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# Fuzzy gain scheduling of PID controller for stiction compensation in pneumatic control valve

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**Abstract**

Inherent nonlinearities like Dead-band, stiction, and hysteresis in control valves degenerate plant performance. Valve stiction standouts as a more widely recognized reason for poor execution in control loops. Measurement of valve stiction is essential to maintain scheduling. For industrial scenarios, loss of execution due to nonlinearity in control valves is an imperative issue that should be tackled. Thus, an intelligent technique is required for automated execution, observation, and enhancement. The paper shows the creative utilization of an intelligent controller for nonlinearity diagnosis in control valves. This is a Fuzzy Gain Scheduling (FGS) PID smart controller that tunes its gain parameters in real time to manage a control valve's inherent nonlinearities. The viability of the FGS PID controller is experimentally verified in a laboratory scale plant. An execution comparison between FGS PID and classical PID controllers are undertaken for their setpoint following and disturbance rejection at different operating points. Experimental results show that the FGS PID controller outperforms the classical PID controller for all explored cases effectively managing stiction based oscillation in the controller output.

## 1. Introduction

Pneumatic control valves play a fundamental role in modern industry. These valves consist of two sections, a valve body, and a flexible diaphragm. A diaphragm is used as an actuator, converting pressure into displacement. The attachment comprises a plug and a plug sheet. The function of the valve actuator is to position the plug accurately and thus control fluid flow. Motion in the valve stem involves static and dynamic frictions. As static friction is higher than dynamic friction, moving the stem from a steady state position requires a higher force compared to that required during motion. Friction leads to stiction, Dead-band and hysteresis in pneumatic control valves, which convert these valves into exceptionally nonlinear systems. The nonlinear phenomena in control valves were defined by Choudhury et al. [1] as "Stiction is a property of an element such that its movement in response to a varying input is

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preceded by a static part (Dead-band plus stick-band) followed by a sudden abrupt jump called slip-jump. Its origin in a mechanical system is static friction which exceeds the friction during smooth movement". The input-output characteristic of an air-to-close pneumatic control valve is shown in Fig.1, in which $S_1$ and $S_2$ are stiction bands. There is a variation in $S_1$ and $S_2$ during the stem's upward and downward movements which in turn create a critical problem in plant parameter estimation making it very tedious to model the valve characteristic [2].

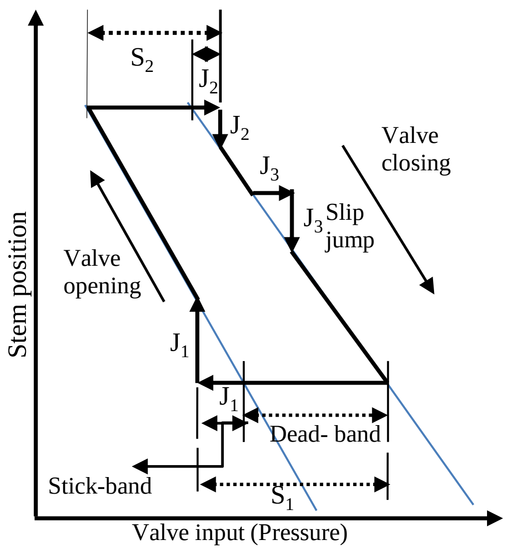


**Fig. 1.** Input-output characteristic of an air-to-close pneumatic control valve [2]

Some potential work on oscillation detection in process variable (PV) and controller output (OP) are as follows:

The nonlinearities in the pneumatic control valve seriously affect the control loop performance by creating oscillations in PV and OP [3]. Hägglund [4] proposed a simple oscillation detection technique based on the integral of absolute error (IAE) of subsequent zero-crossings of control error (e), between the setpoint and controlled variable. Thornhill et al. [5] introduced a regularity index of zero-crossings in the autocorrelation functions to assess loop oscillation, but its accuracy was limited by the manual choice of band pass filters in multiple oscillations. In continuation of this, Matsuo et al. [6] presented an oscillation detection approach with a wavelet transform. Further, Jelali [7] proposed a technique for the detection and estimation of the valve stiction in control loops using least squares and global search algorithms. The control loop oscillation enhanced rejection rates, energy consumption and reduced overall benefit of the plant [8]. Thus, control loop oscillation can be eliminated by the proper maintenance of control valves. But, maintenance work is usually tedious and influences the plant's productivity [9, 10]. Therefore, in order to deal with stiction related problems in the pneumatic control valve, control engineers mainly use two techniques, i.e. compensator along with the existing controller and a standalone control scheme. The related works in this regard are subsequently presented.

Several works reported in the literature for using compensator to deal with stiction related problems are as follows:

Armstrong-Hèlouvry et al. [11] presented a very interesting survey regarding the compensation techniques used to control machines affected by stiction. They proposed stiff proportional-derivative (PD), PD with integral control, impulsive and dithering control techniques. In dithering, a high-frequency signal is introduced into the system to overcome friction. This enhances the controller's performance. But, the pneumatic control valve may filter high-frequency pulses making the technique inconvenient for valves. Kayihan and Francis [12] proposed a local nonlinear controller by utilizing linearization information from the internal model of a control valve. It handles the stiction based on exact information on the plant's parameters. The proposed control strategy was compared with a linear proportional integral (PI) controller in terms of IAE. In any case, the proposed technique required accurate data of the valve or plant parameters, while it is impossible to get exact information about the plant parameters in all cases. Further, Gerry and Ruel [13] suggested reducing the effect of stiction online by retuning the set of rules to replace PI controller to proportional (P) controller using high gain. They proposed a high gain P controller to reduce the impact of hysteresis and stiction in the pneumatic control valve. This technique was not appropriate for an operator to be used in a large plant. Furthermore, Furthermore, Hägglund [14] proposed a "knocker method" where short pulses with constant amplitude, width, and duration are added to the controller to minimize oscillations generate by stiction in the control valve. The proposed controller compared to the linear PI controller to reduce the integral square error (ISE) and IAE by 31% and 55%, respectively. Stiction was compensated at the expense of faster and wider movement in the valve stem thereby increasing the valve's wear rate.

Further, Srinivasan and Rengaswamy [15] developed a "two move approach" to restrain stiction efficiently through the automated choice of compensation parameters. The aggressive movement in the valve stem was reduced by adding some value to the controller through the compensator. It was assumed that the plant was precisely modeled and stiction known, but this is not possible to get exact information about a real plant in all cases. This method requires a precise stiction measurement to conquer stiction. This method is unfit for pneumatic actuators. In line with, Farenzena [16] further improved the "two moves approach" by setting a value in compensator parameters which reveal that closed-loop performance faster than open loop. The proposed method improved setpoint change and disturbance rejection, with variability in PV being observed at setpoint tracking. Furthermore, Cuadros et al. [17] suggested a two moves compensation method to handle valve stiction which needs no prior knowledge of the plant and handles setpoint change by knowing the control error. This method's limitation is the requirement of the valve and process dynamics similar to setting compensator parameters. Their methodology was insufficient to handle the oscillating disturbance in a cascade loop. It has been observed that the compensation technique introduced an additional component to the control scheme which is not well accepted by the industry and in this regard, a standalone control scheme will be always preferred.

Some standalone control schemes reported in the literature to effectively cater the stiction based oscillation in PV and OP are as follows:

Mishra et al. [2, 18] proposed a novel technique using fuzzy logic to combat the stiction in control valve, namely stiction combating intelligent controller (SCIC). It is a variable gain fuzzy PI controller, which effectively suppress the stiction based oscillations from PV and OP and able to exert lesser pressure on stem movement. Further, they proposed another novel method using a conventional PI controller, known as nonlinear PI controller, to deal with the stiction problem in the pneumatic control valve. They validated their proposed technique on pilot plant and found much superior to a conventional controller in every aspect of the study [19, 20]. They further confirmed their findings by efficiently control the ratio on a laboratory scaled plant in the presence of control valve stiction by SCIC controller [21]. There are some other interesting works, regarding stiction compensation, which are reported in the literature [22-24].

Recently, Capaci and Scali presented a very exciting review and performed a comparison of the various techniques available for the valve stiction [25]. Also, Capaci and Scali [26, 27] presented a revised technique and an augmented PID control scheme to compensate the pneumatic control valve stiction. Further, Fang et al. [28] established an analytical relationship between PID controller parameters and oscillation amplitude and period of PV. Based on this, a new compensation technique is proposed to reduce the oscillation amplitude from PV to the desired level. The experimental analysis illustrates the effectiveness of the proposed control scheme.

Available literature shows that the main causes of malfunction in the valves are stiction based nonlinearities, poor controller tuning, and external disturbance. According to the literature, an additional component is required to make up for stiction. In any case, extra parts increase external disturbance, nonlinearity, and total plant cost. Nonlinearities, which are mathematically difficult to express, can be managed through a fuzzy logic controller (FLC) [29, 30]. FLC can deal with real-time problems effectively that cannot be taken care of by conventional controllers [31]. Normally, the FLC has membership functions (MFs), a set of rules and scaling gains [30]. Also, FLC is quite often used with a conventional PID controller for runtime updating its parameters which make it adaptive in nature [32, 33].

In the present work, a fuzzy gain scheduling (FGS) PID controller is implemented in a laboratory scaled pressure process to conquer stiction based oscillations in the control loop and to minimize the effects of Dead-band and hysteresis error. The proposed method is based on fuzzy logic where PID controller gains are updated at runtime by an FGS algorithm. This technique does not require any additional component and overcomes issues of poor controller tuning. Experimental results revealed that the FGS PID controller outperformed the traditional PID controller in all aspects of the study.

## 2. Plant description and problem formulation

In the present work, a laboratory scaled pressure control system is used for practical examination of the control algorithm. The snapshot of the experimental setup used for the current study is shown in Fig. 2. In this setup, two manual control valves ($V_3$ and $V_5$) are used on the inlet and outlet of a 5-liter stainless steel process tank to control air flow. In addition, a pneumatic control valve is utilized to dynamically control the process tank's pressure. A pressure sensor (Piezo-resistive type) converts the pressure (0-2 bars) into the voltage (0-2.5 Volts). A reciprocating air compressor circulates the compressed air in the pipes. Fig. 3 shows the schematic diagram of a closed loop pressure control system. A National Instrument™ (NI USB-6008) DAQ is used for data acquisition and interfacing with the computer. The controllers are implemented in a Matlab/Simulink environment, and fourth-order Runge-Kutta ODE solver with a sampling rate of 0.01s is used. Table 1 shows the different equipment used in this work.

### *2.1. Problem formulation*

The pneumatic control valve is used as a final element to control flow in the pipes. Dead-band and hysteresis error in the control valve are measured by forward (valve closing) and reverse (valve opening) movement of the valve stem on the application of the input voltage signal (0-2.5 V) from 0% to 100% and afterward from 100% to 0%, as shown in Fig. 4. Information on stiction band in the control valve is acquired by applying a gradually changing sinusoidal voltage signal $V_i(t)$ to the plant in an open loop. Initially, when the valve is fully closed, a sinusoidal voltage signal is applied to the plant, due to which the movement in the valve stem takes place and then opens gradually and closes with varying voltage signals, i.e. 2.5 volt to 0 volt and vice-versa. The normalized valve input and the corresponding normalized pressure output in time trends are shown in Fig. 5, in which when the valve is 100% closed (point A) pressure remains constant for about 10s. Then a steep rise is visible after valve input reaches 70% (point B), making a stiction band of $S_1$= 30%. Similarly, at point C where the valve input is 0 %, the valve remains stuck until valve input reaches 50% (point D) making a stick-band of $S_2$ = 50%. The explanation for the variable stiction size is the impact of potential energy which is put away in the stem at various valve positions in addition to the erosion among the stem and packing. The stiction affects the control loop behavior by introducing a non-sinusoidal oscillation in the process variable (PV) and controller output (OP).

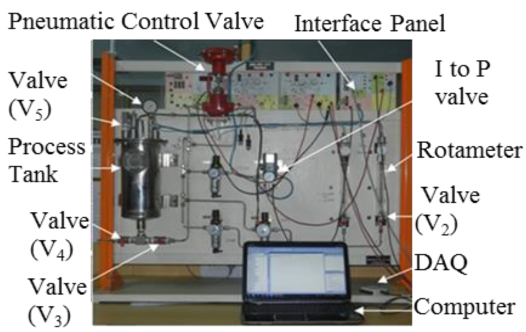


**Fig. 2.** Snapshot of the experimental setup.

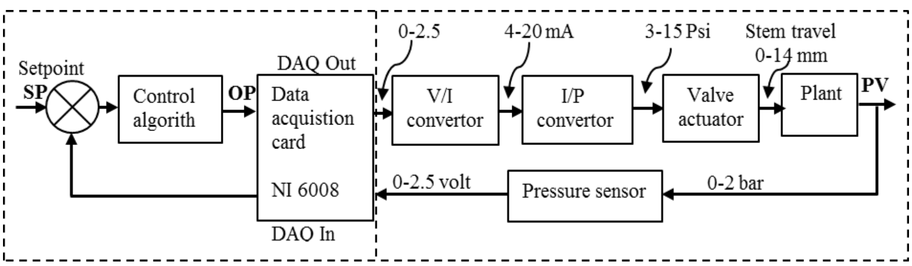


**Fig. 3.** Block diagram of closed-loop pressure control system.

**Table. 1.** Details of various equipment used in plant

| Equipment Name | Specification | Quantity |
|---|---|---|
| Piezo-resistive type pressure sensor | Converts 0-2 bar to 0-2.5 volts | 1 |
| Pneumatic control valve | Equal percentage characteristics,<br>Action: air-to-close, ½ inch port size | 1 |
| Voltage to current (V/I) converter | Converts 0-2.5 volt to 4-20 mA | 1 |
| Current to pressure (I/P) converter | Converts 4-20 mA to 3-15 psi | 1 |
| Single stage reciprocating air compressor | Working pressure 40-100 psi<br>Nominal power 1.5 kW/ 2 HP, 1440 RPM, Voltage 190/240<br>Horizontal tank, cast iron cylinder and crankshaft | 1 |
| Data acquisition card | National instruments™<br>USB-6008 DAQ card, 8 analog inputs at 12 bits,<br>Maximum sampling rate: 48 kS/s ,<br>2 static analog outputs (12-bit), 12 digital I/O;<br>32-bit counter, Range: -10 v to 10 v. | 1 |
| Personal computer | Intel Core™ i7 processor, 2.5 GHz, 4GB RAM<br>Equipped with *MATLAB/Simulink*® version *2011b* software | 1 |

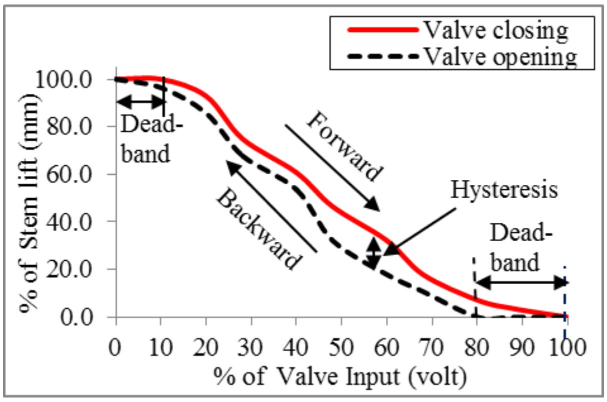


**Fig. 4.** Hysteresis and Dead-band in the control valve

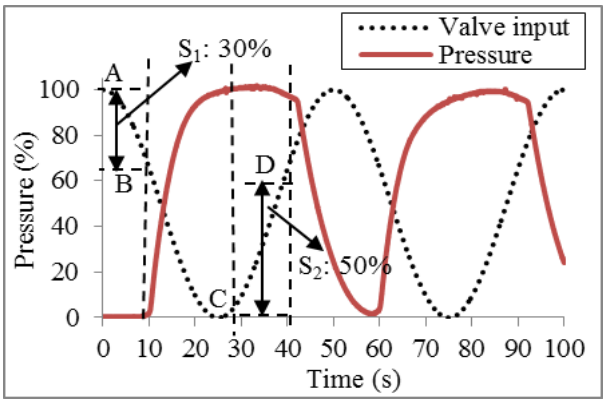


**Fig. 5.** Stick-band in the control valve

A limit cycle is observed in PV-OP graph (Fig. 6). A limit cycle is observed in PV-OP graph (Fig. 8) showing stiction as an elliptical pattern. This proves that the valve is sticky in nature, and it enhances nonlinearity in the process. Pneumatic control valves exhibit inherent nonlinearities like Dead-band, stiction, and hysteresis. These nonlinearities may reduce the lifespan of the control valve and increase maintenance cost and conversely reduce the benefits of the plant. According to the literature review, the PI controller is unable to handle stiction well, and hence an intelligent controller is required to remove the sticky effect.

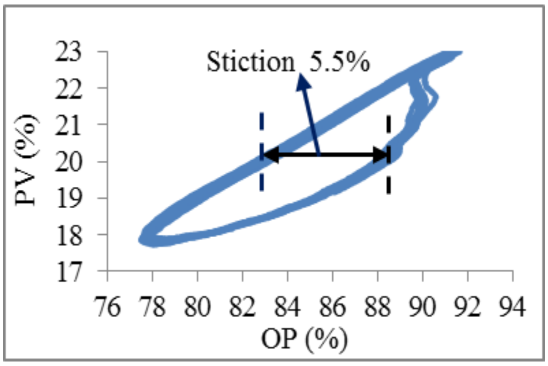


**Fig. 6.** Stiction in control valve shows elliptical pattern in PV-OP plot.

## 3. Design of Fuzzy gain scheduling PID controller

Conventional PID is a powerful controller, which is simple to design and has a straightforward structure. Hence, it is the controller of choice for most industrial processes. In the time domain, a classical PID controller can be defined as follows:

$$U_{pid}(t) = K_p e(t) + K_i \int e(t) + K_d \frac{de}{dt} \qquad (1)$$

where $e(t)$ is the system error, $U_{pid}(t)$ is the control action and $K_p$, $K_{i,}$ and $K_d$ are proportional, integral and derivative gains, respectively. Controller performance parameters like setpoint following, load disturbance attenuation, and robustness depend on the tuning of the above parameters.

### *3.1. Ziegler-Nichols (ZN) tuned PID controller*

Ziegler-Nichols methods [34] are commonly used to adjust the gains of PID controllers. In the present work, the ZN oscillation method is used to obtain tuning parameters. The ultimate gain $K_{cr}$ and oscillation period $P_{cr}$ are obtained by setting the integral and derivative gains at zero. The value of proportional gain $K_p$ is increased (from zero) till it reaches the value of the ultimate gain, $K_{cr} = 4$, and oscillation period, $P_{cr} = 3.83$ s. Based on the values of $K_{cr}$ and $P_{cr}$, gain parameters of the ZN PID controller are $K_p = 2.5$, $K_i = 1.253$, and $K_d = 1.149$, respectively.

### *3.2. FGS PID controller*

The ZN PID controllers behave well in load disturbance attenuation but produce a large overshoot and oscillation in step response due to the presence of stiction, Dead-band, and hysteresis in the control valve. To eliminate oscillation and enhance setpoint tracking, an amendment to PID parameters is needed. This is achieved with an intelligent technique like fuzzy logic. Hence, this paper uses a self-tuning fuzzy inference system to tune three gain parameters $K_p$, $K_{i,}$ and $K_d$ of the PID controller at runtime. Fig. 7 shows the structure of the FGS PID controller, where e(t) and er(t) are error and rate of change of error between the desired set value r(t) and plant output y(t), respectively. $K_1$ and $K_2$ are gain parameters of the FGS PID controller.

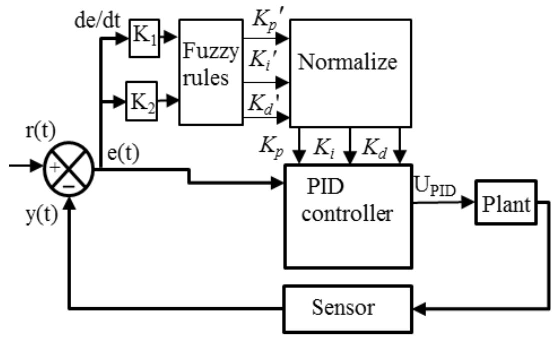


**Fig. 7.** Structure of fuzzy gain scheduling PID controller.

The scaling factors of the PID controller ($K_P$, $K_i$, $K_d$) are determined based on the current error and rate of change of error [32, 33]. It is also assumed that these scaling factors are in the prescribed ranges of [$K_{P,min}$, $K_{P,max}$], [$K_{i,min}$, $K_{i,max}$], and [$K_{d,min}$, $K_{d,max}$], respectively. The appropriate ranges of these scaling factors are computed experimentally based on the ZN oscillation method, $K_P = [0.5,\ 2.5]$, $K_i$ = [0.05, 0.2], and $K_d$ = [0.5, 1.15]. For the sake of convenience, these scaling factors are normalized into a range defined as [0, 1] and by the following linear transformation equations:

$$K_p{}' = \frac{K_p - K_p\,min}{K_p\,max - K_p\,min} = \frac{K_p - 0.5}{2.5 - 0.5}\ ,\quad K_p = 2K_p{}' + 0.5 \qquad (2)$$

$$K_i{}' = \frac{K_i - K_i\,min}{K_i\,max - K_i\,min} = \frac{K_i - 0.05}{0.2 - 0.05},\quad K_i = 0.15K_i{}'.05 \qquad (3)$$

$$K_d{}' = \frac{K_d - K_d\,min}{K_d\,max - K_d\,min} = \frac{K_d - 0.05}{1.15 - 0.5},\quad K_d = 1.1K_d{}'0.5 \qquad (4)$$

For input variables, i.e. the error signal and rate of change of error, signal Gaussian MFs is considered and shown in Fig. 8. Their linguistic values are assigned as Negative Big (NB), Negative (N), Zero (Z), Positive (P), and Positive Big (PB). The ranges of these inputs are from -1 to 1. Also, for output variables $K_p{}'$, $K_i{}'$ and $K_d{}'$, the Gaussian MFs is adopted and shown in Fig. 9. The linguistic levels of these outputs are given as Small (S), Medium Small (MS), Medium (M), Large Medium (LM), and Large (L) ranging from 0 to 1.

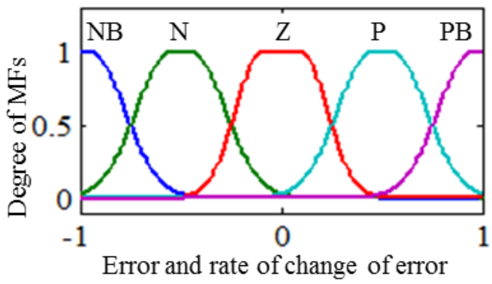


**Fig. 8.** Input MFs for e(t) and er(t).

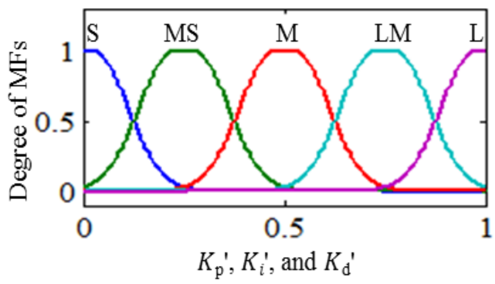


**Fig. 9.** Output MFs for Kp′, Ki′and Kd′.

The 25 fuzzy rules of the FGS PID controller are tabulated in Table 2. The rows represent various linguistic values for error e(t), and the columns indicate values of the rate of change of error er(t). FGS PID controller gain parameters ($K_1$ = 0.8 and $K_2$ = 0.6) are tuned manually. Fuzzy control rules ensure that the pressure inside the process tank at set level is without oscillation. Mamdani Min-Max inference method analyses the output from each rule, and the centroid method is used for defuzzification. The Integral of time absolute error (ITAE) method measured the performance of the closed-loop system.

## 4. Results and discussion

To evaluate the effectiveness of the FGS PID controller as compared to the ZN PID controller, real-time experiments are carried out at various operating points. Controller performance parameters for evaluation are setpoint following, disturbance rejection, and squashing stiction based oscillations. The proportional ($K_p$), integral ($K_i$) and derivative ($K_d$) gains of the PID controller are tuned by the ZN oscillation method at setpoint (SP) = 40% of maximum pressure.

The tuned values of $K_p$, $K_{i,}$ and $K_d$ are 2.5, 1.253, and 1.149, respectively. The range for $K_p$, $K_{i,}$ and $K_d$ in the FGS PID controller is kept the same as those of the ZN PID controller. The robustness of the FGS PID controller for the setpoint following and disturbance rejection are tested at three distinct operating points, i.e. 20%, 40%, and 60% of maximum pressure.

Fig. 10 shows the time trends response of the PV of ZN PID (continuous line) and FGS PID (dotted line) controller for SP = 40%. The corresponding time trends OP is shown in Fig. 11. The output response clearly shows that the ZN PID controller reveals stiction based oscillations on the PV and OP, whereas the FGS PID controller suppresses stiction based oscillations and diligently follows the setpoint.

**Table. 2.** Fuzzy control Rule for Kp′, Ki′and Kd′.

| **e(t)** | **er(t)** | | | | |
|---|---|---|---|---|---|
| | **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\ LM\ LM | MS\ LM\ M | MS\ LM\ M | MS\ L\ L |
| **PB** | M\ M\ L | MS\ LM\ L | MS\ LM\ LM | S\ L\ LM | S\ L\ L |

The mapping between PV-OP is shown in Fig. 12. The PV-OP plot of the ZN PID controller shows an elliptical loop showing stiction, whereas the FGS PID controller conquers stiction without an elliptical loop at SP = 40%. Data for this figure is collected after the transient response, and a total of 5000 samples are used to show the elliptical loop. The performance of the FGS PID controller is also checked on operating points rather than 40% (the point at which the controller is originally tuned). The behavior of the ZN PID and FGS PID controllers for setpoint following at SP = 20% are shown in Fig. 13, and the corresponding controller output is shown in Fig. 14, whereas the PV-OP mapping is seen in Fig. 15.

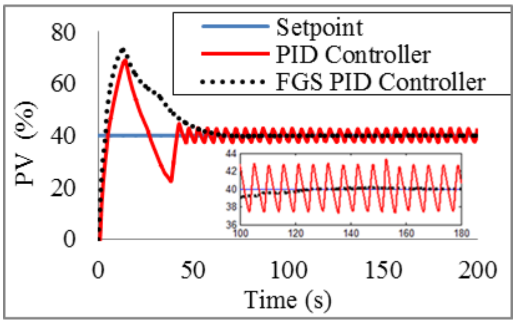


**Fig. 10.** Fluctuation in PV at SP = 40%.

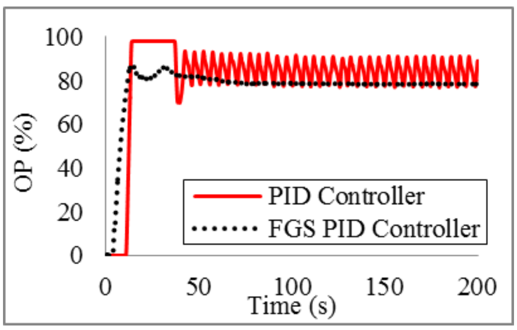


**Fig. 11.** Fluctuation in OP at SP = 40%.

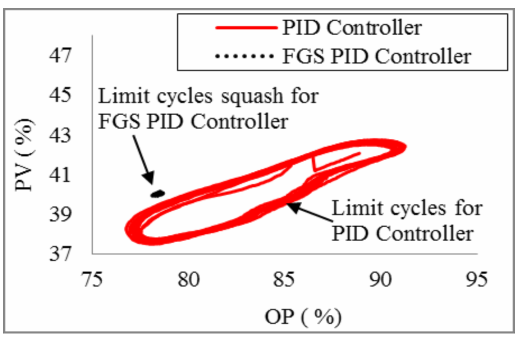


**Fig. 12.** PV-OP mapping at SP = 40%.

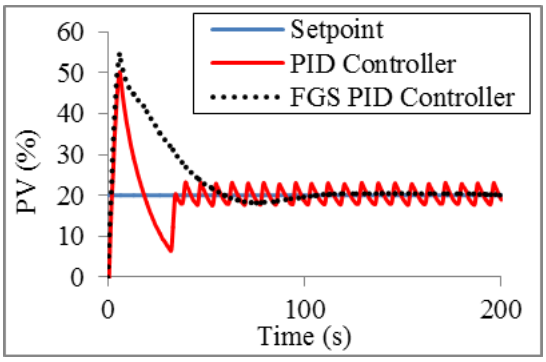


**Fig. 13.** Fluctuation in PV at SP = 20%.

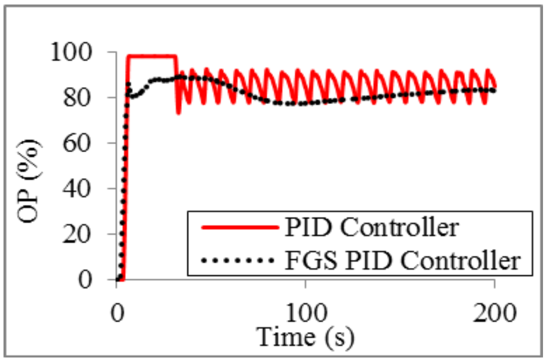


**Fig. 14.** Fluctuation in OP at SP = 20%.

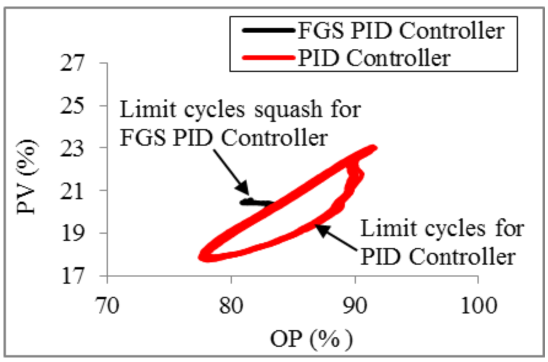


**Fig. 15.** PV-OP mapping at SP = 20%.

The behavior of the ZN PID and FGS PID controller for PV and OP at SP = 60% are shown in Figs. 16 and 17, respectively. The PV-OP mapping is shown in Fig. 18. The results clearly show that, the FGS PID outperformed the ZN PID controller at all operating points to mitigate stiction oscillation in PV-OP and disturbance rejection. The FGS PID controller is robust at all operating points once gain parameters are tuned with there being no need to tune them again. The ITAE is another performance parameter to compare both controllers. This parameter is calculated for 150 s, i.e. from 50 to 200 s. Fig. 19 shows the ITAE for both controllers at different setpoint.

*4.1 Disturbance rejection*

The performance of the controller is also tested for disturbance rejection and setpoint following at distinct operating points, i.e. 20%, 40% and 60% of maximum pressure. To provide disturbance, a simulated disturbance of 0.25 V step input is given at t = 250 s. The time trend responses of PV and OP at SP = 40% are shown in Figs. 22 and 23, respectively and correspondingly the mapping between PV-OP is presented in Fig. 24. The ITAE for disturbance rejection at three distinct SP is shown in Fig. 25 and these are calculated for 100 s after the disturbance (starting from t =150 s). The figures reveal that the FGS PID controller is superior to the ZN PID controller in controlling the disturbances.

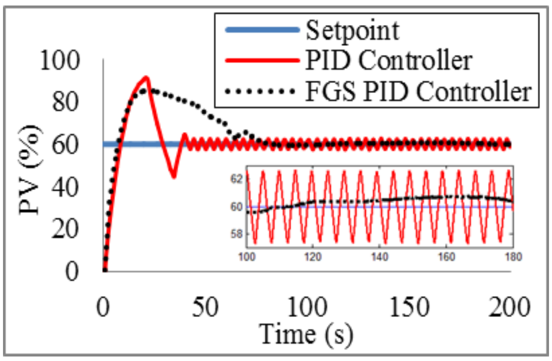


**Fig. 16.** Fluctuation in PV at SP = 60%.

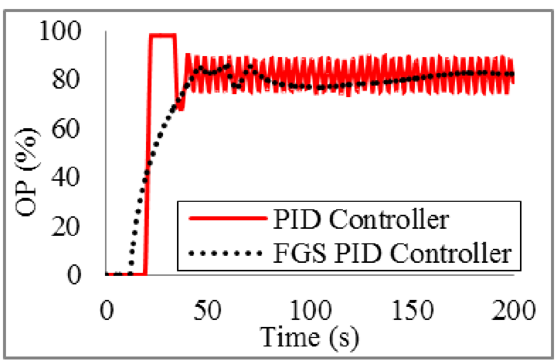


**Fig. 17.** Fluctuation in OP at SP = 60%.

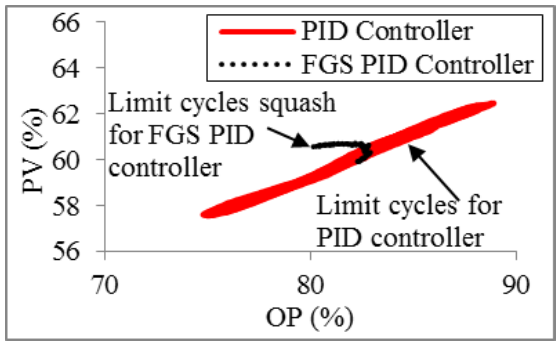


**Fig. 18.** PV-OP mapping at SP = 60%.

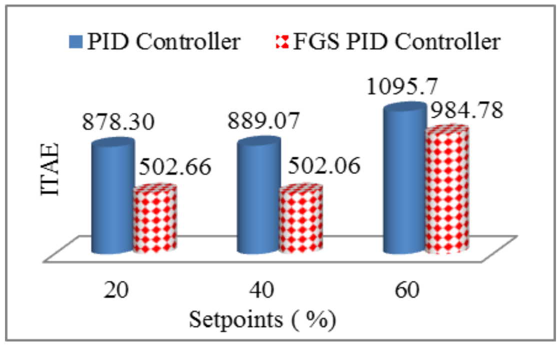


**Fig. 19.** ITAE at different setpoints.

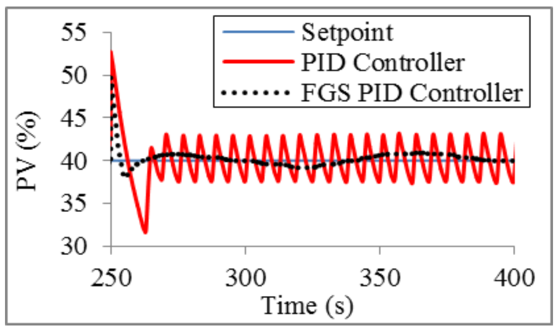


**Fig. 20.** Fluctuation in PV at SP = 40% for disturbance rejection.

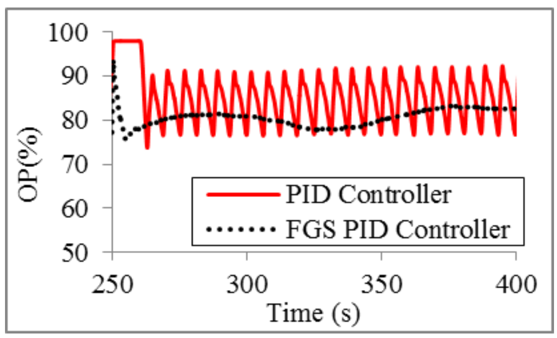


**Fig. 21.** Fluctuation in OP at SP = 40% for disturbance rejection.

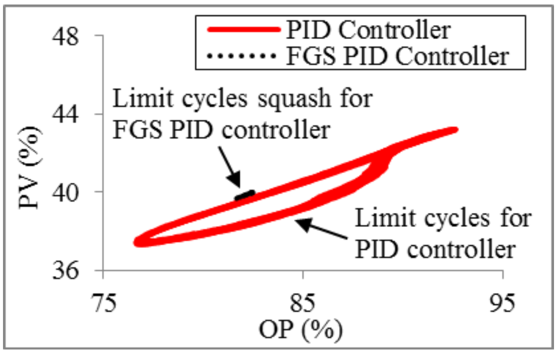


**Fig. 22.** PV-OP mapping at SP = 40% for disturbance rejection.

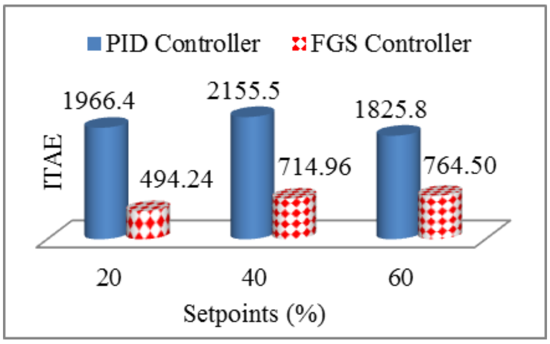


**Fig. 23.** ITAE at different setpoints for disturbance rejection.

## 5. Conclusions

Pneumatic control valves exhibit various nonlinearities like Dead-band, hysteresis, and stiction which affect controller performance and produce oscillations in the process variables and controller output. Control system performance has a major role in the process industry as poor performance impressively diminishes the plant's profitability. In this paper, a simple, robust fuzzy gain scheduling PID controller is experimented on to overcome the stiction influence in the control loop. It is a Mamdani based fuzzy controller. Controller gain is tuned online by fuzzy rules defined on the error and rate of change of error which detract the decay effect of the pneumatic control valve. The performance of the FGS PID controller is compared with a conventional PID controller regarding many indexes like ITAE, setpoint tracking, and disturbance rejection. The linear PID controller is tuned by the popular ZN oscillation method, and the same gains are used in the FGS PID controller. The conventional PID controller is unable to handle innate stiction oscillation, whereas FGS PID controller squashes stiction based oscillation and limits the cycles well in PV and OP at all operating points. The FGS PID controller performs well in volatile environments for setpoint tracking and disturbance rejection. The FGS PID controller does not use additional components to deal with stiction based oscillation, and hence reduces costs and is more robust. The FGS PID controller produces an extremely smooth motion, preventing untimely valve wear, which is able to lower maintenance cost and enhance the plant's profitability.

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