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Friction compensation in control valves: Nonlinear control and usual approaches

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ABSTRACT

This work presents different approaches to reduce the control valve friction effect on a process. One is to use the sliding mode control in different conditions and then to compare this controller to widely used algorithms and devices that reduce the control loop variability. The experiments were performed in the Flow Pilot Plant of Polytechnic School of the University of São Paulo with a pneumatic control valve with high friction in a flow control loop. The sliding mode controller yielded promising results, which can represent new horizons with respect to friction compensation in control valves.

1. Introduction

According to Srinivasan and Rengaswamy (2005), about 20–30% of the oscillations in control loops are caused by friction and hysteresis in control valves. Plant maintenance normally occurs at intervals between 6 months and 3 years (Srinivasan & Rengaswamy, 2005). Thereby, if a control loop is oscillating due to friction in the control valve, it is not desirable to stop the process in order to perform valve maintenance. In this situation, the use of a friction compensator or another kind of control that improves the control loop performance is desired.

In previous works about this topic, Kayihan and Doyle (2000) presented the use of nonlinear control techniques to reduce the friction effect in a simulated valve and the simulations were performed in a valve stem position control loop. After that, in the work of Baeza (2013), the same techniques were used to control the position of a real valve with high friction indexes. In this work, the integrative sliding mode control presented the best results and some properties of its control law proved to be very interesting to compensate friction.

Thus, in this work, the integrative sliding mode controller (SMC) is used in two different approaches. The first is controlling the flow of the plant using the valve as a control element. This implementation can be named integrative SMC under external topology, because it is supposed to be implemented in a Digital Control System (DCS) level, i.e., externally to the control valve. Thereby, considering that the SMC has a switching component in its structure, the algorithm is tested in different sampling times (1 ms, 10 ms and 100 ms), in order to check the influence of this parameter in the control loop performance and to evaluate the possibility of implementing it in an industrial DCS (where the fastest sampling time is usually 100 ms). Besides, the integrative SMC under external topology needs the information of some states of the plant, i.e., the position and velocity of the valve stem and the flow; thus, if the valve stem position and velocity are not measured, a state observer is implemented to estimate them. Therefore, the integrative SMC under external topology is implemented according to the following conditions:

- with measured states with sampling times of 1 ms, 10 ms and 100 ms; and
- with estimated states with sampling times of 1 ms, 10 ms and 100 ms.

The second approach to the integrative SMC is under the internal topology. In this case, the SMC is used as a slave control loop for valve stem position. The master loop is a Proportional-Integrative (PI) controller regulating the flow. It is called internal topology because it can be implemented in an embedded system as a digital positioner installed with the control valve.

In turn, the usual approach to reduce the control loop variability due to friction is to apply a compensation method or to use a digital positioner. In this work, the Constant Reinforcement 2 (CR2) compensation algorithm (Silva & Garcia, 2014) is used with a PI controller, in which the friction compensation is implemented at the same level as the PI, characterizing an external topology as defined previously. Furthermore, the performance of a digital positioner is tested in cascade with a PI controller (characterizing an internal topology), which is a widely used configuration in industry to reduce control loop variability. The results of all these methods are also compared with a PI controlling the flow without any other friction

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compensator or controller.

All the experiments were performed in the Flow Pilot Plant of Polytechnic School of the University of São Paulo, with a control valve with high friction. Another control valve was used to insert disturbances in the flow loop in one of the experiments. The controllers were implemented in a Matlab/Simulink platform, except for the digital positioner algorithm embedded in the referred device. Therefore, this work consists in testing new and common strategies to reduce the friction effect in control valves in a real plant.

In Section 2, the friction models used in this work are presented together with other parameters related to the valve stem model dynamics. Section 3 presents the project of the controllers and friction compensators. Section 4 describes the experiments performed and their results; finally, Section 5 exposes the conclusions.

2. Friction models

This section is a brief introduction to friction models applied to control valves, which are more detailed in the work of Garcia (2008). The first part regards the modified classic friction model and the second, the Kano friction model.

2.1. Modified classic friction model

The original classic friction model is given by Eq. (2.1) (Garcia, 2008):

$$F_{at}(\dot{x}) = \left[F_c + (F_s - F_c)e^{-\left(\frac{\dot{x}}{v_S}\right)^2}\right]\mathrm{sgn}(\dot{x}) + F_v\dot{x} \tag{2.1}$$

where F_{at} is the friction force, F_c is the Coulomb friction, F_s is the static friction, F_v is the viscous friction, v_s is the Stribeck velocity and $\dot{x}$ is the valve stem velocity.

This model is used in the SMC design for the feedback linearization of the plant. However, for this application, the model and its derivatives must be continuous. Therefore, the modified classic friction model is given by Eq. (2.2):

$$F_{at}(\dot{x}) = \left[F_c + (F_s - F_c)e^{-\left(\frac{\dot{x}}{v_S}\right)^2}\right]\tanh(\dot{x}) \tag{2.2}$$

The hyperbolic tangent approximation for the signal function was proposed by Kayihan and Doyle (2000). Furthermore, the elimination of the viscous friction portion is based on (Romano & Garcia, 2008), in which they state that, considering the velocities in which an industrial valve stem normally operates, the viscous friction component is very small as compared to the other components. Consequently, this term can be discarded in the model.

2.2. Kano model

The Kano model (Kano, Maruta, Kugemoto, & Shimizu, 2004) describes friction using two parameters:

- S: variation of the control signal necessary to move the valve stem when there is a change in the movement direction; and
- J: known as stick band (or slip jump, if considering the initial movement of the valve stem), is the variation of the control signal necessary to move the valve stem when it stops and it is required to start moving in the same direction of the movement.

These parameters can be observed in the signature curve of a valve, as in Fig. 1. This model is used in the state observer project and also in the CR2 compensator because it is more precise than the classic model, since it describes the discontinuity of the friction force in the imminence of movement..

2.3. Friction parameters quantification

In industry, it is not usually possible to stop the process to perform open loop tests to quantify the parameters of a friction model to be used in a compensation technique. Therefore, the estimation of the friction parameters for both models is made by non-intrusive methods.

Firstly, the method of interconnected blocks proposed by Romano (2010) is used to estimate parameters S and J from Kano's model. This algorithm quantifies the friction in the valve using the data collected in a normal operation of the plant (in closed loop) and it is based on a nonlinear system identification for systems with considerable friction in the control valve. This method estimates the friction parameters (using Kano's model) and provides a nonlinear model for the plant. Further details are presented in (Romano, 2010) or in (Hidalgo & Garcia, 2012).

The measured friction parameters for the high friction control valve used herein are given in Table 1.

Secondly, the quantification of the classic model parameters is made by the equivalence between the friction models. The work of Uehara (2009) presents a relation between some parameters of the classic model and S and J from Kano's model, given by Eqs. (2.3) and (2.4):

$$F_s = \frac{(S + J)S_a\Delta p_{\max}}{2} \tag{2.3}$$

$$F_c = \frac{(S - J)S_a\Delta p_{\max}}{2} \tag{2.4}$$

where S_a is the section area of the actuator (445 mm^2, information provided by the manufacturer of the valve – Fisher Controls International (2000)) and Δp_{max} is the maximum variation of the pressure that can be applied by the actuator, which is 24 psi (information available on the actuator label). Therefore, the classic model friction parameters are given in Table 2.

Furthermore, in Kayihan and Doyle (2000) and Baeza (2013) the Stribeck velocity and σ had the same values, even though the control valves were completely different. The same approach is used here for these parameters, which are shown in Table 3.

3. Design of the controllers

In Subsection 3.1, the design of the integrative SMC controller under external topology is presented. In the same subsection, the design of the state observer is also shown. In Subsection 3.2, the project of the integrative SMC under internal topology is exposed. In

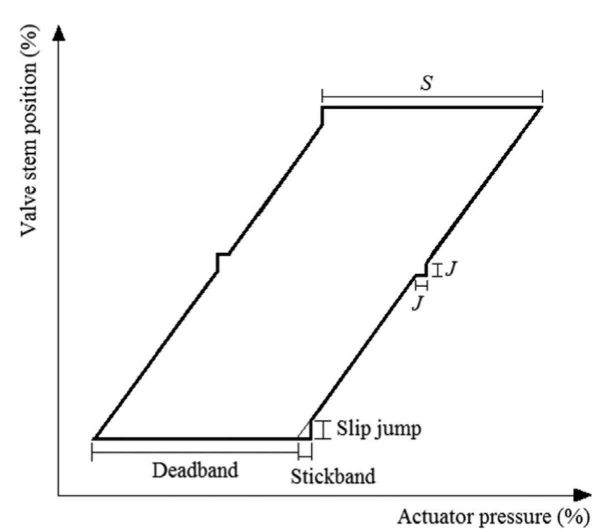

Fig. 1. Valve signature curve with Kano's model parameters (Choudhury, Thornhill, & Shah, 2003).

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