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A new implementation of open-loop two-move compensation method for oscillations caused by control valve stiction

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Abstract: The open-loop two-move method is an effective compensation to remove oscillations caused by control valve stiction, but takes too much time for implementation. This paper proposes a new implementation of open-loop two-move compensation method based on the estimation of controller output associated with the desired valve position, but without any assumption of the valve position in oscillation. The proposed method can reduce the time cost and lower the amplitude of open-loop step responses, leading to a quicker response of the process and less negative effect during the compensation. The effectiveness of the proposed method is demonstrated by several examples.

Keywords: Two-move; Stiction compensation; Control valve; Implementation; Estimation;

1. INTRODUCTION

It has been reported that 20%-30% of control loops show poor performance due to control valve nonlinearities Paulonis et al. (2003), Srinivasan et al. (2005), Srinivasan et al. (2008). Control valve stiction is one typical reason that leads to sustained oscillations in feedback control loops. The negative effects caused by oscillations lead to poor product process and consume more energy unnecessary. Hence, it's very important to do reliable diagnosis and quantification of the valve stiction from oscillatory signals, which has a large economic impact. However, stiction compensation has rarely been paid attention to.

Current approaches published to compensate oscillations caused by control valve stiction in feedback loops mainly include the knocker, first proposed by Hägglund (2002), the idea is to add short pulses to controller output to keep the control valve moving, and then compensate for the stiction. The knocker requires little about the control loop, but increases the frequency of the movements of the valve, this phenomena reduces valve lifetimes. Another compensation method is the controller tuning Mohammad and Huang (2012), it was implemented by tuning PID parameters according to the situation of the limit cycle to eliminate oscillations or reduce the frequency and magnitude. However, this method is just regarded as the guidelines to analyse the compensation problem for oscillations, for it may affect the steady-state in the control loop if we change the controller type. Srinivasan and Rengaswamy (2005) proposed a two-move method adding two compensation movements to the controller output to avoid the aggressive valve moments, it doesn't need to rely on the controller parameters and won't wear the valve quickly, in despite of requiring more information about the control loop.

This paper is inspired by the three sets of implementation of the open-loop two-move compensation method, in other versions, there are probably more drawbacks existing. A new implementation of open-loop two-move method based on the estimation of the most critical parameter m_{ss} Li Tang. et al. (2014) is proposed, but without the assumption, namely, the value position sticks only two unknown places.

2. REVIEW THE TWO-MOVE COMPENSATOR

Consider a closed control loop depicted in Fig. 1, where $r(t)$, $e(t)$, $m(t)$, $v(t)$, $y_m(t)$ and w(t) are the reference, control

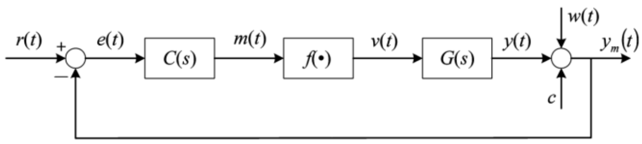

Fig. 1. The diagram of a closed control loop.

error, PID output, valve stem position, measured process output and process noise/disturbance, respectively. The static offset of $y_m(t)$ is c, to be a constant. LTI (linear time-invariant) process $G(s)$ is represented as

$$G(s) = \frac{Ke^{-\theta_0 s}}{\prod_{i=1}^{I}(\tau_{0,i}s + 1)} \quad (1)$$

This process can be approximated by a first-order plus dead time model Seborg, D. et al. (2004),

$$G(s) = \frac{K_p}{T_p s + 1} e^{-\theta s} \quad (2)$$

The controller $C(s)$ is selected a proportional-integral (PI)

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$$C(s) = K_c(1 + \frac{1}{T_i s}) \tag{3}$$

$f(\cdot)$ in Fig. 1 is defined as the valve with stiction in oscillation.

2.1 Modeling of the stiction nonlinearity

Existing data-driven stiction models have been classified comprehensively by Claudio Garcia (2008), among which the model proposed by He et al. (2007) is adopted in this paper. The flowchart of the stiction model is presented in Fig. 2. The parameter f_s and f_d respectively represent the static and kinetic friction bands, m_r is the residual force which hasn't introduced a valve moment, m_{cm} is a current cumulative force acting on the valve.

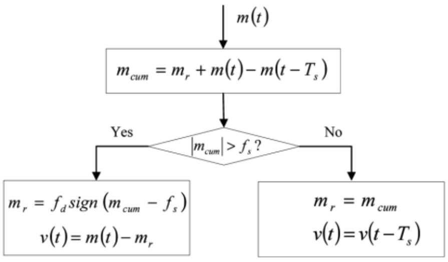

Fig. 2. The flowchart of the stiction model proposed by He et al. (2007).

2.2 Main idea of two-move compensator

This compensator can cause two movements to the valve. First movement, the signal $m(t)$ should be large enough to move the stem from its stuck position. Second, enforce the signal acted on the stem in the opposite direction to the steady-state position v_{ss} associated with the desired setpoint value r_{ss} in order to eliminate the error.

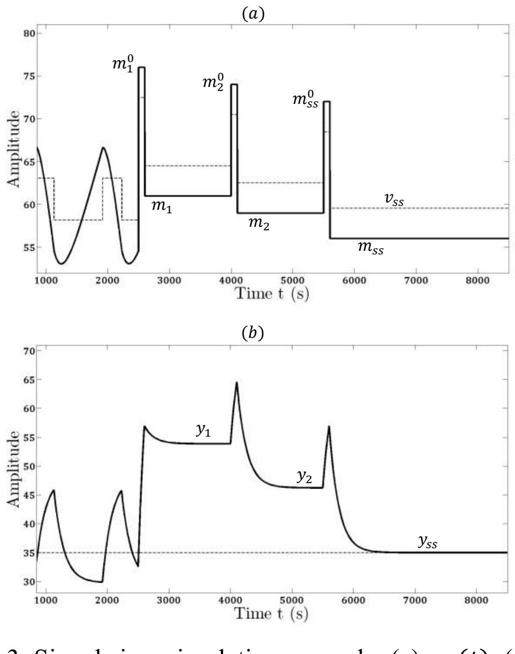

Fig. 3. Signals in a simulation example: (a) $m(t)$ (solid) and $v(t)$ (dash), (b) $y(t)$ (solid) and $r(t)$ (dash)

A simulation example is carried out for illustration. Fig. 3 shows the main idea of three sets of implementation of open-loop two-move method, and oscillations disappear after the last implementation. The variable parameters in these equations are listed as follows:

$$K_p = 3.8163, T_p = 156.46, \theta = 2.5, K_c = 0.25, T_i = 50$$

In the period of two-move compensation, setpoint change or disturbances are not allowed, which is probably the biggest disadvantage. Of course, for this issue, Cuadros et al. (2012) has presented a solution that a certain threshold is set to be compared with the control error $e(t)$, switch the control loop back to the auto mode by monitoring the magnitude of $e(t)$ if it is larger than the threshold. If no accident, Oscillations may appear again in the control loop, then the two-move compensator can be resumed.

This method relies on the value of $m(t)$ associated with the desired valve position v_{ss}, denoted as m_{ss}. If the two-move is realized by increasing $m(t)$ first and decreasing $m(t)$ afterwards, as demonstrated in Fig. 3, a relation can be reached from He's stiction model in Fig. 2.

$$K_p(m_{ss} + f_d) + c = r_{ss} \tag{4}$$

The three implementations of the two-move method introduces two extra open-loop step responses to estimate m_{ss} instead of identification approaches. Then two steady-state equations can be obtained from Fig. 3 where $t \in [2500, 5000]$ sec.

$$K_p(m_1 + f_d) + c = y_1 \tag{5}$$

$$K_p(m_2 + f_d) + c = y_2 \tag{6}$$

Combine (5) with (6)

$$K_p = \frac{y_1 - y_2}{m_1 - m_2}, \quad f_d = \frac{y_1 - c}{K_p} - m_1$$

Due to that the value of K_p, f_d, c have been known, m_{ss} can be calculated from (4)

$$m_{ss} = \frac{r_{ss} - c}{K_p} - f_d$$

Therefore, after the third open-loop step response, the input of the valve is set to be m_{ss} and kept invariant so that the process output can reach the desired point closely. However, considering the actual situation of industry, this compensator takes so long time that the process is in a state of bad performance during the compensation, especially, the process $G(s)$ has a large time constant value. In order to solve this disadvantage, a method was proposed based on estimation of m_{ss} for two-move to compensate oscillations using the oscillatory data samples available in the control loop, without introducing any extra step response. *But it relies on a strong assumption that the valve position $v(t)$ sticks only at two unknown places.*

In despite of the valve position sticking at two places being revealed in most researches, however, this assumption is just validated by observing the valve movement in oscillation. In many industrial occasions, the valve may stick at two more places for some values of f_s and f_d in He's stiction model, or move in a ramp and pause manner if the sampling step of stiction model changes.

3. PROPOSED OPEN-LOOP TWO-MOVE MOTHOD

This section focuses on presentation of an improvement on the estimation of m_{ss} to avoid the above strong assumption, and overcome the long period of time required by open-loop two-move compensation method.

3.1 Preparatory work

When oscillations appear in industrial process control loop, some process parameters can be measured by taking a certain detection technology of oscillations Saneej et al. (2010). Through the analysis of historical data available, we can define several special variables in advance. Since the noise $w(t)$ in Fig. 1 is absent, in one complete oscillation, $max(m(t))$ is the maximum value of controller output $m(\mathrm{t})$, and $min(m(t))$ is the minimum value of $m(\mathrm{t})$. $m(\mathrm{t})$ is decreased first and increased afterwards, T_1 is the time interval that the first peak to valley experiences, T_2 the valley to the second peak that $m(\mathrm{t})$ returns to $max(m(t))$.

The parameters f_s, f_d of He's stiction model can be obtained through the detection and quantification of valve stiction in advance, which is not within the scope of discussion in this paper.

3.2 Steps of proposed open-loop two-move

The proposed open-loop two-move compensation is implemented in the following steps:

1) When the controller output $m(t)$ is in the rising stage close to the peak, switch the control loop into open-loop mode, and enforce $m(t)$ to the maximum value $max(m(t))$, in order to enable the control valve move away from the current sticky position.
2) Enforce $m(t)$ in the opposite direction to the minimum value $min(m(t))$ at the time instant $t_0-\theta$, and $y(t)$ takes the maximum value at the time instant t_0.
3) $m(t)$ returns to $max(m(t))$ at the time instant $t_1-\theta$, where $t_1 = t_0+T_1$. Keep $m(t)$ invariant for the time interval T_2, the time node is $t_2-\theta$, i.e., $t_2 = t_1+T_2$.
4) $m(t)$ is driven to $min(m(t))$ again. In the meantime, the minimum of $y(t)$ is recorded, i.e., $y(t_1) = min_t(y(t))$, and $y(t_0)=y(t_2)=max_t(y(t))$. The principle is the same in the declining stage.

In simple terms, the purpose of above process is to ensure a case that the valve position $v(t)$ is bound to stick only two places. The selection of $m(t)$ is also to ensure the valve's being moved, but not too large. It is easy to see the actual movement of $m(t)$.

$$m(t_0)=m(t_2)=min(m(t)) \tag{7}$$

$$m(t_1)=\max(m(t)) \tag{8}$$

which is a very simple relationship, it doesn't involve any complicated calculation. Next, the differential equation corresponding to the process model $G(s)$ in (2) by deriving the expression of $y(t)$ is

$$y^{(1)}(t)=\frac{K_p}{T_p}v(t-\theta)-\frac{1}{T_p}y(t) \tag{9}$$

The valve position $v(t)$ takes a constant value denoted as v_1 for $t\in[t_0-\theta,t_1-\theta)$, while $v(t)$ stays at another value denoted as v_2 for $t\in[t_1-\theta,t_2-\theta)$. Thus, taking the nonzero initial value of $y(t)$ at t_0 into consideration, the Laplace transformation of (9) for $t\in[t_0,t_1]$ is

$$sY(s)-y(t_0)=\frac{K_pv_1}{T_ps}-\frac{1}{T_p}\mathrm{Y(s)}$$

leading to

$$Y(s)=\frac{K_pv_1}{T_ps\left(s+\frac{1}{T_p}\right)}+\frac{y(t_0)}{s+\frac{1}{T_p}}$$
$$=\frac{K_pv_1}{s}+\frac{y(t_0)-K_pv_1}{s+\frac{1}{T_p}} \tag{10}$$

Equation (10) yields the time-domain representation of $y(t)$ for $t\in[t_0,t_1)$ as

$$y(t)=K_pv_1+(y(t_0)-K_pv_1)e^{-\frac{t-t_0}{T_p}} \tag{11}$$

Similarly, $y(t)$ for $t\in[t_1,t_2)$ is

$$y(t)=K_pv_2+(y(t_1)-K_pv_2)e^{-\frac{t-t_1}{T_p}} \tag{12}$$

then (11), (12) respectively give

$$y(t_1)=K_pv_1+(y(t_0)-K_pv_1)e^{-\frac{T_1}{T_p}}, \tag{13}$$

$$y(t_2)=K_pv_1+(y(t_1)-K_pv_1)e^{-\frac{T_2}{T_p}}, \tag{14}$$

Where $T_1=t_1-t_0$, $T_2=t_2-t_1$. Substituting (11), (12) and $r_{ss}=K_pv_{ss}+c$ into

$$\int_{t_0}^{t_0+T_1}(r_{ss}-y(\tau)-c)d\tau+\int_{t_1}^{t_1+T_2}(r_{ss}-y(\tau)-c)d\tau=0$$

The above formula can be proved by the Laplace transformation of a differential equation associated with the controller $C(s)$ in (3), which can be finally rewritten as

$$K_pv_{ss}(T_1+T_2)-K_p(v_1T_1+v_2T_2)=$$
$$T_p\big(y(t_0)-K_pv_1\big)\left(1-e^{-\frac{T_1}{T_p}}\right)+$$
$$T_p\big(y(t_1)-K_pv_2\big)\left(1-e^{-\frac{T_2}{T_p}}\right) \tag{15}$$

get (13), (14) together with the fact $y(t_0)=y(t_2)$, then the left-hand side of (15)

$$K_p v_{ss}(T_1+T_2) - K_p(v_1T_1+v_2T_2) = 0,$$

So it becomes

$$v_{ss}(T_1+T_2) - (v_1T_1+v_2T_2) = 0 \tag{16}$$

Based on He's data-driven model and the positive gain K_p,

$$v_1 = m(t_0) + f_d \tag{17}$$

$$v_2 = m(t_1) - f_d \tag{18}$$

so that (16) yields

$$v_{ss} = \frac{m(t_0-\theta)T_1 + m(t_1-\theta)T_2}{T_1+T_2} + f_d\frac{T_1-T_2}{T_1+T_2}$$
$$= \frac{m_0T_1+m_1T_2}{T_1+T_2} + f_d\frac{T_1-T_2}{T_1+T_2}$$
$$= \frac{min(m(t))T_1 + max(m(t))T_2}{T_1+T_2} + f_d\frac{T_1-T_2}{T_1+T_2} \tag{19}$$

Here $min(m(t))$, $max(m(t))$ and f_d are known, the desired valve position v_{ss} can be estimated easily, and we can save many steps as well to focus on the emphases of open-loop two-move method, without worrying about the assumption of the valve movement in oscillation.

If $m(t)$ is increased first and decreased afterwards, then the m_{ss} is written as following equality according to He's data-driven model,

$$m_{ss} = v_{ss} + f_d \tag{20}$$

In reverse, if the two-move method is implemented in an opposite direction, i.e., $m(t)$ is decreased first and increased afterwards, the m_{ss} is

$$m_{ss} = v_{ss} - f_d \tag{21}$$

Hence, the two-move compensator needs only another implementation, and reduces the time cost.

Finally, the implementation of the proposed two-move compensation has the last step: If $m(t)$ is increased (decreased) first and decreased (increased) afterwards, obtain m_{ss} from (20) (or (21)), and set $m(t)$ to the value $m(t_{switch})$ in (22) at a time instant t_{switch}, and then hold $m(t)$ unchanged for a time duration T_{hold}, i.e, $m(t) = m(t_{switch})$ for $t \in [t_{switch}, t_{switch}+T_{hold}]$. After that, we can directly set $m(t)$ to the value m_{ss} so that $y(t)$ is expected to approach the desired setpoint value r_{ss} at the steady state

$$m(t_{switch}) = m_{ss} + a(m_1 - m_0) \tag{22}$$

The selection of t_{switch} doesn't have to be a specific time instant, only a time instant to ensure that $y(t)$ has changed direction. Here a is just a coefficient to enable the valve overpass the stiction band. T_{hold} should not be too large to move $y(t)$ deviated much from the setpoint value r_{ss}, which is undesirable, so T_{hold} should be as small as possible. However, the values of $m(t_{switch})$ and T_{hold} are not confined to the recommended ones.

4. SIMULATION EXAMPLE

Two simulation examples are provided to illustrate the proposed compensation method in this section. One of them is without any noise, another with white noise. The closed control loop described in Fig. 1 is formulated.

The process model $G(s)$ is

$$G(s) = \frac{3.8163}{156.46s+1}e^{-2.5s}$$

The PI controller $C(s)$ is

$$C(s) = 0.25(1+\frac{1}{50s})$$

The setpoint $r(t)$ is kept a constant $r_{ss} = 35$, the static offset constant $c = -192.3411$, and $w(t)$ is absent. It's to be found that $T_1 = 307$ sec, $T_2 = 788$ sec via the detection of oscillations, so the time instants $t_0 = 4100$ sec, $t_1 = 4888$ sec, and $t_2 = 5195$ sec. The input value of the control valve is $m_0 = m_2 = 53.3$, $m_1 = 66.7$, respectively. The stiction model parameters $f_s = 8.4$, $f_d = 3.5243$ of He's stiction model are given.

In this simulation, $m(t)$ is decreased first and increased afterwards, then the estimate of m_{ss} is obtained via (21) as $m_{ss} = 51.6$. Before setting $m(t)$ to the value m_{ss}, $m(t)$ is firstly increased to the value $m(t_{switch}) = 70.9$ via (22) at the time instant $t_{switch} = 5400$ sec, secondly kept at this value for the duration $T_{hold} = 10$ sec, and finally decreasing $m(t)$ to the value $m_{ss} = 56.1$. The compensation results are presented in Fig. 4.

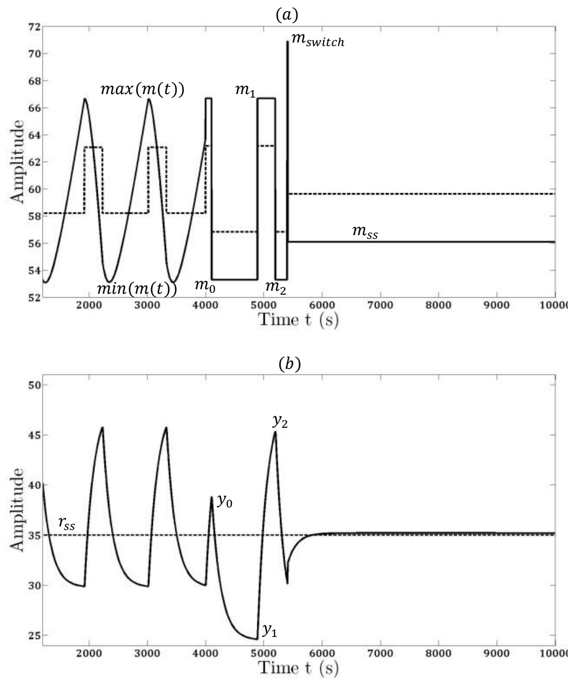

Fig. 4. Signals in simulation example: (a) $m(t)$ (solid) and $v(t)$ (dash), (b) $y(t)$ (solid) and $r(t)$ (dash)

In order to verify the feasibility of this method in practice, on the basis of the above simulation, add white noise to the output of the process.

First of all need to make sure that the energy of white noise cannot change the oscillation too much before introducing compensator in the control loop. Open-loop two-movement compensation is different from the other compensation methods in essence, for during the compensation, it needs to maintain the valve at its steady state position, corresponding to the setpoint. So if the noise is too large, the compensation effect is not obvious, which is the cancer of two-movement compensation itself.

In second simulation, the white noise power is 0.01 and connected a proportional gain $K_{noise} = 0.013$, other parameters unchanged. The compensator's control effect is presented as follows:

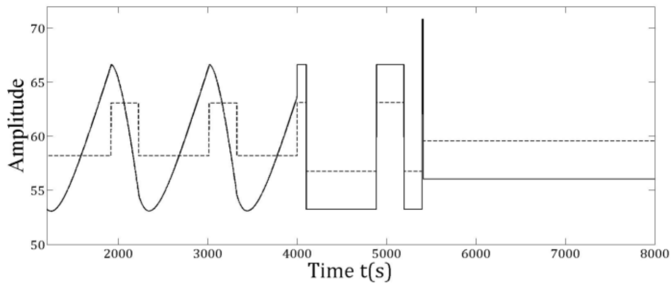

Fig. 5.1. Signals in simulation example with white noise: $m(t)$ (solid) and $v(t)$ (dash)

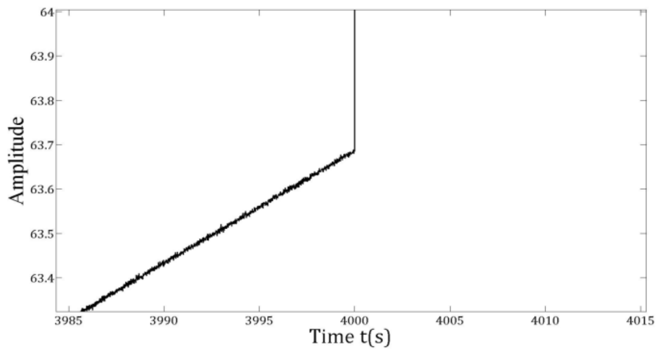

Fig. 5.2. Zoomed t: [3985 4015] in Fig. 5.1.

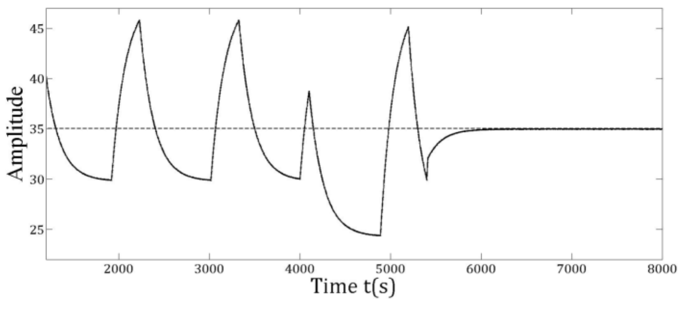

Fig. 5.3. Signals in simulation example with white noise: $y(t)$ (solid) and $r(t)$ (dash)

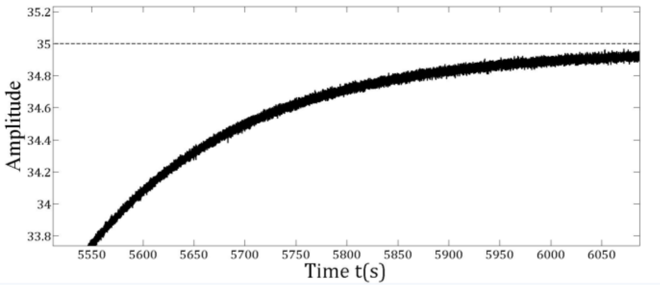

Fig. 5.4. Zoomed t: [5500 6100] in Fig. 5.3.

When K_{noise} increases to 0.02, $y(t)$ starts to deviate from setpoint. It can be concluded that the implementation of the proposed two-move compensation is able to adapt to a smaller noise, however, in the face of a large noise level, this method presents its disadvantage.

5. LABORATORY EXAMPLE

This section provides a laboratory example carried out at Zhejiang University to illustrate the proposed compensation method.

This laboratory example is a feedback control loop for a water tank system, where the water level controlled by adjusting the inlet flow via a control valve driven by a PI controller is the process output, while the outlet valve has a fixed opening position.

Here the PID controller,

$$C(s) = 0.25(1 + \frac{1}{50s})$$

control valve is pneumatic whose sticion is introduced by tightening the valve stem packing screw, the sticiton model is depicted by He model, $f_s = 7.9$, $f_d = 3.1464$. The setpoint $r(t)$ is also kept at a constant value $r_{ss} = 35$ and invariant during compensation. Via a preliminary experiment, e.g., the offset constant $c = -196$. Via the detection of oscillations, $T_1 = 310$ sec, $T_2 = 792$ sec.

The steps of the proposed implementation of open-loop two-movement method are done as same as Section 4. At the time $t = 4000$ sec, the control loop is switched to open loop control mode. However, in order to increase the accuracy of the experiment, based on the most simple implementation of the proposed method, an extra compensation signal is introduced to ensure that the water level presents a stable oscillation during the period of compensation, lasts T_2 sec. So the time instants $t_0 = 4892$ sec, $t_1 = 5202$ sec, and $t_2 = 5994$ sec. The input of control valve changes in two values only, $m_{min} = 54.2$, $m_{max} = 67.7$.

In this laboratory example, $m(t)$ is increased first and decreased afterwards, then the estimate of m_{ss} is obtained via (21) as $m_{ss} = 63.8$. Before setting $m(t)$ to the value m_{ss}, $m(t)$ is firstly decreased to the value $m(t_{switch}) = 45$ via (22) at the time instant $t_{switch} = 6188$ sec, secondly kept at this value for the duration $T_{hold} = 10$ sec, and finally increasing $m(t)$ to the value $m_{ss} = 63.8$. The compensation results are presented in Fig. 6.

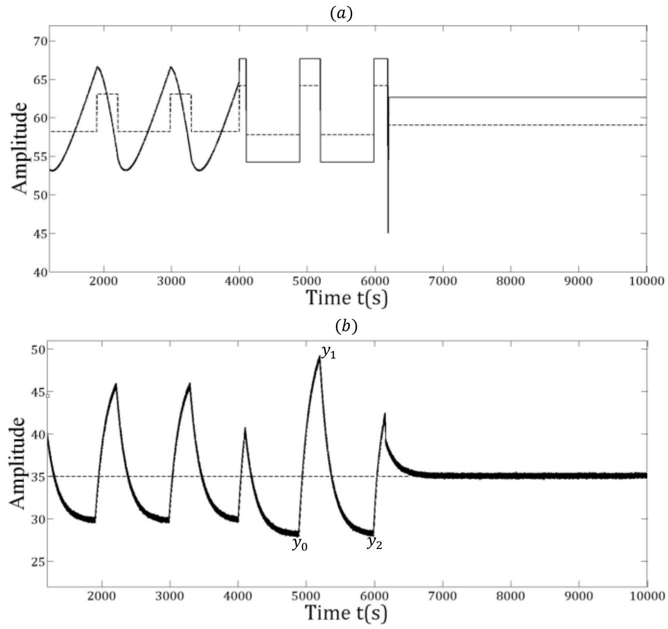

Fig. 6. Signals in laboratory example: (a) $m(t)$ (solid) and $v(t)$ (dash), (b) $y(t)$ (solid) and $r(t)$ (dash)

6. CONCLUSIONS

In this paper, a new implementation of open-loop two-move compensation method for oscillations caused by control valve stiction is proposed. This method avoids validating the assumption, namely, the value position sticks only two unknown places, and it reduces the time cost of open-loop two-move, and also produces less negative effect on the performance of the control loop during the compensation. Without any assumption, the proposed method can be applied more widely in practice. A laboratory example and two simulations are used to demonstrate the optimization approach. Future work will include the implementation of the proposed open-loop two-move stiction compensation on several industrial control loops.

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