Improving the pneumatic control valve performance using a PID controller

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Abstract: Pneumatic control valves are still the most used in process industries due to their low cost and simplicity. This paper presents a design procedure of a PID controller for a pneumatic control valve. For comparison, P and PI controllers are also utilized for the control valve. The bond graph method is used to model the control valve, in order to compare the response characteristics of the valve. Simulation results are found for three controllers of the valve. The integral time absolute error criterion is used to calculate the controller parameters. The results show excellent closed-loop performance and the PID controller is more robust than the P and PI controller.

Key words: Pneumatic, control valve, PID controller, bond graph

1. Introduction

Process plants consist of hundreds or even thousands of control loops all networked together to produce a product to be offered for sale. Each of these control loops is designed to keep some important process variable, such as pressure, flow, level, or temperature, within a required operating range to ensure the quality of the end product. Each of these loops receives and internally creates disturbances that detrimentally affect the process variable, and interaction with other loops in the network provides disturbances that influence the process variable. To reduce the effect of these load disturbances, sensors and transmitters collect information about the process variable and its relationship at some desired set point. A controller then processes this information and decides what must be done to get the process variable back to where it should be after a load disturbance occurs. When all the measuring, comparing, and calculating are done, some type of final control element must implement the strategy selected by the controller. The most common final control element in process control industries is the control valve. The control valve manipulates a flowing fluid, such as gas, steam, water, or chemical compounds, to compensate for the load disturbance and keep the regulated process variable as close as possible to the desired set point. Control valves adjust the temperature, pressure, flow rate, etc. by changing the flow rate. Figure 1 shows a sliding-stem pneumatic control valve. Pneumatic control valves are still the most used valves in the process industries due to their low cost and simplicity. Pneumatic valves are used extensively in various industries today. Industry standards have been established with details for the vibration, humidity, thermal, salt spray, and temperature extremes that these valves must operate within. This makes the design of valve control systems a very challenging task. Control valves have two major components, the valve body housing and the actuation unit. Hägglund presented a procedure that compensates for static friction (stiction) in pneumatic control valves [1]. The compensation is obtained by adding pulses to the control signal. The characteristics of

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the pulses are determined from the control action. The compensator is implemented in industrial controllers and control systems, and the industrial experiences show that the procedure reduces the control error during stick-slip motion significantly compared to standard control without stiction compensation. The oscillations caused by stiction in pneumatic control valves cause losses in quality and expense of raw materials. The input–output behavior of a pneumatic control valve is affected by stiction in valve. De Souza et al. [2] presented a well-known stiction compensation method that reduces variability both at process variable and pneumatic valve stem movement. The two-move method is revisited in this paper and it is shown that assumptions on the knowledge of steady-state stem position of the control valve that assures equality of the set point and the controlled variable is not easily achievable. One factor in the quality of the final end product is the improvement of the control loop performance. A critical component in the loop is the final control element, the control valve package. Optimized actuator parameters play a vital role in the dynamic performance of the pneumatic control valve. Champagne and Boyle [3] reviewed the pneumatic actuator and positioner parameters that affect the control package performance. This was done through the use of a control valve package computer model to assess the dynamic performance. The attributes of spring return versus double acting actuators were illustrated. The effects of supply pressure, step size, load margin, flow, actuator volume, and design style were investigated through the use of mathematical simulations of pneumatic control valve dynamic performance. A bond graph is a graphical representation of a physical dynamic system. It is similar to the better known block diagram and signal flow, with the major difference that the arcs in bond graphs represent bidirectional exchange of physical energy, while those in block diagrams and signal-flow graphs represent unidirectional flow of information. Bond graphs are also multidomain and domain-neutral. This means that a bond graph can incorporate multiple domains simultaneously. The fundamental idea of a bond graph is that power is transmitted between connected components by a combination of 'effort' and 'flow' (generalized effort and generalized flow). Bond graphs were devised by Paynter [4] at MIT in April 1959 and subsequently developed into a methodology together with Karnopp et al. [5]. Early prominent promoters of bond graph modeling techniques, among others, included Thoma [6]. They contributed substantially to the dissemination of bond graph modeling in Europe, Australia, Japan, China, and India. Athanasatos and Costopoulos [7] used the bond graph method for finding the proactive fault in a 4/3-way direction control valve of a high pressure hydraulic system. The accuracy of the bond graph model was verified by comparing its response to the response of an actual hydraulic system. Zuccarini et al. [8] utilized the bond graph as a boundary condition for a detailed model of an idealized mitral valve. A specifc application in cardiovascular modeling was demonstrated by focusing on a specifc example: a 3D model of the mitral valve coupled to a lumped parameter model of the left ventricle. Heidari and Homaei [9] presented a quadratic optimal regulator for pneumatic control valve. They used a regulator for pneumatic control valves using the pole-placement method, optimal control, full-order state observer, and minimum-order state observer and their responses were compared with each other. Heidari and Homaei [10] also used a recurrent neural network to control the stem of a sliding-stem pneumatic control valve. The bond graph method has been used to model the actuator of a control valve in order to compare the response characteristics of the valve.

In this study, a sliding-stem pneumatic control valve is modeled by the bond graph method. Several control schemes are then used for control of the flow rate of the valve in order to compare the response characteristics of these different schemes. This research is organized as follows: Section 2 recalls the bond graph model of the valve and proposes equations for motion of valve. Section 3 develops the PID controller. Section 4 presents some results of this study, and finally the conclusions of this work are given.
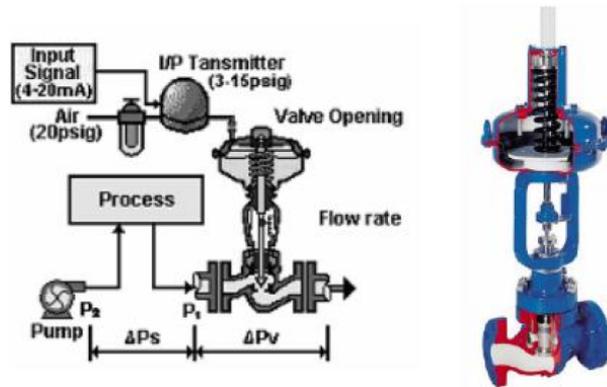
Figure 1. Pneumatic control valve and brief description of process control.

2. Bond graph model of valve and equations

The bond graph model of the valve is shown in Figure 2. In this model, SE is the inlet pressure of the system. The pressure changes to force by multiplying in the effect area of the diaphragm. In the bond graph, this transformer is modeled by TF. Element R is the friction of the system.
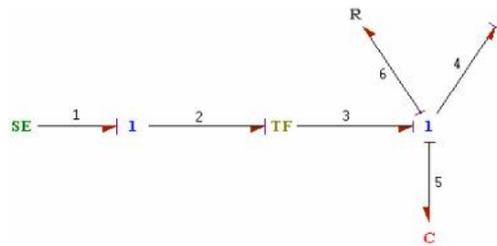

Figure 2. Bond graph control valve actuator.

Element I is the movable mass of the valve and diaphragm. Element C represents the spring of the valve actuator, and the 1-junction is a common flow junction. The 1-junctions have equality of flows and the efforts sum up to zero with the same power orientation. In fact, junctions can connect two or more bonds. The direction of the half arrows ($\rightharpoonup$) denotes the direction of power flow given by the product of the effort and flow variables associated with the power bond. The bonds in a bond graph may be numbered sequentially using integers starting with 1. The two 1-junctions in the bond graph shown can be uniquely identified as (S 1 2) and (S 4 5 6); similarly, symbols like SE_1, R_6 can be used to identify a particular element. This system has two state variables, P_4 and q_5. q_5 is the displacement of the valve stem and the variation of the spring length. Also, $v_4 = \frac{P_4}{I_4}$ is the velocity of the valve stem. The equations of motion are derived using the bond graph method as below:

$$\dot{P}_4 = A \times SE_1 - K_3 q_5 - \frac{R_6}{I_4} P_4, \tag{1}$$

$$\dot{q}_5 = \frac{P_4}{I_4}. \tag{2}$$

Now, if the velocity and position of stem are zero in the initial condition, *X(0)*, then we have:

$$X(0) = \left[\begin{array}{c} P_4(0) \\ q_5(0) \end{array} \right] = \left[\begin{array}{c} 0 \\ 0 \end{array} \right].$$

For finding the transfer function of the valve, by derivation of Eq. (2) with respect to time we have:

$$\ddot{q}_5 = \frac{\dot{P}_4}{I_4}. \tag{3}$$

By substitution of $\dot{P}_4$ from Eq. (1) into Eq. (3):

$$\ddot{q}_5 = \frac{A \times SE_1 - K_3 q_5 - \frac{R_6}{I_4} P_4}{I_4}. \tag{4}$$

By substitution of P_4 from Eq. (2) in to Eq. (4):

$$\ddot{q}_5 = \frac{1}{I_4}(A \times SE_1 - K_3 q_5 - R_6 \dot{q}_5). \tag{5}$$

Using Laplace transformation of Eq. (5), we have:

$$\frac{q_5(s)}{SE_1(s)} = \frac{(A/I_4)}{s^2 + (\frac{R_6}{I_4})s + (\frac{K_3}{I_4})}. \tag{6}$$

Eq. (6) is the transfer function of the valve. The results of the bond graph model of the valve show that the response of the system is identical to the results in [9–12].

3. PID controller

The PID control action can be expressed in the time domain as:

$$u(t) = K_P e(t) + \frac{K_P}{T_i}\int_0^t e(t)dt + K_P T_d \frac{de(t)}{dt}, \tag{7}$$

where $u(t)$ and $e(t)$ are defined as the controller output and error signal, respectively, and T_i and T_d are the integral time and derivative time constants, respectively. Taking the Laplace transform from Eq. (7) yields:

$$U(s) = K_P E(s) + \frac{K_P}{T_i s}E(s) + s K_P T_d E(s). \tag{8}$$

The resulting PID controller transfer function is:

$$\frac{U(s)}{E(s)} = K_P(1 + \frac{1}{T_i s} + T_d s). \tag{9}$$

K_P is called the proportional gain. K_I, the integral gain, and K_D, the derivative gain, can be defined as follows:

$$K_I = \frac{K_P}{T_i}, \qquad K_D = K_P T_d. \tag{10}$$

A performance index can be calculated or measured and used to evaluate the system's performance. A quantitative measure of the performance of a system is necessary for the operation of modern adaptive control systems, for automatic parameter optimization of a control system, and for the design of optimal systems. A

system is considered an optimum control system when the system parameters are adjusted so that the index reaches an extreme, commonly a minimum value. To reduce the contribution of the large initial error to the value of the performance integral, as well as to emphasize errors occurring later in the response, the following index has been proposed [9]:

$$ITAE = \int_0^\infty t\,|e(t)|dt, \tag{11}$$

where t is time and $e(t)$ is absolute error. According to this criterion, the integral time absolute error (ITAE) criterion is the one that minimizes the performance index that has been given in Eq. (11). A system designed by use of the ITAE criterion has the characteristic that the overshoot in the transient response is small and oscillations are well damped [13].

4. Results and discussion

Table 1 shows the parameters of a sliding-stem pneumatic control valve.

Table 1. Valve parameters [10].

Name of variable	Parameter	Value
Effective area of diaphragm	A	0.196 ft^2
Spring constant	K	6790
Movable mass	I (M)	0.03 slug
Resistance and friction coefficient	R	1 lb S/ft
Air pressure	SE	140 lb/ft^2

By substitution of Table 1 in Eq. (6), we have:

$$\frac{q_5(s)}{SE_1(s)} = \frac{6.53}{s^2 + 33.33s + 226333.3}.$$

Matrices of the state space equations of the valve are as follows:

$$A = \begin{bmatrix} 0 & 1 \\ -226333.33 & -33.33 \end{bmatrix}, B = \begin{bmatrix} 0 \\ 914.2 \end{bmatrix}, C = \begin{bmatrix} 1 & 0 \end{bmatrix}, \text{ and } D = 0,$$

where A is called the system matrix, B the input matrix, C the output matrix, and D the direct transmission matrix.

The performance of the proposed control scheme is illustrated in this section through a series of simulations. PID control is perhaps the most widely used control method. It can provide fast response, good system stability, and small steady-state errors in linear systems with known parameters. The simulation results are presented below. They illustrate the effects of different controllers on the performance of the control systems.

We would like to design the P, PI, and PID controllers such that in the unit step response the maximum overshoot is less than 10% and the settling time is about 0.5 s.

Figure 3 shows the results of the step response of the valve stem without any controller. Here, the overshoot is very big and the settling time is 0.178 s. The output has an overshoot of less than 901% and the rise time is 0.00337 s.

Using the P controller for the valve system, the results of the overshoot and settling time for the closed loop system are shown in Figure 4. The best gain parameter is 1.012. The overshoot is very big at 89.6%. The settling and rise times are 0.178 and 0.00337 s, respectively. Thus, the P controller is not suitable for this valve.
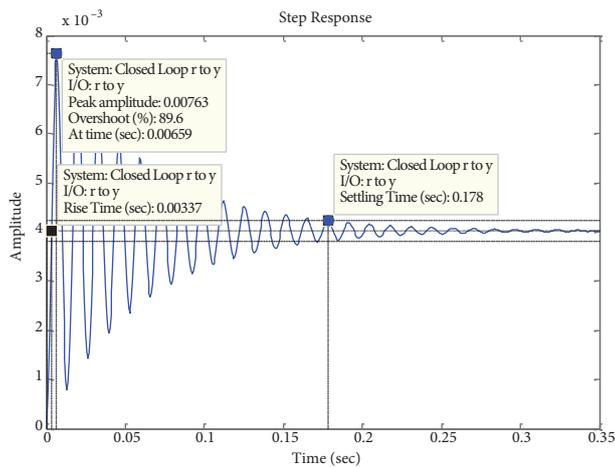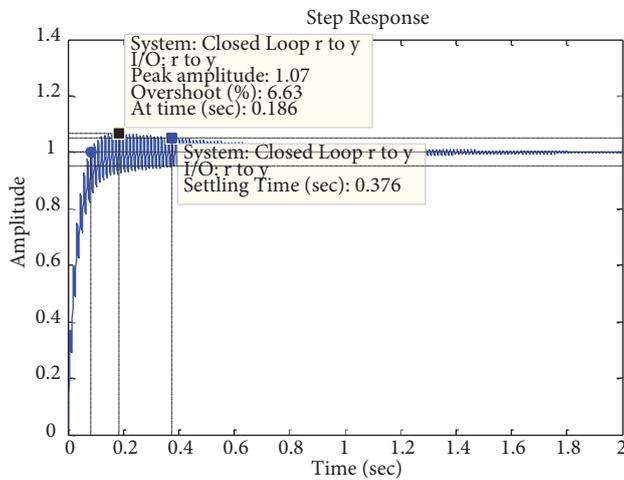
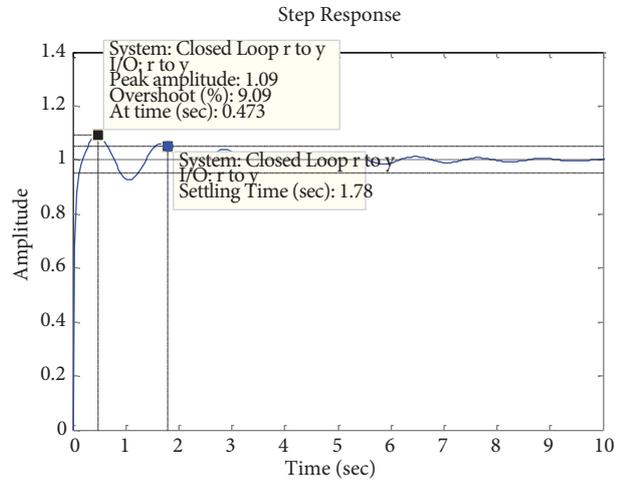**Figure 3.** The unit step response of valve without any controller.
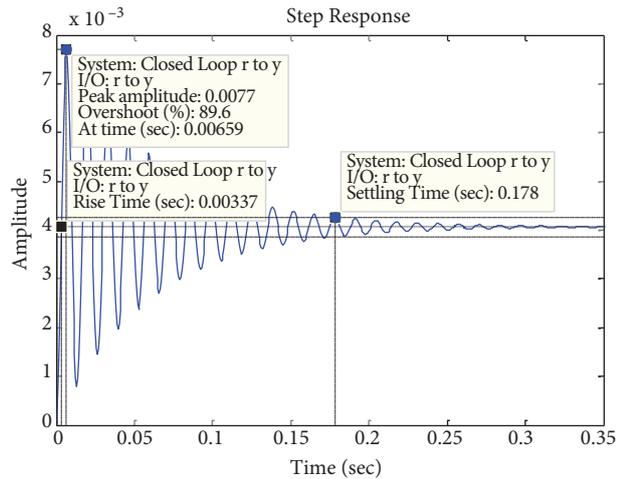
Figure 4. The unit step response of valve using P controller.

The results are shown in Figure 5 for the case with the PI controller. The gain parameters of the controllers were chosen with singular frequency method and again set to $K_P = 20$, $K_I = 8000$. This method implements robust control design techniques to locate stabilizing PID regions in the parameter space [14]. As can be seen from Figure 5, overshoot and settling time are 6.63% and 0.376 s. The rise time is 0.0831 s.

Figure 6 shows the unit step response of the valve changing 10% in valve parameters such as movable mass (M), resistance and friction coefficient (R), and effective area of diaphragm (A). Note that overshoot and settling time are 9.09% and 1.78 s. The rise time is 0.184 s.

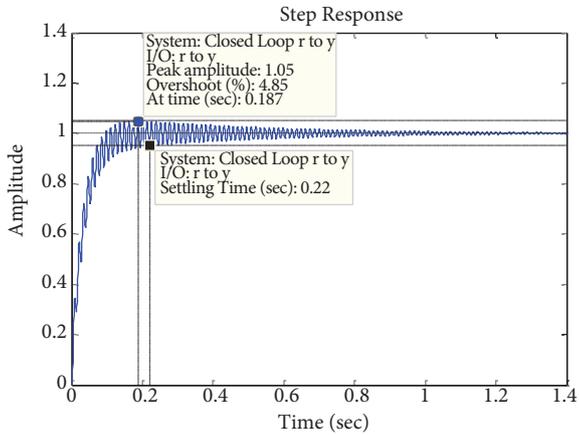**Figure 5.** The unit step response of valve using PI controller.

Figure 6. The unit step response of valve with PI controller in feedforward path (with changes of about 10% in M, R, and A).

Using the PID controller, the results of the overshoot and settling time are shown in Figure 7.

The tuning algorithm is of singular frequency and performance metric was ITAE. The gain parameters K_P, K_I, and K_D of the PID controller were set to $K_P = 7570$, $K_I = 1.514e-4$, and $K_D = 15.21$. From Figure 7, the settling time is 0.22 s and the overshoot is small. The output has an overshoot of less than 5%. The rise

time is 0.0967 s. Figure 8 shows the unit step response of the valve with 10% percent changes in valve parameters such as movable mass (M), resistance and friction coefficient (R), and effective area of diaphragm (A). Note that overshoot and settling time are 5.53% and 0.302 s. The rise time is 0.0968 s. Table 2 shows the results of the three controllers for the valve. All controllers were tuned with MATLAB (http://www.mathworks.com/).

Figure 7. The unit step response of valve with PID controller.
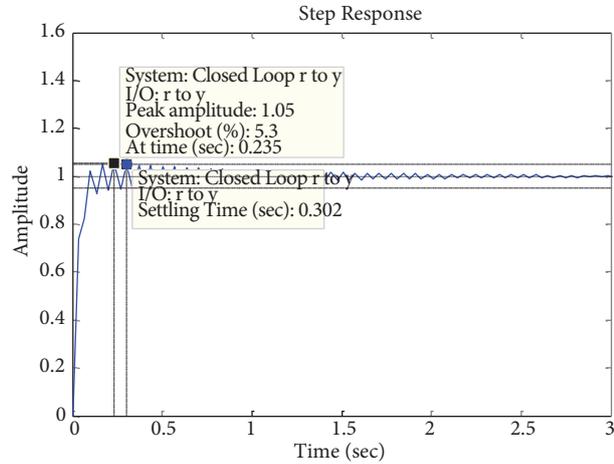
Figure 8. The unit step response of system with PID controller in feedforward path (with changes of about 10% in M, R, and A).

Table 2. Results of controllers for valve.

Controller	Overshoot (%)	Settling time (s)	Rise time (s)
P	89.6	0.178	0.00337
PI	6.63	0.376	0.0831
PID	4.85	0.22	0.0967

5. Conclusion

The aim of this study was the development of the design of some controllers to meet transient response specifications of a sliding-stem pneumatic control valve. For comparison, P, PI, and PD controllers were utilized to control the valve. In this project we proposed a type of pneumatic control valve by bond graph method. Optimal control design, servo control, and full and minimum observer controller were also applied to the control of the stem position in a pneumatic control valve system. The effect of control gain parameters on the step response of the valve motion was studied in detail. From this study, the following conclusions can be drawn:

1. The PID controller has the minimum overshoot of the controllers designed for the valve.

2. Based on the studies conducted, the PID controller is more robust than the P and PI controllers.

3. The PID controller in a feedforward path has better response characteristics than the PID in a feedback path.

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