ELUCIDATION AND HANDLING OF VALVE ACTUATOR NONLINEARITY IN PROCESS CONTROL LOOPS: AN OVERVIEW OF RECENT RESULTS

Helen Durand[1] and Panagiotis D. Christofides *,[1,2]

[1]Department of Chemical and Biomolecular Engineering, University of California, Los Angeles, CA 90095-1592
[2]Department of Electrical Engineering, University of California, Los Angeles, CA 90095-1592

Abstract

Valves are ubiquitous as final control elements in the chemical process industries. Motivated by heavy valve usage by the process industries and many industrial reports of valve nonlinearities undermining effective process control, this article summarizes our recent work that unifies concepts related to valve dynamics with frequently reported industrial concerns, and also investigates methods of compensating for these valve dynamics employing suitable modifications to widely-used control systems such as proportional-integral control and model predictive control.

Keywords

Valve dynamics, stiction, classical control, model predictive control.

Introduction

Because continuous chemical processes involve flows undergoing reaction or separation, adjusting the flow rates of the reacting and separating streams is a central issue for the safe and profitable operation of a chemical process system. Typically, flows are adjusted using valves. Valves themselves have dynamics, often nonlinear dynamics. Undergraduate chemical engineering coursework typically assumes that the valve output is exactly equal to the set-point of the valve output dictated by the control system or addresses valve dynamics only in a limited fashion, such as by discussing actuator saturation or valve dynamics that can be modeled with a first- or second-order transfer function. Sometimes, nonlinear relationships between the flow rate out of the valve and the valve percentage open (e.g., an equal percentage inherent valve characteristic) are discussed, but typically in the context of valve sizing and not in a manner that suggests the impact of such a valve nonlinearity on control system performance.

Despite the lack of focus on valve dynamics in traditional undergraduate chemical engineering studies, valve dynamics have significant implications for industrial chemical process control. For example, stiction, a nonlinear friction effect that plagues many industrial valves worldwide, has been repeatedly cited as responsible for control loop and even plantwide oscillations (Choudhury (2011)). Other nonlinearities, such as deadzone and backlash, also reduce control system effectiveness. The issues due to the stiction nonlinearity have been so pervasive that a large body of literature has been devoted to the study of stiction modeling, quantification, detection, and compensation (Brásio et al. (2014)). Some of the stiction compensation strategies developed include those that modify the control signal sent to the valve (Hägglund (2002); Srinivasan and Rengaswamy (2008)) or those that retune the controller for the loop (Ale Mohammad and Huang (2012)).

Based on the above, it is clear that a wide range of valve dynamics exist that can have impacts with varying degrees of severity on chemical process control. This work summarizes our recent research on this topic. A goal of this work is to highlight the manner in which

*To whom all correspondence should be addressed, Email: *pdc@seas.ucla.edu.*

the type of valve dynamics, control loop architecture, and type of controller interact to cause different negative effects on process control due to valve dynamics. We also highlight three compensation techniques that take advantage of industrially employed control strategies.

Nonlinear Process-Valve Modeling

We consider nonlinear processes modeled with systems of first-order ordinary differential equations with the following state-space form:

$$\dot{x} = f(x, u_a, w) \tag{1}$$

where $x \in R^n$ is the state vector, $u_a \in R^m$ is the vector of process inputs, and $w \in R^l$ is the disturbance vector. The components of u_a are the outputs of valves used to control the process, and they depend on the valve set-points $u_{m,i}$, $i = 1, \ldots, m$, that are calculated by a controller (e.g., a model predictive controller (MPC) or proportional-integral-derivative (PID)-type controller). The dependence of each $u_{a,i}$ on each $u_{m,i}$ may be described either through a static function or through differential equations, depending on the type of dynamics that the valve exhibits and the control loop configuration in which the valve is situated. We define the vector q to be the process-valve state vector, for which the components include all states x_i, $i = 1, \ldots, n$, of the process as well as all states of the valve layer (e.g., the valve position and velocity if modeling of the dynamic evolution of those states is required to adequately characterize those states; this may occur, for example, when the valve is sticky and the position and velocity vary according to a force balance that includes the friction force on the valve). The dynamic equations for the process and the valve layer are combined into one process-valve model defined by the vector function f_q as follows:

$$\dot{q} = f_q(q, u_m, w) \tag{2}$$

It is assumed that the origin is the equilibrium of this process-valve system when $w(t) \equiv 0$.

Types of Valve Dynamic Responses

Valve dynamics that can be described by static equations include nonlinear valve characteristics such as equal percentage or fast opening (square root) characteristics (Coughanowr and Leblanc (2009)). A valve characteristic describes the relationship between the percentage that the valve has traveled between its fully open and fully closed positions (percentage open) and the percentage of the fully open valve output flow rate that is achieved at the given percentage open of the valve. Figure 1 shows several common valve characteristic curves (linear (L), equal percentage (EP), and square root (SR) valve characteristics) for a valve for which the fully closed valve is associated with the minimum stem position and the fully open valve is associated with the maximum stem position.

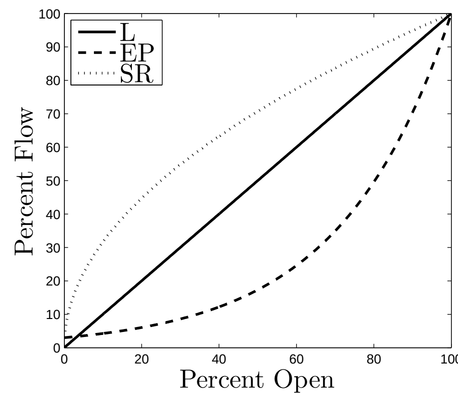

Figure 1. Common valve characteristics: L (linear), EP (equal percentage), and SR (square root).

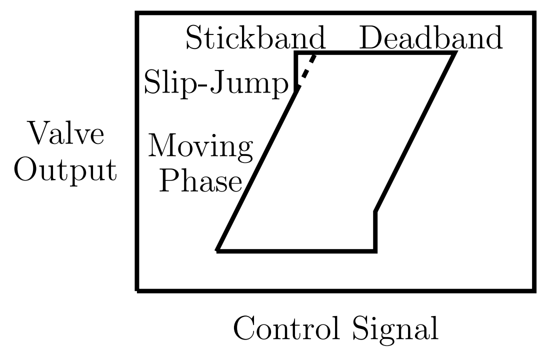

Figure 2. Plot showing major characteristics of the response of a sticky valve to changes in the control signal to the valve.

In addition to valve dynamics described by static equations, some valve dynamic behavior can be described by dynamic systems of equations. One example that is typically incorporated within the undergraduate process control curriculum is valve dynamics that can be modeled using a first- or second-order transfer function. Such dynamics may arise from, for example, the lag as air is transferred into the area containing the valve diaphragm in a pneumatic spring-diaphragm valve (Coughanowr and Leblanc (2009)). A common dynamic valve nonlinearity in the chemical process industries is stiction. Stiction is caused by friction between the moving surfaces of the valve. It prevents the valve output (flow rate) from responding to the control signal to the valve until the force on the valve from the actuation due to the control signal sent to the valve has built up enough to overcome the deadband and stickband of the valve. When the breakaway force is exceeded, the valve stem moves quickly to a new position (slip-jump), after which the valve responds linearly (moving phase) to the control signal to the valve until the changes in the control signal to the valve change direction, as shown in Figure 2. It is notable that the response of a sticky valve to changes in the control signal to the valve has similarities to the response of a valve subject to other valve nonlinearities described by dynamic equations, such as deadband (resulting from, for example, backlash) and hysteresis (Choudhury et al. (2005)). Thus, analysis and compensation of valve stiction can provide some insight for other valve nonlinearities.

As the severity of stiction worsens, the nonlinear effects of stiction become more pronounced. For example, Figure 3 shows the output from a pressure-to-close pneumatic spring-diaphragm sliding-stem globe valve with low stiction (Vendor valve) and from a valve with more significant stiction (Nominal valve) as the pressure applied to the valve is ramped up and down (Vendor and Nominal parameters taken from Garcia (2008)). As shown in the figure, after the initial transient, the valve output enters a loop that takes a different shape for the Vendor valve than for the Nominal valve. The overall effect is that the same range of pressures cannot reach the same range of valve outputs for the Nominal valve as it can for the Vendor valve. In effect, the worsening of stiction has constrained the range of valve outputs that can be achieved with a given available actuation energy.

Types of Control Loops

The prior section shows that different types of valve dynamics affect the valve output differently, and thus they negatively impact effective process control in different ways. The manner in which they affect process control is dependent on the type of controller used (e.g., MPC or PID) as well as the control loop architecture (i.e., the valve may be operated in open-loop such that it receives a valve output flow rate set-point from the controller of the process variable but the flow rate out of the valve is not itself controlled to bring it to the flow rate set-point more quickly, or it may be operated in closed-loop such that a flow controller is used to attempt to drive the valve output to its set-point more quickly). Figure 4 demonstrates the control loop design considered, allowing for different controller types and architectures.

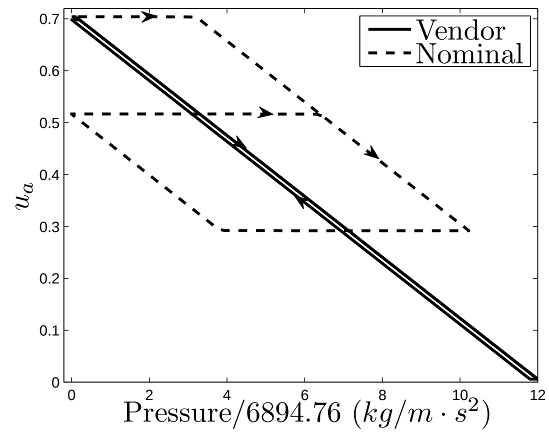

Figure 3. Plot showing response of valve output to ramping of pressure applied by pneumatic actuation to valve stem for different levels of valve stiction.

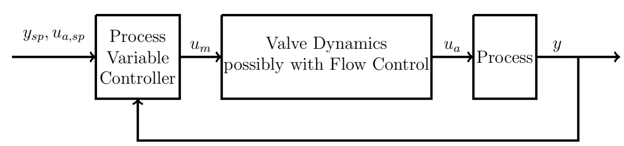

Figure 4. Figure demonstrating various control architectures that may be used in a loop with a valve as the final control element. In this figure, y represents the controlled variable and y_{sp} and $u_{a,sp}$ represent the controlled variable set-point and valve output set-point respectively.

The relationship between the type of valve dynamics, the type of controller, and the control loop architecture can be better understood through several examples that highlight different issues observed due to actuator dynamics, as follows:

- Consider a level control loop operated under PI control, with a sticky valve operated in open-loop (where the open-loop control law determines the pressure that the pneumatic actuation should apply to the valve stem for a given valve output set-point determined by the PI controller based on a linear relationship developed for the valve when stiction was low). An issue that may be observed in this case is stiction-induced oscillations in the controlled variable. These occur due to the balance of forces on the valve: the dynamics of the friction force allow it to change more rapidly than the force from the actuation that is linearly related to the output of the PI controller. The result is that the friction dynamics work together with the dynamics of the PI controller (e.g., its integral action) and the fact that the valve is operated in open-loop which restricts the force from the actuation to change at a rate dictated by the PI controller dynamics to produce the control loop oscillations (Durand and Christofides (2017)).

- Consider a level control loop operated under a tracking MPC (MPC with a quadratic stage cost) with a valve with an equal percentage characteristic. This valve is operated in open-loop such that each valve output flow rate set-point calculated by the MPC is translated directly to a stem position that the valve takes, but the relationship used between the set-point and the stem position assumes that the valve has a linear characteristic. Due to the significant plant-model mismatch because of this, coupled with the open-loop valve operation that does not allow for the error to be eliminated unless the flow rate set-point corresponding to the correct valve position is output by the MPC, the level under the MPC may show permanent offset if the model used to predict the closed-loop state does not include the valve dynamics.

- Consider a sticky valve operated in closed-loop and controlling the feed flow rate for an ethylene oxide process under economic model predictive control (EMPC; this is a model predictive control strategy formulated with an economics-based objective function that may not operate a process at steady-state (Durand and Christofides (2016a))). If the EMPC is unaware of the valve dynamics, it may calculate set-points higher than the sticky valve can achieve without the pressure becoming negative (see Figure 3), with the result that the pressure will saturate at its minimum value and the valve output set-points will not be reached, which can lead to violation of process constraints.

- Consider a valve operated in closed-loop for which the dynamics can be modeled as a first-order transfer function, which is adjusting the feed flow rate to an ethylene oxide process under economic model predictive control. Concurrently, another actuator operated in closed-loop with first-order dynamics is adjusting the ethylene feed concentration (Durand et al. (2014)). When the dynamics of these actuators are not included within the model used by the EMPC to predict optimal control actions, a constraint on the amount of reactant available in a given operating period can be exceeded due to the plant-model mismatch.

From the above examples, it follows that one major cause of poor performance in control loops containing valves with significant dynamics is that the valve dynamics are not accounted for correctly in the control loop design. In the following section, we will discuss three methods for dealing with valve dynamics that seek to either modify the valve dynamics or to include the valve dynamics within the control loop design to avoid the negative issues noted above.

Valve Dynamics Compensation Methods

In our recent work (Durand and Christofides (2017)), we have analyzed three methods for compensating for valve dynamics as follows:

- If the valve is operated in open-loop, add flow control to operate it in closed-loop.
- If the controller for the process variable contains an integral term, adjust the integral term with an anti-windup-inspired term.
- If the controller for the process variable is an MPC, incorporate the valve dynamics (in addition to the process dynamics) in the model used by the MPC.

These compensation techniques will now be discussed.

Closed-Loop Valve Operation

In this method, the value of u_a is regulated to u_m using flow control. This methodology has similarities to the concept of applying a positioner to a valve, which has also been cited as a method for stiction compensation (Ivan and Lakshminarayanan (2009)). An advantage of this method is that it utilizes a standard control design (e.g., a standard PI controller can be used to control the flow rate out of the valve). A disadvantage is that it requires additional instrumentation and control to be added to an open-loop valve.

Modification of Integral Error

For process variables controlled by a controller with an integral error (e.g., PI control), but with a sticky valve in the control loop, the dynamics of the integral term may affect the rate at which the force on the valve due to the actuation can change (i.e., when the valve is operated in open-loop), which can result in oscillations in the value of u_a and in the process variable. To prevent these oscillations, the integral error can be modified with a term containing a tuning parameter L. To illustrate, the formulation of a PI control law with a modified integral error to prevent stiction-induced oscillations is as follows:

$$u_m = u_{as} + K_c(y_{sp} - y) + K_c\zeta/\tau_I \tag{3}$$

$$\dot{\zeta} = (y_{sp} - y) + L(u_a - u_m),\ \zeta(0) = 0 \tag{4}$$

where u_{as} is the flow rate out of the valve immediately before the process variable set-point is changed to y_{sp}, and K_c and τ_I are the proportional gain and integral time of the PI controller. The value of L can be tuned for the desired set-point so that it is not so small that there is hardly any modification of the integral term, but also not too large such that $\dot{\zeta}$ becomes zero before y_{sp} is reached. An additional modification that can be used in Eq. (4) to reduce offset due to this latter effect ($\dot{\zeta} = 0$) is to determine a rate at which L can exponentially decay such that it remains large for a sufficient period of time to ameliorate the oscillations but then decays to prevent ζ from remaining stagnant when y_{sp} has not been reached.

An advantage of the above approach is that it uses a technique inspired by anti-windup methods that are already employed in industrial PID-type controllers and requires only an adjustment to the integral term. A disadvantage is the need to tune L and also to determine a suitable decay rate for L when offset from the set-point is observed due to the inclusion of the anti-windup-inspired term in Eq. (4).

Incorporation of Valve Dynamics in MPC

A final method for utilizing a standard control formulation to compensate for valve dynamics is to include the valve dynamics in the model used to determine the optimal control actions applied in MPC. Thus, the proposed valve dynamics compensation strategy is the use of a process-valve model, with additional constraints based on the effects of valve nonlinearities on process control, within MPC. Some examples of constraints that may be considered for stiction compensation are actuation magnitude constraints (e.g., constraints that require the pressure applied by the pneumatic valve actuation to never reach non-physical values such as negative pressures), input rate of change constraints (i.e., constraints that require that the control actions calculated by the MPC between two sampling periods differ from each other by no more than a pre-specified value), or adjusting the bounds on the allowable control actions. These three constraints are different methods for addressing the issue observed in Figure 3, which shows that as stiction develops, negative pressures would be required to reach certain valve output flow rates that were achievable with a positive pressure when the valve exhibited minor stiction. The actuation magnitude constraints are a robust means of handling the issue, but they may be computationally expensive to implement and also require that the details of the pressure signal from the pneumatic actuation be modeled. Input rate of change constraints are a more ad hoc method for dealing with the issue, but they may help in situations where the MPC calculates that drastic changes in the control action between two sampling periods are required to optimize the objective function, which may cause the value of u_m to reach values that the pressure available from the pneumatic actuation cannot drive the valve output to meet. From Figure 3, it is also seen that adjusting the bounds on u_m as stiction develops is another method for preventing the pressure from becoming negative (see also, for example, (del Carmen Rodríguez Liñán and Heath (2012)) for other examples of MPC's employed in stiction compensation where bounds on the control actions are adjusted), but this requires that the bounds be re-estimated throughout time as stiction worsens and the bounds change. The formulation of MPC-based valve dynamics compensation is as follows:

$$\min_{u_m(t)\in S(\Delta)} \int_{t_k}^{t_{k+N}} L_{MPC}(\tilde{q}(\tau), u_m(\tau))\, d\tau \tag{5a}$$

$$\text{s.t.} \quad \dot{\tilde{q}}(t) = f_q(\tilde{q}(t), u_m(t), 0) \tag{5b}$$

$$\tilde{q}(t_k) = q(t_k) \tag{5c}$$

$$\tilde{q}(t) \in Q,\ \forall\, t \in [t_k, t_{k+N}) \tag{5d}$$

$$u_m(t) \in U,\ \forall\, t \in [t_k, t_{k+N}) \tag{5e}$$

$$g_1(\tilde{q}(t), u_m(t)) = 0,\ \forall\, t \in [t_k, t_{k+N}) \tag{5f}$$

$$|u_{m,i}(t_k) - u^*_{m,i}(t_{k-1})| \le \epsilon,\ i = 1, ..., m \tag{5g}$$

$$|u_{m,i}(t_j) - u_{m,i}(t_{j-1})| \le \epsilon,\ i = 1, ..., m,\quad j = k+1, ..., k+N-1 \tag{5h}$$

where $S(\Delta)$ signifies the set of vector-valued piecewise-constant functions with period Δ, L_{MPC} is the objective function, $\tilde{q}$ represents the predicted process-valve state from the nominal process-valve model of Eq. (5b) with the initial condition derived from the state measurement at time t_k in Eq. (5c), Eqs. (5d) and (5e) are state and input constraints, Eq. (5f) represents equality and inequality constraints that may include actuation magnitude constraints, and Eqs. (5g)-(5h) are input rate of change constraints ($u^*_{m,i}(t_{k-1})$ represents the implemented value of $u_{m,i}$ at the previous sampling time). In addition to the flexibility that the MPC-based valve dynamics compensation method offers for modifying the constraints depending on the characteristics of the valve dynamics being handled, it also provides a number of other advantages. Most notably, the MPC can anticipate the full response of both the process and also of the valves by incorporating models of each, with the result that it is well-suited to handle the negative effects of a variety of types of valve dynamics for both open and closed-loop valves. It can also account for multivariable interactions when there are multiple loops impacted by valve dynamics. In addition, the method is applicable for a wide variety of objective functions (e.g., tracking or economics-based), and stability constraints based on Lyapunov functions have been developed for the method that provide provable feasibility and closed-loop stability properties to the controller even with input rate of change constraints (Durand and Christofides (2016a)).

A disadvantage of the method is that a model of the valve dynamics is required, which may be difficult to obtain and may result in a stiff process-valve model if the valve dynamics are significantly faster than the process dynamics. A method for improving the computation time under such circumstances, while reducing the need for modeling the details of the valve layer, is to determine an empirical model of the valve dynamics and to use this within the MPC for valve nonlinearity compensation. This was demonstrated in Durand and Christofides (2016b) in the context of EMPC by developing a model for the valve layer dynamics using only data on u_m and u_a for a valve operated in closed-loop.

Acknowledgments

Financial support from the National Science Foundation and the Department of Energy is gratefully acknowledged.

References

Ale Mohammad, M. and Huang, B. (2012). Compensation of control valve stiction through controller tuning. *Journal of Process Control*, 22:1800–1819.

Brásio, A. S. R., Romanenko, A., and Fernandes, N. C. P. (2014). Modeling, detection and quantification, and compensation of stiction in control loops: The state of the art. *Industrial & Engineering Chemistry Research*, 53:15020–15040.

Choudhury, M. A. A. S. (2011). Plantwide oscillations diagnosis-current state and future directions. *Asia-Pacific Journal of Chemical Engineering*, 6:484–496.

Choudhury, M. A. A. S., Thornhill, N. F., and Shah, S. L. (2005). Modelling valve stiction. *Control Engineering Practice*, 13:641–658.

Coughanowr, D. R. and Leblanc, S. E. (2009). *Process Systems Analysis and Control*. McGraw-Hill, Boston, MA, 3rd edition.

del Carmen Rodríguez Liñán, M. and Heath, W. P. (2012). MPC for plants subject to saturation and deadzone, backlash or stiction. In *Proceedings of the 4th IFAC Nonlinear Model Predictive Control Conference*, pages 418–423, Noordwijkerhout, Netherlands.

Durand, H. and Christofides, P. D. (2016a). Actuator stiction compensation via model predictive control for nonlinear processes. *AIChE Journal*, 62:2004–2023.

Durand, H. and Christofides, P. D. (2016b). Empirical modeling of control valve layer with application to model predictive control-based stiction compensation. In *Proceedings of the 10th IFAC Symposium on Nonlinear Control Systems*, pages 41–46, Monterey, CA.

Durand, H. and Christofides, P. D. (2017). Elucidation and compensation of valve stiction-induced oscillations in closed-loop systems. In *Proceedings of the American Control Conference*, submitted, Seattle, Washington.

Durand, H., Ellis, M., and Christofides, P. D. (2014). Integrated design of control actuator layer and economic model predictive control for nonlinear processes. *Industrial & Engineering Chemistry Research*, 53:20000–20012.

Garcia, C. (2008). Comparison of friction models applied to a control valve. *Control Engineering Practice*, 16:1231–1243.

Hägglund, T. (2002). A friction compensator for pneumatic control valves. *Journal of Process Control*, 12:897–904.

Ivan, L. Z. X. and Lakshminarayanan, S. (2009). A new unified approach to valve stiction quantification and compensation. *Industrial & Engineering Chemistry Research*, 48:3474–3483.

Srinivasan, R. and Rengaswamy, R. (2008). Approaches for efficient stiction compensation in process control valves. *Computers & Chemical Engineering*, 32:218–229.