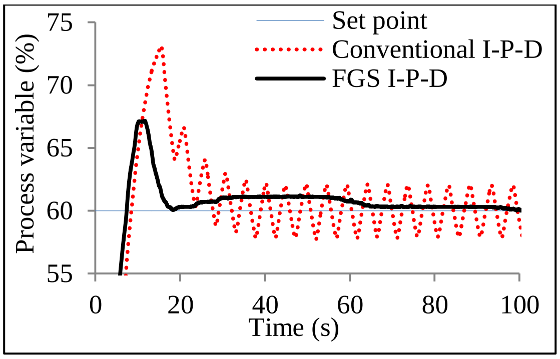

Figure 15. Response of the PV at the level of 60% of maximum pressure.

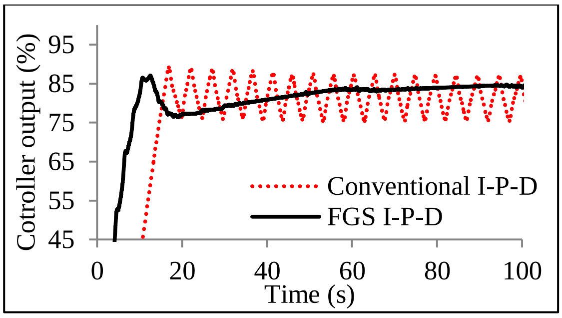

Figure 16. Response of the controller output at the level of 60% of maximum pressure.

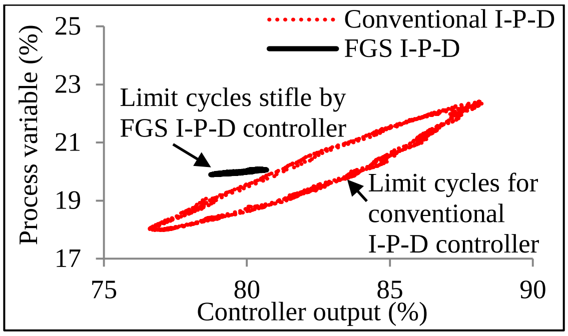

Figure 17. Mapping of the PV and controller output at the level of 20% of maximum pressure.

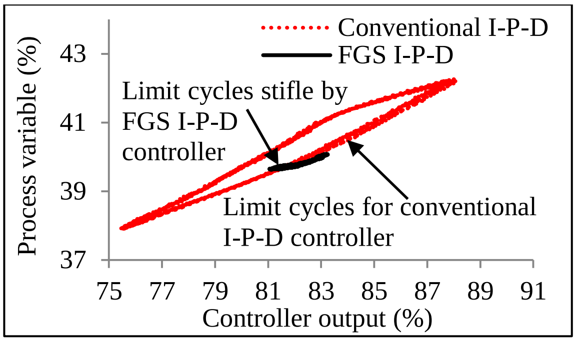

Figure 18. Mapping of the PV and controller output at the level of 20% of maximum pressure.

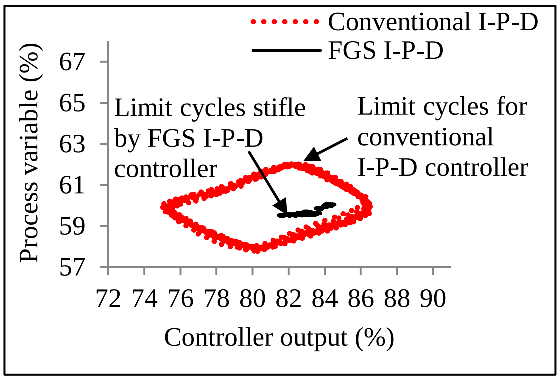

Figure 19. Mapping of the PV and controller output at the level of 20% of maximum pressure.

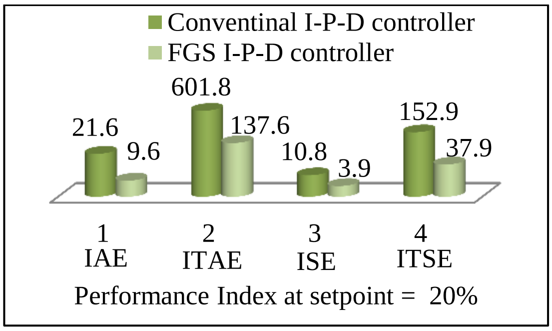

Figure 20. Integral performance indices for 20% of the maximum pressure.

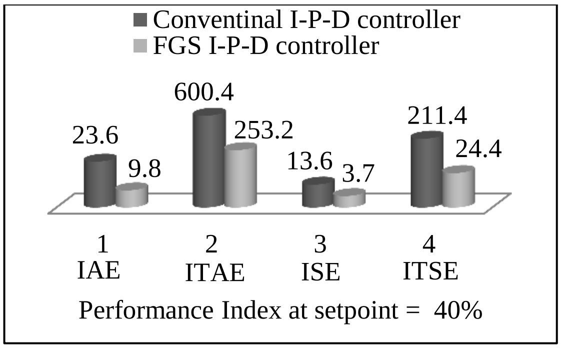

Figure 21. Integral performance indices for 40% of the maximum pressure.

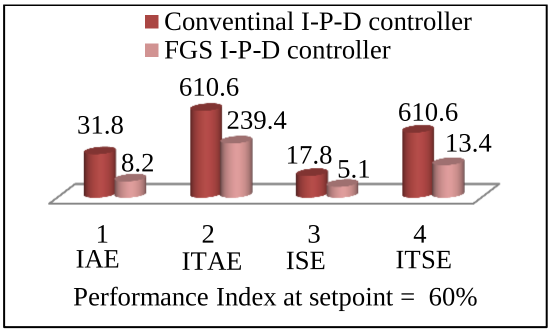

Figure 22. Integral performance indices for 60% of the maximum pressure.

A. Disturbance Rejection

Elimination of the effect of disturbance is a main concern for control engineers while designing the control system. In the industrial environment, external disturbances have a noticeable impact on the closed-loop response of mechanical systems. The effectiveness of the controllers was tested for disturbance rejection (i.e., reducing the undesirable effects of external disturbances on the PV). A step disturbance of 0.25 V (pressure signal) was added to the PV in the closed-loop system (step starts at t = 250 s). The step disturbance rejection response of the PV and OP is shown in Figs. 23 and 24, respectively. The mapping of PV-OP is shown in Fig. 25.

It is clear that the conventional I-P-D controller was incapable of controlling the disturbance, whereas the FGS I-P-D controller effectively mitigated undesirable oscillation. The IAE, ITAE, ISE, and ITSE for disturbance rejection performance were calculated for a duration of 100 s (i.e., from 250 s to 350 s). The results are graphically compared and shown in Fig. 26. All the integral performance indices show a considerable reduction in case of FGS I-P-D controller as compared to the classical I-P-D controller.

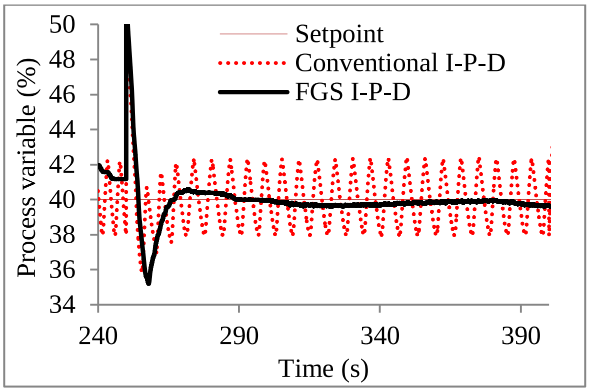

Figure 23. Disturbance rejection response of the PV at the level of 40%.

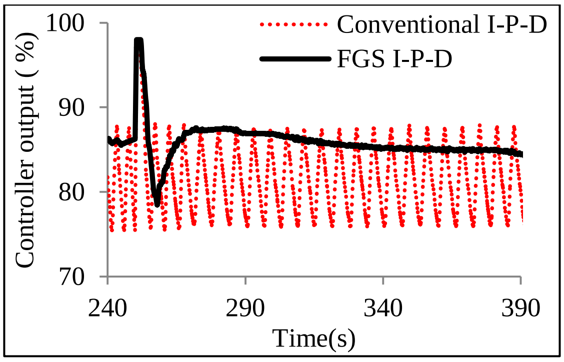

Figure 24. Disturbance rejection response of the controller output at the level of 40%.

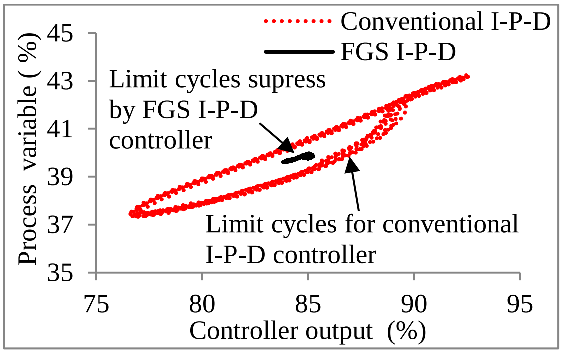

Figure 25. Mapping of the PV and controller output for disturbance rejection at the level of 40%.

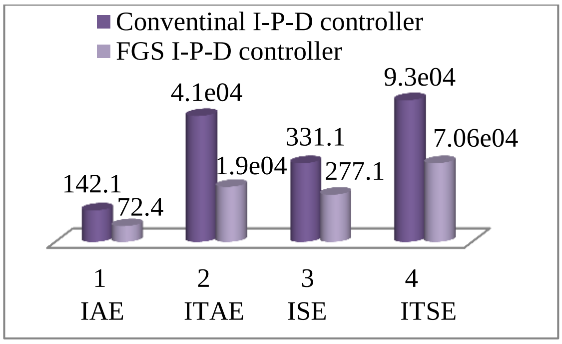

Figure 26. Integral performance indices for disturbance rejection at the level of 40%.

VI. CONCLUSION

In this paper, an FGS I-P-D controller has been proposed. This is an adaptive controller that utilizes a fuzzy logic algorithm to change the gains of the conventional I-P-D controller at runtime. The performance of the FGS I-P-D controller was assessed in a highly nonlinear experimental setup (i.e., a pressure control unit with a sticky pneumatic control valve). The closed-loop control response of a process having a sticky control valve is quite oscillatory and makes the overall process a challenging task for control engineers. A comparative study is performed for both controllers for setpoint tracking and disturbance rejection. Experimental studies revealed that the proposed FGS I-P-D controller demonstrates much better setpoint tracking and disturbance rejection performance and effectively eliminates the stiction-based variation from the PV and controller output at all the considered operating points. On the other hand, the conventional I-P-D controller shows an oscillatory response both for setpoint following and for disturbance rejection. The main advantage of the proposed control technique is that it is a standalone solution and no other compensating device is required in the feedback control loop as reported in the literature.

REFERENCES

[1] Srinivasan, Ranganathan, and Raghunathan Rengaswamy, "Stiction compensation in process control loops: A framework for integrating stiction measure and compensation," *Industrial & engineering chemistry research*, vol. 44, no. 24, pp. 9164-9174, November 2005.

[2] Hägglund, Tore, "A control-loop performance monitor," *Control Engineering Practice*, vol. 3, no. 11, pp. 1543-1551, November 1995.

[3] Thornhill, Nina F., Biao Huang, and H. Zhang, "Detection of multiple oscillations in control loops," *Journal of Process Control* vol. 13, no. 1, pp. 91-100, February 2003.

[4] Choudhury, M.A.A. Shoukat, Mridul Jain, and Sirish L. Shah, "Stiction–definition, modelling, detection and quantification," Journal of Process control, vol. 18, no. 3, pp. 232-243, March 2008.

[5] Hägglund, Tore, "A friction compensator for pneumatic control valves," *Journal of process control*, vol. 1, no. 8, pp. 897-904, December 2002.

[6] Srinivasan, Ranganathan, and Raghunathan Rengaswamy. "Techniques for stiction diagnosis and compensation in process control loops," In American Control Conference, 2006, pp. 6-pp. IEEE, June 2006.

[7] Cuadros, De Souza L., Marco Antônio, Celso J. Munaro, and Saul Munareto, "Novel model-free approach for stiction compensation in control valves," *Industrial & Engineering Chemistry Research,* vol. 51, no. 25, pp. 8465-8476, June 2012.

[8] Xiang Ivan, Lee Zhi, and S. Lakshminarayanan, "A new unified approach to valve stiction quantification and compensation," *Industrial & Engineering Chemistry Research*, vol. 48, no. 7, pp. 3474-3483, March 2009.

[9] Srinivasan, Ranganathan, and Raghunathan Rengaswamy, "Approaches for efficient stiction compensation in process control valves," *Computers & Chemical Engineering*, vol. 32, no. 1, pp. 218-229, Jan uary 2008.

[10] Farenzena, Marcelo, and J. O. Trierweiler, "Modified PI controller for stiction compensation," *IFAC Proceedings,* vol. 43, no. 5, pp. 799-804, January 2005.

[11] Cuadros, Marco Antonio de Souza L., Celso J. Munaro, and Saul Munareto, "Improved stiction compensation in pneumatic control valves," *Computers & Chemical Engineering*, vol. 38, pp. 106-114, March 2012.

[12] Wang, Jiandong, "Closed-loop compensation method for oscillations caused by control valve stiction," *Industrial & Engineering Chemistry Research*, vol. 52, no. 36, pp. 13006-13019, August 2013.

[13] P. Mishra, V. Kumar, and K.P.S Rana, "A novel intelligent controller for combating stiction in pneumatic control valves," *Control Engineering Practice*, vol. 31, no. 33, pp. 94-104, December 2014.

[14] P. Mishra, V. Kumar and K..P.S Rana, "Stiction Combating Intelligent Controller Tuning: A Comparative Study," *in Proceedings of the IEEE International Conference on Advances in Computer Engineering and Applications (ICACEA)*, IMS Ghaziabad, India, March 19-20, 2015, pp. 534-541.

[15] P. Mishra, V. Kumar and K. P. S. Rana, "Intelligent Ratio Control in Presence of Pneumatic Control Valve Stiction," *Arabian Journal for Science and Engineering*, vol. 41, no. 2, pp. 677-89. February 2016.

[16] P. Mishra, V. Kumar and K. P. S. Rana, "An online tuned novel nonlinear PI controller for stiction compensation in pneumatic control valves," *ISA Transactions*, Elsevier, vol. 58, pp. 434-445, September 2015.

[17] P. Rohilla, V. Kumar and B. C. Nakra, "Investigation of Intelligent Control System for Non-Linear Real Time Pressure Control System," *International Journal of Advanced Computer Research* , vol. 5, no.19, pp.212-219, June 2015.

[18] P. Rohilla, V. Kumar, and Feras Al Hakkak, "Fuzzy I+PD controller for stiction compensation in pneumatic control valve," *International Journal of Applied Engineering Research*, vol. 12, no. 13, pp.3566-3575, July 2017.

[19] V. Kumar**,** B. C. Nakra and A.P. Mittal, "A Review of Classical and Fuzzy PID Controllers," *International Journal of Intelligent Control and Systems*, vol. 16, no. 3, pp. 170-181, September 2011.

[20] V. Kumar and A.P. Mittal, "Architecture, Performance and Stability Analysis of a Formula-Based Fuzzy I - Fuzzy P - Fuzzy D controller," *Soft Computing*, Springer Berlin Heidelberg publication, vol. 15, no. 3, pp. 517-531, February 2011.

[21] V. Kumar and K. P. S. Rana, "Some Investigations on Hybrid Fuzzy IPD controllers for Proportional and Derivative Kick Suppression," *International Journal of Automation and Computing*, Springer publication, vol. 13, no. 5, pp. 516-528, October 2016.

[22] Zhao, Zhen-Yu, Masayoshi Tomizuka, and Satoru Isaka, "Fuzzy gain scheduling of PID controllers," *IEEE transactions on systems, man, and cybernetics*, vol. 23, no. 5, pp. 1392-1398. September 1993.

[23] Kanthaphayao, Yutthana, and Viboon Chunkag, "Current-sharing bus and fuzzy gain scheduling of proportional–integral controller to control a parallel-connected AC/DC converter," *Iet Power Electronics*, vol. 7, no. 10, pp. 2525-2532, July 2014.

[24] Costa, Edson BM, and Ginalber LO Serra, " Fuzzy gain scheduling design based on multiobjective particle swarm optimization," In *Computational Intelligence (LA-CCI), 2015 Latin America Congress on*, pp. 1-6. IEEE, October 13, 2015.

[25] Lekshmi, R. R., K. Hareesh, P. Swetha, Vibin Raj, and Anant Narula, "Fuzzy gain schedule based load frequency control of multi area thermal system under deregulated environment," *In Power Electronics, Drives and Energy Systems (PEDES), 2016 IEEE International Conference on*, pp. 1-6. IEEE, December 2016.

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