An augmented PID control structure to compensate for valve stiction

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Abstract: It is well-known that valve stiction causes sustained oscillations on process variables when a traditional PID controller is implemented in the feed-back loop. In the literature, there is a vast collection of solid techniques to compensate for valve stiction which employ different approaches and require various prior knowledges on process and stiction dynamics. Among others methods, PID retuning or changes to the traditional algorithm and structure of PID can be useful solutions to mitigate or remove negative effects of valve stiction. Appropriate controller retuning can reduce significantly amplitude and frequency of oscillation, but it cannot remove the problem permanently. Modifying traditional PID algorithm or augmenting standard structure of the controller are also robust approaches for the scope. This paper briefly revises some PID-based stiction compensation techniques and illustrates a new version of stiction-aware PID. A standard PI(D) controller is augmented with a two-move compensator and, by monitoring the control error, it is able to remove effect of valve stiction and to guarantee set-point tracking and disturbance rejection. This PID-based structure requires the estimation of controller output associated with the desired valve position at steady-state and the estimate of valve stiction parameters.

Keywords: PID-based controller; alternatives to PID controllers; control valve; stiction compensation

1. INTRODUCTION

Static friction (stiction) in control valve is considered one of the most common sources of sustained oscillations in industrial control loops. This severe malfunction has been extensively studied in last three decades, but maintenance and repair still remain as definitive solutions to fix a sticky valve. Nevertheless, these actions may be feasible only during scheduled plant shutdowns, which typically happen from every six months to even some years. Therefore, compensation methods become good solutions to combat routinely and with significant economic savings the negative effects of valve stiction on control loop performance.

A comprehensive and updated review of stiction compensation methods has been recently presented by Bacci di Capaci and Scali (2018). This survey work summarizes the features of almost 40 different techniques, in terms of mitigation or removal of oscillations on process variable, reduction of valve movements, requirement of a-priori process knowledge, and closed-loop performance in terms of set-point tracking and disturbance rejection. Eight different categories of compensation methods for valve stiction can be mentioned:

- compensation through retuning of PID controller, e.g., Gerry and Ruel (2001);
- knocker methods, e.g., Hägglund (2002);
- constant reinforcement (Ivan and Lakshminarayanan, 2009);
- alternate knocker (Srinivasan and Rengaswamy, 2007);
- two- or N-move compensators, e.g., Srinivasan and Rengaswamy (2008); Cuadros et al. (2012a);
- optimization approaches, as Srinivasan and Rengaswamy (2008);
- MPC-based methods, e.g., Rodríguez and Heath (2012);
- miscellaneous techniques.

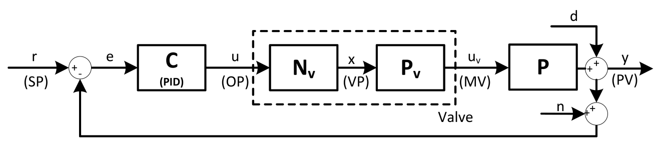

Fig. 1. The closed-loop system with the (sticky) control valve followed by the process.

All these approaches require to some extent modifications to standard PID controller, simply in terms of tuning and algorithm or even in terms of whole control structure. It is indeed well-known that a traditional PID controller, tuned only on process dynamics, generates sustained oscillation due to its integral component which causes excessive variations of control action in order to force pneumatic valve actuator overcome static friction. Smart valve positioners can be a good alternative to mitigate negative effects of stiction. Fast responses can be obtained and amplitudes and frequencies of oscillation can be reduced by the use of the positioner-embedded controller, usually with a P(D) algorithm based on valve position error. Nevertheless, these devices are not the ultimate solution as proved by Bacci di Capaci et al. (2013) and Hidalgo and Garcia (2017).

Figure 1 shows the variables of a standard feedback control loop. Set-Point (SP, r), Process Variable (PV, y), and Controller Output (OP, u) are usually recorded, while the actual valve position (VP, x) can be measured only by smart sensors as encoders. The manipulated variable (MV, u_v), i.e. the flow rate exiting the valve, that is, the actual process input, is often measured, and in a flow rate control loop the process dynamics P coincides basically with the valve dynamics itself P_v. PID controller and stiction nonlinearity are represented by block C and N_v, respectively.

This paper briefly discusses the main PID-based methods for stiction compensation and illustrates a new version of stiction-aware PID. A standard PI(D) controller is augmented with a two-moves compensator in order to remove sustained oscillations and to still guarantee set-point tracking and disturbance rejection. Such controller is stiction model-based and requires some prior-knowledge, as the stiction amount, and some on-line variables, as the desired valve position at the steady-state. An estimate of valve stiction parameters is obtainable with specific identification and quantification techniques, while the steady-state valve position can be evaluated by using the controller output oscillating before compensation.

The remainder of the paper is as follows. Existing PID-based compensation methods are briefly discussed in Section 2. The proposed method is introduced in Section 3. Several simulation examples are provided in Section 4. Finally, conclusions are drawn in Section 5.

2. PID-BASED COMPENSATION METHODS

The existing PID-based compensation methods are briefly revised below, in order to highlight their positive features and possible drawbacks. Three main categories can be identified: that is, retuning, modifying, and augmenting PID controller.

2.1 Retuning PID

Gerry and Ruel (2001) firstly suggested to address valve stiction on-line by simply acting on the tuning of traditional PID controller. A set of qualitative retuning rules were introduced to decrease the impact of stiction-induced oscillations. Detuning the controller, that is, decreasing proportional gain and/or increasing integral time constant, are two practical suggestions for control operators. Alternatively, for the hardest cases of valve stiction, switching from PI to P algorithm can be a solution. Obviously, the price to pay are slower responses or even large steady-state control errors.

Ale Mohammad and Huang (2012) have suggested a detailed scheme of stiction compensation actions. By using frequency analysis and harmonic balance, and by following some guidelines for controllers retuning, the occurrence and the amplitude of oscillations can be predicted, and then stiction can be mitigate or removed for different process and controller dynamics. For example, for a PI algorithm and a first-order process with time-delay, controller integral time τ_I must be greater than the sum of the process time constant τ and its time-delay θ to avoid oscillations.

Li et al. (2014) specifically analyzed stiction-induced oscillations in cascade control loops by using frequency analysis. A set of practical techniques of oscillation compensation through outer and inner controller tuning, and through changes of control strategies were proposed. Recently, another compensation method by means of controller retuning has been proposed by Fang et al. (2016). By using Newton-Raphson method, this time-domain approach improves the accuracy in calculating the amplitude and period of oscillation, with respect to the describing function technique, thus augmenting performance of stiction compensation. In addition, as shown in some experimental examples, the method is quantitative and avoids tuning the controller parameters in a trial-and-error manner, even though it requires the knowledge of the actual valve output.

2.2 Modified PID

A second approach is to modify traditional PID algorithm, but without the use of an additional compensator.

Mishra et al. (2014) introduced a stiction combating intelligent controller (SCIC) based on fuzzy logic. The SCIC is a fuzzy PI controller, making use of Takagi-Sugeno scheme, with variable integral gain which depends upon the value and the rate of change of the control error. This novel adaptive approach, being also independent of the stiction band value, seems to outperform a traditional PI controller, by yielding less variability in PV and less aggressiveness in valve input, both in the case of set-point tracking and disturbance rejection.

Mishra et al. (2015) have proposed another compensation solution, by using a nonlinear PI controller (NPIC), which is tuned on-line through a Differential Evolution algorithm for ITAE as cost function to be minimized. Again, a nonlinear control law is employed to vary the integral gain, based on the error and its rate of change. Similarly to previous work (Mishra et al., 2014), pilot plant experiments reveal good performance for the modified PID.

Recently, Rohilla et al. (2017) have employed another intelligent controller by cascading a fuzzy integral action with a conventional PD element. The proposed fuzzy I + PD controller outperformed traditional PID controllers by testing set-point tracking and disturbance rejection abilities at different operating points on a laboratory scale pressure control loop with a pneumatic air-to-close valve.

2.3 Augmented PID

The third group of solutions is to add a compensator block to PID controller. Depending on the nature of this supplementary block, various methods can be distinguished.

Knocker. The knocker approaches consist of adding a predefined signal to controller output before entering the valve. This auxiliary block generates short pulses with constant amplitude, width, and duration, in the direction of the rate of change of controller output (Hägglund, 2002). This first method, by producing a faster motion of the valve, reduces oscillations in process variable, but it can cause mechanical problems even worse with respect to normal operating.

Srinivasan and Rengaswamy (2005) suggested some guidelines for the automated choice of compensation parameters of the knocker. This revised approach, by integrating two stiction detection techniques, guarantees reduction of PV variability ensuring less aggressive valve movements.

Later, Cuadros et al. (2012b) presented a revised knocker compensator based on a supervision layer which analyzes the control error and interacts with the standard PI(D) controller. This integrated strategy shows a lower integral absolute error and even a reduced number of valve movements.

Recently, Munaro et al. (2016) developed a compensation method by using pulses with variable amplitude in order to limit the increase of valve movements. The amplitude of pulses becomes zero when a specified limit for the error on process variable is achieved. Another advantage is the ability to cope with variability and uncertainty on friction. The method has been then implemented in an industrial DCS system interfaced to a pilot scale process with features identical to those found in industry including a valve positioner (Arifin et al., 2018).

Two-move compensator. The so-called two-move methods ought to remove oscillations and keep the valve position at its steady-state value, by performing at least two moves in opposite directions. The magnitude of the compensating signal should be large enough to exceed stiction and move the valve, but not too large to saturate it.

In the first implementation of Srinivasan and Rengaswamy (2008), this compensator does not need the controller parameters and, most importantly, does not increase valve wear rate, as the knocker does, since the valve is not constantly forced to move. The compensating signal (f_k) is added to the controller output (u_c), to obtain the valve input ($u = \mathrm{OP}$).

The additional signal can assume only two values by imposing two consecutive movements to the valve. The first signal moves the valve from its stuck position, according to:

$$f_k(t) = |u_c(t)| + \alpha \cdot d \tag{1}$$

and setting:

$$u(t) = u_c(t) + sign\left(\frac{du_c(t)}{dt}\right) \cdot f_k(t) \tag{2}$$

where d is the stick band of the valve, and α is a real number greater than 1. Then, the second signal ought to bring the valve to its steady-state position in order to eliminate the error on PV by:

$$f_k(t+1) = -u_c(t+1) \tag{3}$$

Note that after this second movement, the valve cannot move from the steady-state position since the controller output is canceled by (3). The input signal to the valve (u) is thus constant (zero), that is, the control loop operates as in manual mode.

Nevertheless, this first version of two-move method presents several drawbacks, which heavily hinder its on-line implementation. Firstly, accuracy is reduced by assuming the one-parameter (d) model of Stenman et al. (2003) to predict the valve behavior. Moreover, the steady-state value of valve position (VP_{ss}) is assumed to be known, while VP is not usually measurable in traditional process plants. In particular, the method relies on the strong assumption that all measurements are represented by deviation variables and their respective steady-state values are zero. Thus, for the second movement (3), it is assumed that $\mathrm{VP} = 0$ causes $\mathrm{PV} = \mathrm{SP}$.

Another two-moves method was introduced by Farenzena and Trierweiler (2010). Instead of using an additional compensator block, the traditional PI controller is modified. This technique seems to achieve faster closed-loop performance and efficient rejection of load disturbances. A fair set-point tracking, with a small offset, would be also possible, and a reduction of valve travel is shown. Nevertheless, this method is based on one-parameter stiction model of Stenman et al. (2003), which was proved to be partially inaccurate.

To overcome previous limitations, Cuadros et al. (2012a) revisited the standard two-move approach. Authors showed that assumptions on the knowledge of VP_{ss} that assures $\mathrm{PV} = \mathrm{SP}$ could be not easily achievable in practice. Significant experimental results on a flow rate control loop of a pilot plant are thus provided. Two improved compensation methods are then proposed: the first, consisting of four movements, is sensitive to load disturbances. The second, based on two movements and four states, and especially suited to tackle disturbances, proves more robust. Exact knowledge of the plant model is not required, and loop perturbations (SP changes and disturbances) are handled by monitoring the increase of control error and by switching back and forth to a standard PI(D) controller. However, both methods can be applied only to self-regulating processes, and the second approach requires similar dynamics between valve and process.

Recently, Wang et al. (2015) have presented other two compensation solutions. Firstly, three consecutive implementations of the standard two-move are used. This technique allows on-line estimation of the steady-state value of valve input, thus, no a priori assumption on VP is required. However, this approach could take very long times in real applications, since two extra open-loop step responses must be awaited to compute the final input to valve OP_{ss}.

Simultaneously, the same authors have proposed another implementation which outperforms the three-times two-move method in terms of velocity and lower amplitude of the response. A practical estimate of the desired valve position VP_{ss} is introduced, thus the value of OP_{ss} to impose to get the steady-state can be computed faster. The amplitude of oscillation of controller output before compensation is measured ($\Delta\mathrm{OP} = \mathrm{OP}_{max} - \mathrm{OP}_{min}$), then the amount of valve stiction is estimated in advance solving a set of equations. The objective is to ensure a case that the valve is bound to stick only at two positions. In total, this method imposes six open-loop movements to the valve.

In our previous work (Bacci di Capaci et al., 2016), a further improvement of two-move compensation has been proposed, by employing some practical simplifications. Only four open-loop movements are now required. The first two moves are as in the approach of Wang et al. (2015). Whether OP oscillates, when is increasing, close to its peak, the controller is firstly switched into open-loop mode and OP is set to its maximum OP_{max}. Then, after a tuned time interval, OP is imposed to its minimum OP_{min}. Afterwards, other two suitable moves are forced: the third is needed to unblock the valve, the last aims to place the valve to its steady-state position VP_{ss}. Note that VP_{ss} can be estimated simply as:

$$\widehat{\mathrm{VP}}_{ss} = \frac{\mathrm{OP}_{min} + \mathrm{OP}_{max}}{2} \tag{4}$$

This relation is consistent with the fact that, when a "standard" (symmetrical) data-driven stiction model is used (Kano et al., 2004; He et al., 2007), the valve typically oscillates around its steady-state position and within the two extremes of oscillation of OP, and mean values of valve input and output are really close, that is, $\overline{\mathrm{OP}} \simeq \overline{\mathrm{VP}} \simeq \mathrm{VP}_{ss}$.

3. THE PROPOSED METHOD

The novel PID-based compensation method proposed in this paper is here illustrated.

Algorithm derivation. Valve stiction is a nonlinear phenomenon with memory, which can be efficiently described with data-driven models. In the case of standard model of He et al. (2007), the sticky valve has a nonlinear dynamics $x_k = N_v(x_{k-1}, u_k)$ expressed by the following two relations:

$$x_k = \begin{cases} x_{k-1} + [e_k - \mathrm{sign}(e_k) f_D] & \text{if} \quad |e_k| > f_S \\ x_{k-1} & \text{if} \quad |e_k| \leq f_S \end{cases} \tag{5}$$

where f_S and f_D are static and dynamic friction parameters, respectively, and $e_k = u_k - x_{k-1}$. Note that e_k is a sort of valve position error, and $f_S \geq f_D$ by definition. After some simple algebra, valve dynamics can be rewritten as:

$$x_k = \begin{cases} u_k - f_D & \text{if} \quad u_k - x_{k-1} > f_S \\ u_k + f_D & \text{if} \quad u_k - x_{k-1} < -f_S \\ x_{k-1} & \text{if} \quad |u_k - x_{k-1}| \leq f_S \end{cases} \tag{6}$$

The stiction nonlinearity is thus formed by a set of three, relatively simple, linear and parallel relations, constituting a multi-mode discontinuous model.

Recently, Bacci di Capaci et al. (2017) have proposed and tested three different formulations of model predictive controller (MPC) to address valve stiction. A pure linear formulation, a stiction embedding structure, and a stiction inversion controller were designed. The traditional two-move compensation method was revised and then used as a warm-start to build a suitable trajectory for the stiction embedding MPC, which hence showed good performance, by removing oscillations and producing close-to-zero offset on process variable and very limited valve movements.

By adapting the approach of Bacci di Capaci et al. (2017), the stiction compensation method proposed in this paper employs the following two-move sequence as valve input:

$$\begin{aligned} u_k &= \begin{cases} u_{k-1} + a\hat{f}_S & \text{if} \quad u_{k-1} \geq \hat{x}_{ss} \\ u_{k-1} - a\hat{f}_S & \text{if} \quad u_{k-1} < \hat{x}_{ss} \end{cases} \\ u_{k+1} &= \begin{cases} \hat{x}_{ss} - \hat{f}_D & \text{if} \quad u_{k-1} \geq \hat{x}_{ss} \\ \hat{x}_{ss} + \hat{f}_D & \text{if} \quad u_{k-1} < \hat{x}_{ss} \end{cases} \\ u_{k+j} &= u_{k+1} (= u_{ss}) \qquad \text{if} \quad j > 1 \end{aligned} \tag{7}$$

where $\hat{f}_S$ and $\hat{f}_D$ are the estimates of static and dynamic friction, and $\hat{x}_{ss}$ is the estimate of steady-state position of valve. The *first* input u_k (for $j = 0$) moves the valve away from its stuck position, if $a > 2$, with $a \in \mathbb{R}$. According to (6), it is evident that the maximum value of the difference between valve input and output which does not cause any valve movements is $|u_{k-1} - x_{k-1}| = f_S$. If $u_{k-1} \geq \hat{x}_{ss}$, in the worst case $u_{k-1} - x_{k-1} = -f_S$. Therefore, if $a > 2$, one gets $|u_k - x_{k-1}| > f_S$ and can move the valve: $x_k \neq x_{k-1}$. Then, the *second* signal u_{k+1} (for $j = 1$) brings the valve position to its steady-state value (x_{ss}) in order to eliminate error on control variable. After this second movement ($j > 1$), the valve cannot move from x_{ss}, since the input signal is kept constant. Note that steady-state valve position $\hat{x}_{ss}$ is estimated with (4). Otherwise, it can be evaluated on the basis of the process gain K_P, once identification of whole system of Figure 1 has been performed.

It is to be noted that the proposed methodology has been derived for model of He et al. (2007), but it is also perfectly valid for model of Chen et al. (2008), which represents a suitable extension. Appropriate input sequences similar to (7) can be derived for other types of stiction models, as the one of Kano et al. (2004).

Overall control structure. Another feature of proposed compensator is the ability to address loop perturbations. In particular, load output disturbances can be tackled by monitoring the control error ($e = \mathrm{SP} - \mathrm{PV}$). Note that compensating signal (f_k) is not added to the controller output (u_c), but the valve input (u) is switched between these two signals, as shown in Figure 2.

When sustained oscillations are detected, the compensator is activated by using the moves in (7). Once the steady-state is reached ($\mathrm{SP} \simeq \mathrm{PV}$), eventual external disturbances can be managed. If somehow PV diverges, as the absolute value of error passes a predetermined limit (e_{lim}), the loop is switched back to the standard PI(D) controller.

Then, once the oscillation has returned stable, a certain number

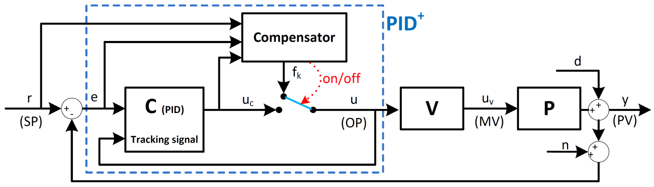

Fig. 2. Structure of the augmented PID controller.

of periods (N_P) are counted before reactivating the compensator. Note that when the compensator takes control, PI(D) controller tracks the valve input, in order to avoid abrupt changes in OP and PV once the control is switched again. Note that with respect to our previous implementation (Bacci di Capaci et al., 2016), set-point changes can be handled without switching to the traditional PID controller, but simply by keeping compensator activated. This feature allows one to avoid awaiting an additional time window of sustained oscillations, provided that a new steady-state for valve input could be suitably computed.

4. SIMULATION ANALYSIS

Some simulation results are briefly reported below.

A benchmark example. This case study has been presented by two previous works (Wang et al., 2015; Bacci di Capaci et al., 2016), so that a direct comparison of performance is possible. The process model is a first order plus time delay (FOPTD):

$$P(s) = \frac{3.8163}{156.46\,s + 1} e^{-2.5s} \tag{8}$$

The PI controller is:

$$C(s) = 0.25 \left(1 + \frac{1}{50\,s} \right) \tag{9}$$

Valve stiction is described by model of Chen et al. (2008), with parameters $f_S = 8.4$ and $f_D = 3.5243$. A white noise with zero-mean and standard deviation $\sigma = 0.01$ is added.

As shown by Bacci di Capaci et al. (2016), the compensator of Wang et al. (2015) employs more than 1100 seconds to complete compensation moves. The mean steady-state error results: $e_{ss} = \mathrm{SP} - \mathrm{PV}_{ss} = -0.157$. Our previous compensator (Bacci di Capaci et al., 2016) performs better, as it saves two moves and needs only 250 seconds. The steady-state valve input and output are $\mathrm{OP}_{ss} = 5.665$ and $\mathrm{VP}_{ss} = 9.189$, respectively. The mean steady-state error results $e_{ss} = -0.023$. Finally, the behavior of the proposed compensator is showed in Figure 3.

By using the proposed method, further two valve movements can be avoided, and a significant time can be saved. As a matter of fact, the novel compensator is instantaneous as it requires only 1 second - one sample period for each move - to impose the desired steady-state position to valve. The steady-state valve input and output are now $\mathrm{OP}_{ss} = 12.696$ and $\mathrm{VP}_{ss} = 9.171$, respectively. The mean steady-state error results $e_{ss} = 0.019$. It is also worth noting that for both previous compensators, a set of additional parameters must be fixed, which are not required instead by the proposed method. Therefore, this new compensation procedure proves to be much simpler and faster than two previous solutions.

Note that the same general result can be obtained by using different values of process, controller, and stiction parameters. As can be seen from (7), the design of the proposed two-move sequence does not depend on process and controller parameters, therefore, they cannot influence the compensation results. In addition, note that proposed compensator has good performance

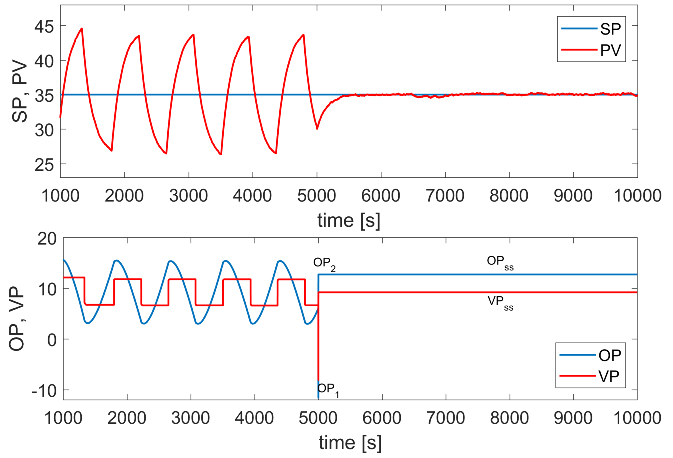

Fig. 3. Results for the proposed compensator.

also for other types of controllers and self-regulating processes. However, it does not work for the case of pure integral processes or open-loop unstable processes, since no steady-state value for PV is permissible in open-loop.

In the case of perturbations. The behavior of the proposed compensator in the presence of loop perturbations is here illustrated. The process model is now a third-order transfer function:

$$P(s) = \frac{1}{(1\,s+1)(5\,s+1)(10\,s+1)} \tag{10}$$

The PI controller is:

$$C(s) = 4\left(1 + \frac{1}{25\,s}\right) \tag{11}$$

Valve stiction is described by model of He et al. (2007), with parameters $f_S = 5$ and $f_D = 2$. A white noise with zero-mean and standard deviation $\sigma = 0.01$ is added. As shown in Figure 4, the compensation is activated in two different occasions, at time $t_{on} = 1125$ and 2650. Three set-point changes occur at time 750, 1500 and 1875. Note that traditional PI controller handles set-point change, but does not remove sustained oscillation. On the opposite, the augmented controller, by keeping compensator active, can both handle set-point changes and remove oscillation. In addition, a step disturbance of amplitude -0.3 affects the output at time 2250. The error limit is set to $e_{lim} = 0.5A_{PV}$, where A_{PV} is the average amplitude of oscillation of PV before the start of compensation. The extremes of oscillation of OP are computed on-line to get estimates of the steady-state valve position VP_{ss} by using (4). As the error limit is violated, the PI controller is resumed. Then, compensation is renewed and the augmented controller removes again the oscillation. In this case, $N_P = 5$ periods of stable oscillation are awaited before compensation restarts. Therefore, the proposed implementation for stiction compensation shows robust performance also in the presence of loop perturbations.

Uncertainty in stiction parameters. It is important to state that the proposed method is strictly dependent on the estimate of stiction parameters ($\hat{f}_S$,$\hat{f}_D$), as seen in (7). In addition, an accurate stiction detection is assumed a priori, since the augmented-PID should not be implemented in the case that oscillations are not due exclusively to valve malfunction. Nevertheless, stiction can be estimated in advance through established identification and quantification methods (Bacci di Capaci and Scali, 2018), which are beyond the scope of this paper. Finally, it is important to stress that the proposed method

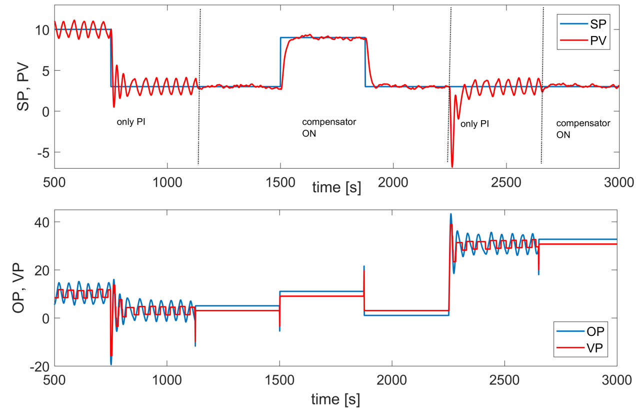

Fig. 4. Stiction compensation in the case of perturbations.

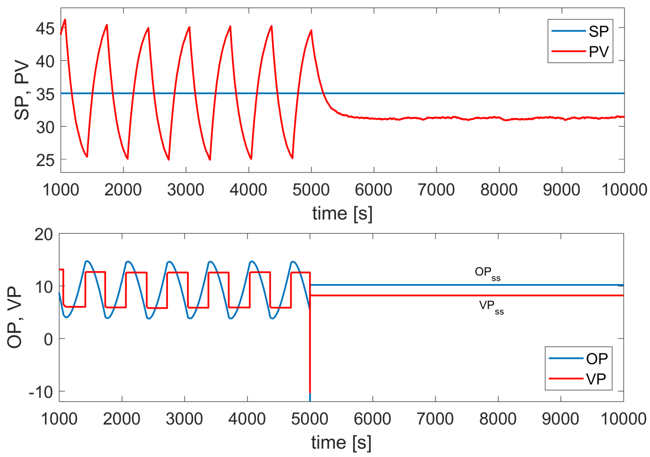

Fig. 5. Results with 50% mismatch in stiction parameter f_D.

is not based on the direct knowledge of valve position VP.
Here the effect of an incorrect estimate of dynamic friction f_D is analyzed. Valve is simulated using a value of $f_D = 2$, while the estimated value for designing the steady-state input (OP_{ss}) is $\hat{f}_D = 1$, that is, a 50% mismatch. The other parameters are set as in the benchmark example. As shown in Figure 5, the compensator cannot bring the PV to the reference, and the mean steady-state error is quite high: $e_{ss} = 3.79$. Note that the steady-state valve position, estimated from (4), is however accurate: $\widehat{\mathrm{VP}}_{ss} = 9.17$. Nevertheless, due to the wrong estimate of f_D, the steady-state valve input is $\mathrm{OP}_{ss} = 10.17$ and the actual final position is lower: $\mathrm{VP}_{ss} = 8.17$. The correct input value ($\simeq 11.17$) should have moved the valve to the proper VP_{ss}, as shown in Figure 3. However, note that this drawback is common to previous two approaches of Wang et al. (2015) and Bacci di Capaci et al. (2016).
Similar problems may arise in the case of uncertainty on static friction f_S, since first move of (7) may not unblock the valve as awaited. To enhance method robustness, the value of parameter a has to be increased, noting that $a = 3$ may be fair choice.

Other possible limitations. Other possible drawbacks of the proposed method are listed below:

- Poor performance could be obtained for the hard case of inhomogeneous stiction. However, when stiction-induced limit cycles arise, the valve operates generally in a small range. Therefore, under closed-loop conditions and without large SP variations, stiction amount can reasonably be assumed to be independent of the valve position.

- Note also that, after the valve is brought to its steady-state position by compensator, the minimum time needed to reach the reference is equal to the settling time of the process. Nevertheless, these two issues are common to all other open-loop compensation methods.
- Finally, the proposed method might show poor performance in cases when perturbations change continuously, e.g. control loops in cascade configuration or in the presence of oscillating or time-varying disturbances, and also when a step perturbation arise exactly during the execution of the compensating moves (7).

5. CONCLUSIONS

In this paper, existing PID-based methods for valve stiction compensation are discussed, and a new stiction-aware PID is proposed. The traditional PID controller is augmented with a novel version of two-move compensator, which is proved to overcome several issues of previous formulations. Two movements in open-loop operation are employed, by causing faster responses as well as complete removal of the oscillation. The control error is monitored to switch to standard PI(D) controller in order to reject external disturbances. Set-point changes can be handled both keeping compensator active or switching back to PID controller. Nevertheless, a reliable detection and a solid estimation of stiction, and robust assessments of steady-state valve position are important prerequisites. Several simulation examples are used to demonstrate the effectiveness of the new method. Implementation of the compensation algorithm on a pilot plant and on industrial control loops, and efforts to overcome residual limitations might feature our future activity.

REFERENCES

Ale Mohammad, M. and Huang, B. (2012). Compensation of control valve stiction through controller tuning. *J. Proc. Contr.*, 22, 1800–1819.

Arifin, B., Munaro, C., Angarita, O., Cypriano, M., and Shah, S. (2018). Actuator stiction compensation via variable amplitude pulses. *ISA Transactions*, 73, 239 – 248.

Bacci di Capaci, R. and Scali, C. (2018). Review and comparison of techniques of analysis of valve stiction: From modeling to smart diagnosis. *Chem. Eng. Res. Des.*, 130, 230–265.

Bacci di Capaci, R., Scali, C., and Huang, B. (2016). A revised technique of stiction compensation for control valves. In *Proc. of 11th IFAC DYCOPS-CAB*, 1038–1043. Trondheim, Norway, 6–8 June.

Bacci di Capaci, R., Scali, C., Pestonesi, D., and Bartaloni, E. (2013). Advanced diagnosis of control loops: Experimentation on pilot plant and validation on industrial scale. In *Proc. of 10th IFAC DYCOPS*, 589–594. Mumbai, India.

Bacci di Capaci, R., Vaccari, M., and Pannocchia, G. (2017). A valve stiction tolerant formulation of MPC for industrial processes. In *Proc. of the 20th IFAC World Congress*, 9374–9379. Toulouse, France, 9–14 July.

Chen, S.L., Tan, K.K., and Huang, S. (2008). Two-layer binary tree data-driven model for valve stiction. *Ind. Eng. Chem. Res.*, 47, 2842–2848.

Cuadros, M.A.d.S.L., Munaro, C., and Munareto, S. (2012a). Improved stiction compensation in pneumatic control valves. *Comput. Chem. Eng.*, 38, 106–114.

Cuadros, M.A.d.S.L., Munaro, C., and Munareto, S. (2012b). Novel model-free approach for stiction compensation in control valves. *Ind. Eng. Chem. Res.*, 51, 8465–8476.

Fang, L., Wang, J., and Tan, X. (2016). Analysis and compensation of oscillations induced by control valve stiction. *IEEE/ASME Transactions on Mechatronics*, 21(6), 2773–2783.

Farenzena, M. and Trierweiler, J. (2010). Modified PI controller for stiction compensation. In *Proc. of 9th IFAC DYCOPS*, 791–796. Leuven, Belgium.

Gerry, J. and Ruel, M. (2001). How to measure and combat valve stiction on line. In *Proc. of the ISA International Fall Conference*, 750–755. Houston, TX, USA, 10–12 September.

Hägglund, T. (2002). A friction compensation for pneumatic control vavles. *J. Proc. Contr.*, 12, 897–904.

He, Q.P., Wang, J., Pottmann, M., and Qin, S.J. (2007). A curve fitting method for detecting valve stiction in oscillating control loops. *Ind. Eng. Chem. Res.*, 46, 4549–4560.

Hidalgo, M.C. and Garcia, C. (2017). Friction compensation in control valves: Nonlinear control and usual approaches. *Control Engineering Practice*, 58, 42–53.

Ivan, L.Z.X. and Lakshminarayanan, S. (2009). A new unified approach to valve stiction quantification and compensation. *Ind. Eng. Chem. Res.*, 48, 3474–3483.

Kano, M., Hiroshi, M., Kugemoto, H., and Shimizu, K. (2004). Practical model and detection algorithm for valve stiction. In *Proc. of 7th IFAC DYCOPS*, 859–864. Boston, USA.

Li, C., Choudhury, M.A.A.S., Huang, B., and Qian, F. (2014). Frequency analysis and compensation of valve stiction in cascade control loops. *J. Proc. Contr.*, 24, 1747–1760.

Mishra, P., Kumar, V., and Rana, K.P.S. (2015). An online tuned novel non linear PI controller for stiction compensation in pneumatic control valves. *ISA Transactions*, 58, 434–445.

Mishra, P., Kumar, V., and Rana, K. (2014). A novel intelligent controller for combating stiction in pneumatic control valves. *Control Eng. Pract.*, 33, 94–104.

Munaro, C.J., de Castro, G.B., da Silva, F.A., Angarita, O.F., and Cypriano, M.V.G. (2016). Reducing wear of sticky pneumatic control valves using compensation pulses with variable amplitude. In *Proc. of 11th IFAC DYCOPS-CAB*, 389–393. Trondheim, Norway, 6–8 June.

Rodríguez, M.C. and Heath, W. (2012). MPC for plants subject to saturation and deadzone, backlash or stiction. In *Proc. of 4th IFAC NMPC Conference*, 418–423. Noordwijkerhout, The Netherlands.

Rohilla, P., Kumar, V., and Al Hakkak, F. (2017). Fuzzy I+PD controller for stiction compensation in pneumatic control valve. *International Journal of Applied Engineering Research*, 12, 3566–3575.

Srinivasan, R. and Rengaswamy, R. (2005). Stiction compensation in process control loops: A framework for integrating stiction measure and compensation. *Ind. Eng. Chem. Res.*, 44, 9164–9174.

Srinivasan, R. and Rengaswamy, R. (2007). Apparatus and method for stiction compensation in a process control system. patent 2007/0088446 a1.

Srinivasan, R. and Rengaswamy, R. (2008). Approaches for efficient stiction compensation in process control valves. *Comput. Chem. Eng.*, 32, 218–229.

Stenman, A., Gustafsson, F., and Forsman, K. (2003). A segmentation-based method for detection of stiction in control valves. *Int. J. Adapt. Control*, 17, 625–634.

Wang, T., Xie, L., Tan, F., and Su, H. (2015). A new implementation of open-loop two-move compensation method for oscillations caused by control valve stiction. In *Proc. of 9th IFAC ADCHEM*, 433–438. Whistler, BC, Canada, 7–10 June.