Compensating the Effect of Control Valve Stiction

by

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CERTIFICATION OF THESIS WORK

The thesis titled **"Compensating the Effect of Control Valve Stiction"** submitted by **Ahaduzzaman** Roll No. **1015022040P** Session **October, 2015** has been accepted as satisfactory in partial fulfillment of the requirement for the degree of **Master of Science in Engineering (Chemical)** on **June 3, 2018.**

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DECLARATION

The thesis on "Compensating the effect of Control Valve Stiction" completed by Ahaduzzaman has been submitted in partial fulfillment of the degree of Master of Science in Chemical Engineering. This thesis is my original work and wherever contributions of others are involved, every effort is made to indicate this clearly, with due reference to the literature, and acknowledgement of collaboration is made explicitly.

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In my capacity as supervisor of the candidate's thesis, I certify that the above statements are true to the best of my knowledge.

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ABSTRACT

Valve stiction is the hidden culprit of the process control loop. The presence of stiction in a control valve limits the control loop performance. It is the most commonly found problem in pneumatic control valves. It decreases the control loop efficiency and causes oscillations in process variables. Repair and maintenance is the only definitive solution to fix a sticky valve. However, this fact implies to stop the operation of the control loop, which is only possible during plant shutdown. Compensation of its effect is beneficial before the sticky valve can be sent for maintenance. In this study, a new stiction compensation method has been developed by adding an extra pulse for a certain period of time to the detuned controller action. A method for estimating the parameters required for the proposed compensator has also been developed. The performance of the proposed stiction compensation scheme is evaluated using MATLAB Simulink software. The developed compensator has the desired capability of reducing the process variability with a minimum number of valve reversals. It has the capability of good set-point tracking and disturbance rejection. The performance of the proposed compensator has been compared with other compensators available in the literature. The proposed compensator outperforms all of other compensators. The proposed compensator has been implemented to a level control loop in a pilot plant experimental set-up. It has been found to be successful in removing valve stiction induced oscillations from process variables. Moreover, the proposed compensation scheme is simple. It requires minimum process knowledge. It would not be difficult to implement it in real process plants.

TABLE OF CONTENTS

Contents **Page No.**

Contents **Page No.**

Contents **Page No.**

Contents **Page No.**

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

G_P	Process Transfer Function
G_C	Controller Transfer Function
G_k	Knocker Transfer Function
G_f	Filter Transfer Function
$y_{sp}(t)$	Desired Set-Point
$y(t)$	Process Output
$u_k(t)$	Knocker Output
h_k	Pulse Period
τ_d	Pulse Width
d	Stiction Parameter
$u(t)$	Summation of Knocker and Controller Output
u_c	Controller Output
OP	Compensator Output
PV	Process Variable
SP	Set-Point
MV	Valve Position
S	Deadband plus Stickband
J	Slip Jump
K_P	Process Gain
τ	Time Constant
θ	Time Delay
K_C	Controller Gain
T_s	Sample Time
T_P	Time of One Set of Previous Pulse
T_G	Specific Time during which Pulse Generator Remains Activated
ϵ	Permissible Error
e	Difference Between Set-point and Process Output, (*SP*-*PV*)
A_P	Pulse Amplitude
α	Detune Parameter
N	Total Number of Data Points

μ	Mean Value of Process Variable
x	Valve Position
σ^2	Process Variable Variance
η	Performance Index
P(s)	Process Transfer Function
C(s)	Controller Transfer Function

ABBREVIATIONS

EWMA	Exponentially Weighted Moving Average
FOPTD	First Order Plus Time Delay
IAE	Integral Absolute Error
IMC	Internal Model Control
PI(D)	Proportional Integral
PID	Proportional Integral Derivative
VT	Valve Travel
SISO	Single Input Single Output
SSE	Summation of Square Error

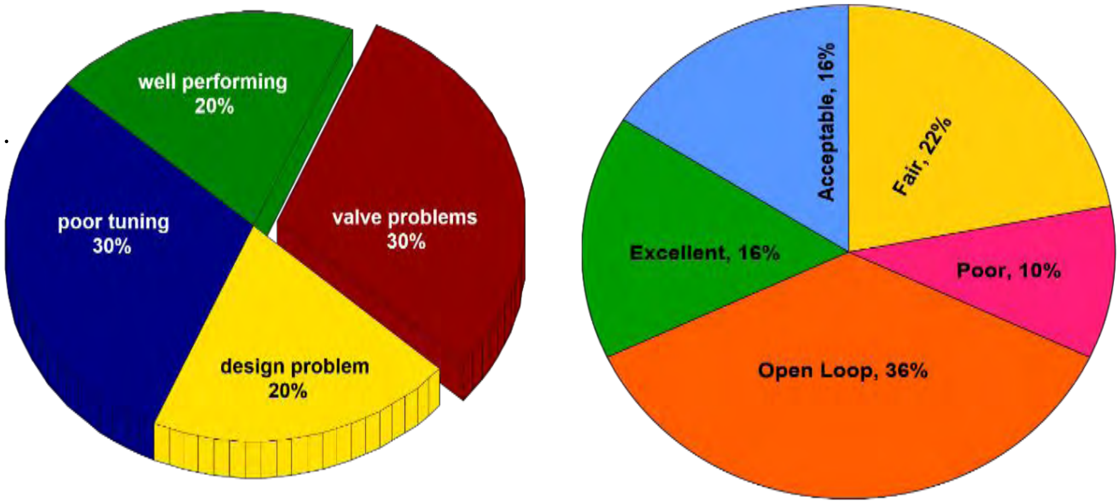
CHAPTER 1

INTRODUCTION

1.1 Background

Constrained resources, stringent environmental regulations and tough business competition have resulted in efficient manufacturing operations in terms of energy usage, raw material utilization, superior quality products and plant safety. Most of the modern plants are now automated to achieve these goals. Control loops are the essential part of these automated processes. Large-scale, highly integrated processing plants include hundreds or thousands of such control loops.

The aim of each control loop is to maintain the process at the desired operating conditions safely and efficiently. A poorly performing control loop can result in disrupted process operation, degraded product quality, higher material or energy consumption. Thus the poor performance of the control loop decreases plant profitability. Control loop performance has been an active research area for academia and industry for the last three decades. Control loops often suffer from poor performance due to process non-linearities, process disturbances, poorly tuned controllers and misconfigured control strategies. Performance of over 26,000 PID controllers from a wide range of continuous process industries was investigated by Desborough, et al. (2001). It was found that the performance of over two thirds of control loops was not satisfactory. Another survey by Bialkowski (1993) also reported that only one third of industrial controllers provided acceptable performance. These survey results are shown in Figure 1.1, where the left panel shows the survey result of Bialkowski (1993) and the right panel shows the survey result of Desborough et al. (2001).



Bialkowski (1993) Desborough et al. (2001)

Figure 1.1: Global Multi-Industry Performance.

Oscillatory variables are one of the main causes for poor performance of control loops. The presence of oscillations in a control loop increases the variability of the process variables, which makes it difficult to keep operating conditions close to their bounds. The reason for oscillations in a control loop may be due to poor controller tuning, poor process and control system design, valve non-linearities, oscillatory disturbances and other causes (Choudhury et al., 2005). Among various valve nonlinearities, stiction is the most commonly encountered one. Hence, it is practically very important to find the loops where control valves are sticky and thereby compensate stiction to reduce its negative effect.

1.2 What is Stiction?

The word "Stiction" comes by combining two words-Static and Friction. Stiction is the static friction that keeps an object from moving. When the external force to the object overcomes the static friction, it starts moving. Often stiction is confused with some similar problems such as backlash, hysteresis, deadband and deadzone. Therefore, these terms are defined below for a better understanding of the term 'stiction'.

1.2.1 Definition of Terms Relating to Valve Stiction

According to the Instrument Society of America (ISA) (ISA Committee SP51, 1979) the definitions of the terms backlash, hysteresis, deadband and deadzone are as follows:

Backlash: "In process instrumentation, it is a relative movement between interacting mechanical parts, resulting from looseness, when the motion is reversed".

Hysteresis: Hysteresis is that property of the element evidenced by the dependence of the value of the output, for a given excursion of the input, upon the history of prior excursions and the direction of the current traverse. It is usually determined by subtracting the value of deadband from the maximum measured separation between upscale going and downscale going indications of the measured variable (during a full range traverse, unless otherwise specified) after transients have decayed". Figure 1.2(a) and 1.2(c) illustrate the concept. Some reversal of output may be expected for any small reversal of input. This distinguishes hysteresis from deadband.

Deadband: *"*In process instrumentation, it is the range through which an input signal may be varied, upon reversal of direction, without initiating an observable change in output signal". There are separate and distinct input-output relationships for increasing and decreasing signals (Figure1.2 (b)). Deadband produces phase lag between input and output". Deadband is usually expressed in percent of span. Deadband and hysteresis may be present together. In that case, the characteristics in Figure 1.2(c) will be observed

Deadzone: "It is a predetermined range of input through which the output remains unchanged, irrespective of the direction of change of the input signal". "There is but one input-output relationship as shown in Figure 1.2(d). Deadzone produces no phase lag between input and output.

The above definitions show that the term "backlash" specifically applies to the slack or looseness of the mechanical part when the motion changes its direction. Therefore, in control valves it may only add deadband effects if there is some slack in rack-and-pinion type actuators. Deadband is quantified in terms of input signal span (i.e., on the x-axis) while hysteresis refers to a separation in the measured output response (i.e., on the y-axis).

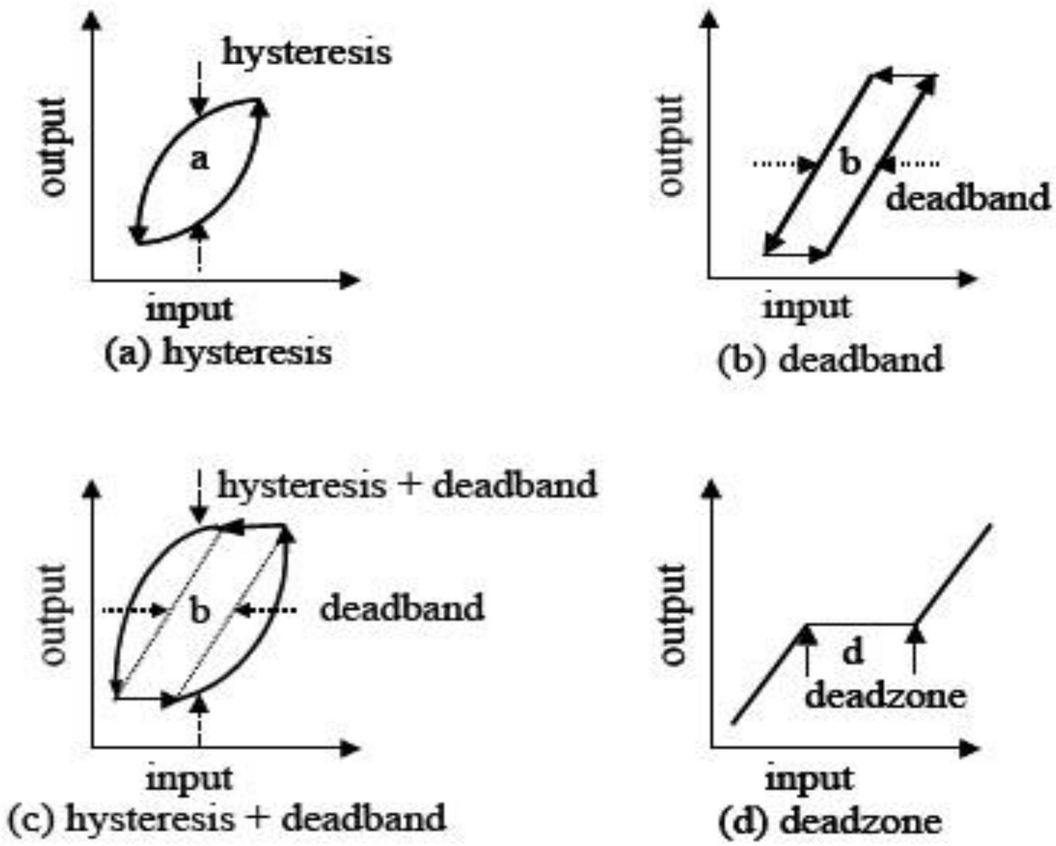


Figure 1.2 Typical input–output behavior of hysteresis, dead band and dead zone (ISA Committee SP51, 1979).

1.2.2 Mechanism and Definition of Stiction

As discussed above, the Instrument Society of America (ISA) (1979) provided the phase plots for the hysteresis, deadband and deadzone as shown in Figure 1.2. There was no such phase plot for stiction. Choudhury et al. (2005) provided the phase plot of the input–output behavior of a valve suffering from stiction as shown in Figure 1.3. It consists of four components: deadband, stickband, slip jump and the moving phase. When the valve comes to restore changes the direction at point A in Figure 1.3, the valve becomes stuck. The valve output remains same though the valve input keeps changing. The input of the valve is generally the controller output. After the controller output (valve input) overcomes the deadband (AB) and the stickband (BC) of the valve, the valve jumps to a new position (point D) and continues to move. Due to very low or zero velocity, the valve may stick again in between points D and E while travelling in the same direction. In such a case, the magnitude of deadband is zero and only stick band is present. This can be overcome if the controller output signal is larger than the stickband only (Choudhury, et al., 2005). The deadband and stickband represent the behavior of the valve when it is not moving, though the input to the valve keeps changing. Slip jump represents the abrupt release of potential energy stored in the actuator chambers due to high static friction in the form of kinetic energy as the valve starts to move. The magnitude of the slip jump is very crucial in determining the limit cyclic behavior introduced by stiction. Once the valve slips, it continues to move until it sticks again (point E in Figure 1.3). In this moving-phase, dynamic friction is present which may be much lower than the static friction. When the controller signal changes its direction, the valve behavior would be same as before in reverse direction to the path EFGH in Figure 1.3. Thus stiction is defined as a property of an element such that its smooth movement in response to a varying input is preceded by a sudden abrupt jump called the slip-jump. Slip-jump is expressed as a percentage of the output span. Its origin in a mechanical system is static friction which exceeds the dynamic friction during smooth movement.

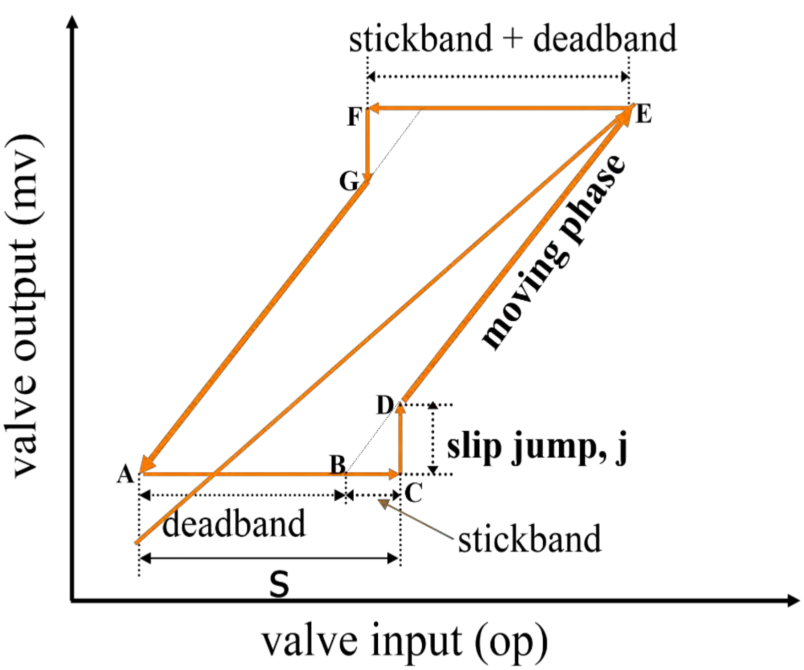


Figure 1.3: Typical input–output behavior of a sticky valve (Choudhury et al., 2005).

In industry, stiction is measured as a certain percentage of the valve travel or the span of the control signal. For example, 2% stiction means that when valve gets stuck it will start moving only after the cumulative change of its control signal is greater than or equal to 2%. If the range of the control signal is 4 to 20 mA then 2% stiction means a cumulative change of the control signal less than 0.32 mA in magnitude will not be able to move the valve.

1.2.3 Where does Stiction Occur in Control Valve?

The cross- sectional diagram of a control valve is shown in Figure 1.4.

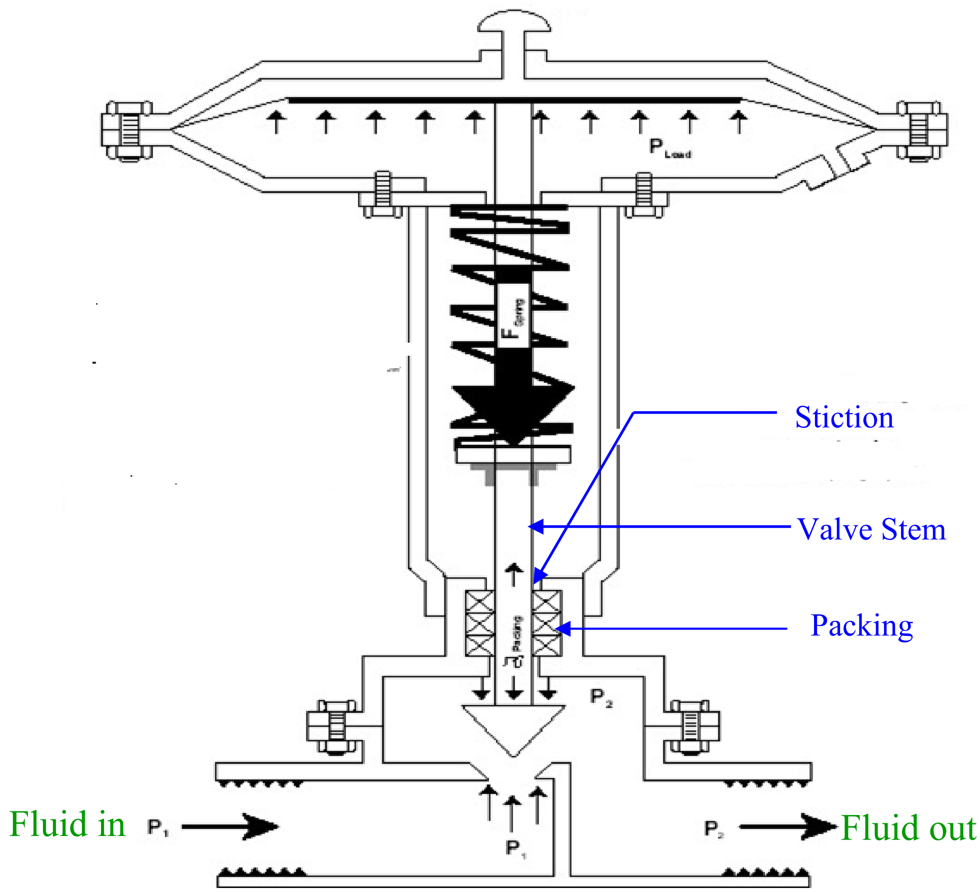


Figure 1.4: A cross sectional diagram of a spring-diaphragm pneumatic control valve (Choudhury et al., 2008).

As shown in Figure 1.4, the valve stem moves up and down through the packing box to restrict the flow of process fluid through the pipe. The packing stops process fluid from leaking out of the valve but the valve stem nevertheless has to move freely relative to the packing. There is a trade-off because too tight packing reduces emissions and leaks from the valve but at the same time increases the friction. Loose packing reduces the friction but there is a potential for process fluids to leak. Stiction happens when the smooth movement of the valve stem is hindered by excessive static friction at the gland packing section (Arumugam et al., 2014). So, stiction appears in the packing boxes around the valve stem (Choudhury et al., 2005). There is a possibility of leaking the fluid in hard packing. To avoid leakage, the packing boxes are often tightened after some period of operation. Stiction can appear due to tightened hard packing in the control valve.

In ball valves, ball segment valves, and throttle valves there is often also a significant friction between the ball/throttle and the seat. The friction in the pilot valve may also increase and cause problems if the air is polluted. Hysteresis may appear at several places in the mechanical configuration due to wear and vibrations. The stiction varies both in time and between different operating points. Temperature variations cause friction variations. A high temperature means that the material expands, and therefore the friction force increases. Some media give fouling that increases the friction. Particles in the media may cause damage on the valve. The wear is often non-uniform. Therefore, friction is different at different valve positions. Experimental investigations showed that the force required to overcome stiction is dependent on the rate at which the force is applied.

Stiction is a major problem in control valve. It reduces the performance of the control loop. Compensation of such problem is the main objective of this research work.

1.3 Stiction Compensation

Control valve stiction compensation is an active area of research in the literature to increase the control loop performance (Arifin et al., 2014). To deal with the valve stiction problem, the very first step is to detect whether a control valve is sticky and then to quantify the severity of the stiction. Repair and maintenance must be considered the only definitive solution to fix a sticky valve. However, this fact implies to stop the operation of the control loop, which is only possible during plant shutdown. Since the plant overhauling takes place generally every two to three years, compensation of stiction can be a useful alternative to mitigate the negative effects of stiction until the next shutdown. Therefore, methods for compensating the effect of stiction are of great importance to avoid unscheduled plant shut-down.

A good stiction compensator should have the following characteristics (Souza et al., 2012):

a) Reduction of oscillations in process variables
b) Reduction of valve movements or minimizing valve reversals
c) No requirement of prior process knowledge except for routinely available operating data and
d) Ensuring good set point tracking and disturbance rejection.

None of the current stiction compensation methods available in the literature can fulfill these requirements. In this study, a simple yet powerful stiction compensation method meeting all above criteria has been developed. It has been evaluated successfully both in simulation and laboratory experiments.

1.4 Objectives of the Study

The objectives of this study is

a) Developing a new stiction compensation technique and comparing the proposed techniques with different available compensation techniques in the literature.

b) Evaluating the performance of proposed technique in both simulation and experimental cases.

1.5 Outline of the thesis

Chapter 1 is the introduction to the thesis. It describes the background, objectives and outline of the thesis.

Chapter 2 reviews the compensation techniques available in the literature.

Chapter 3 describes the proposed stiction compensation method for a sticky control valve.

Chapter 4 evaluates the performance of the proposed compensation method.

Chapter 5 compares the proposed stiction compensation technique with some other compensation methods available in the literature.

Chapter 6 describes the validation of the proposed compensator by implementing it in a pilot plant.

Finally, **Chapter 7** draws the conclusion and recommendation for future work.

1.6 Chapter Summary

The research work studies the compensation of stiction problem in control valves. The definition and mechanism of stiction is briefly discussed in this chapter. This chapter also presented the objectives and outline of the thesis.

CHAPTER 2

LITERATURE REVIEW

Control valve stiction compensation is an active area of research for the last two decades. Repair and maintenance are the most effective solutions for a sticky valve. However, these actions may not be feasible between scheduled plants shut-down. Therefore, as a matter of principle, stiction compensation can be a valid alternative to mitigate its negative impact on loop performance. It can help to minimize the effect of stiction up to the next process shutdown. Therefore, the methods for compensating stiction are of great importance to avoid unscheduled plant shutdown. There are many methods for detection and quantification of stiction but only a few for stiction compensation (Arifin et al., 2014). Among the available methods of compensation, the most commonly used methods are described in this section.

2.1 The Knocker Method

The idea behind the stiction compensation procedure proposed by Hägglund (2002) is to add short pulses of equal amplitude and duration to the control signal to move the valve from stuck position. The direction of the pulse signal depends on the rate of change of the control signal. Each pulse has an energy content that is used to compensate the effect of stiction in control valve. With a lower energy content, the valve will remain stuck. With a higher energy content, the valve slip will be larger than desired. This method is known as knocker method. The principle of the knocker method is illustrated in Figure 2.1.

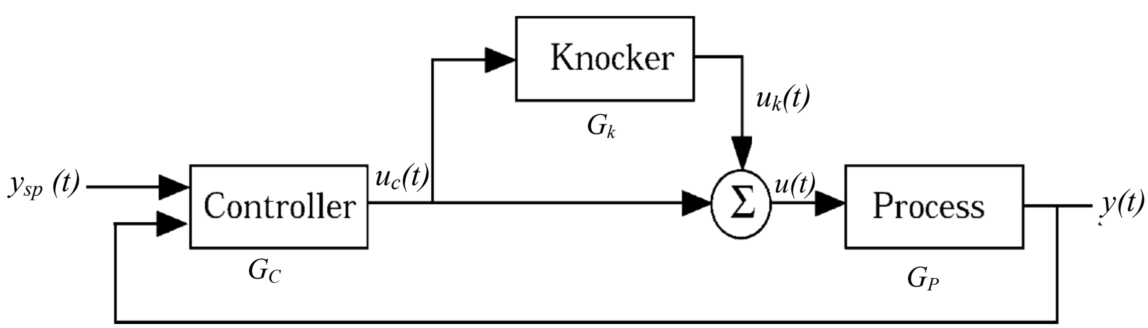


Figure 2.1: Block diagram illustrating the knocker used in a feedback loop.

The input signal of valve, $u(t)$ consists of two terms:

$$u(t) = u_c(t) + u_k(t) \tag{2.1}$$

Where $u_c(t)$ is the output from a standard controller, and $u_k(t)$ is the output from the knocker. Output $u_k(t)$ from the knocker is a pulse sequence that is characterized by three parameters:

The time between each pulse is h_k, the pulse amplitude is a and the pulse width is τ. Figure 2.2 shows a typical pulse sequence.

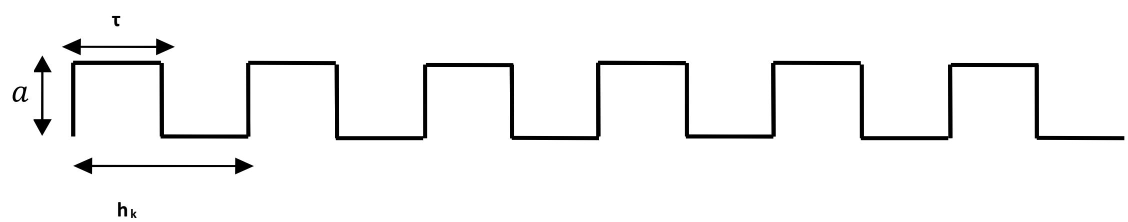


Figure 2.2: Pulse sequence characterization of Knocker method.

During each pulse interval, $u_k(t)$ is given by:

$$u_k(t) = \begin{cases} a\ sign\left(u_C(t) - u_C(t_p)\right), & t \leq t_p + h_k + \tau \\ 0 & , t > t_p + h_k + \tau \end{cases} \tag{2.2}$$

Where t_p is the time of onset of one previous pulse. Hence, the sign of each pulse is determined by the rate of change of control signal $u_c(t)$.

The knocker method are considered the simplest compensation methods. They can achieve higher reduction in the output variability. This method removes oscillations at the cost of a faster and wider motion of the valve stem. Excessive valve stem movements reduce the longevity of the valve. This may lead to frequent maintenance actions and unavailability of the plant.

2.2 Two Move Method

To avoid the aggressive valve movements in the Knocker method (Hägglund, 2002), Srinivasan and Rengaswamy (2008) proposed a two move method for stiction compensation. The two move method adds two compensation movements to the controller output in order to make the control valve eventually arrive at a desired steady-state position.

Stiction prevents the valve stem from reaching its final steady state position, instead it makes the stem jump around it. This jumping behavior continues between two positions, one above and another below the steady state position. If stiction does not occur and enough time is given to the transient to die out, the process variable, control signal and valve stem position will reach their final steady state values.

From these observations, the authors claimed that if a compensation signal can be added to force the valve stem to reach its steady state position, the controller can achieve the desired process variable value, provided no further set-point change or disturbance occurs during that period. To accomplish this, at least two moves are necessary. The first move is used to push the stem to a steady state position and the second move to force the stem to remain at the steady state position. In this case, the compensator is inserted between controller and process, as shown in Figure 2.3.

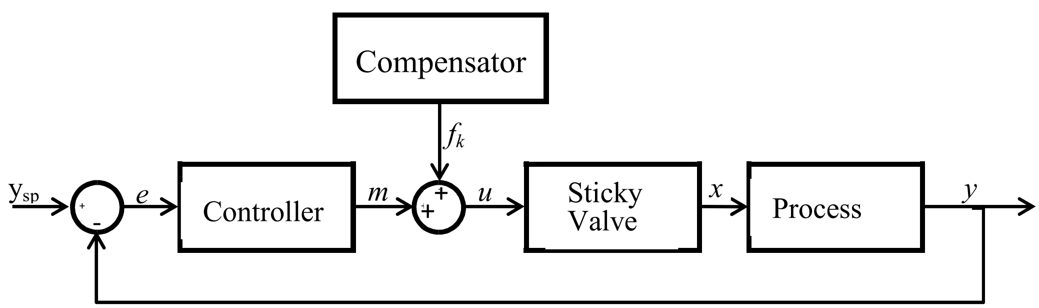


Figure 2.3: Control loop with a two move method compensator.

Where, m is the controller output, f_k is the compensator action, y_{sp} is process setpoint, y is process output, e is the error, u is the additive signal ($m + f_k$) that is being fed to the sticky control valve and x represents the stem position.

The two compensative moves (f_k and f_{k+1}) for stiction compensation are:

$$f_k(t) = u_C(t) + sign(\frac{du_C(t)}{dt})f_k(t) \quad (2.3)$$

$$f_k = |u_C(t)| + \alpha d \quad (2.4)$$

Where d is stick band and α is a real number greater than 1

$$f_k(t+1) = -u_C(t+1) \quad (2.5)$$

It can be seen that from Equation 2.5, the design of the second move is not dependent on the first move.

The two move method (Srinivasan and Rengaswamy, 2008) was an improvement of knocker method (Hägglund, 2002) which can reduce the aggressiveness of control valve. Instead of continuous stem movements, the approach tries to bring the valve stem to steady state in predefined moves. But this method requires the exact stiction *(d)* quantification, the plant should be stable and the process should not be affected by disturbances or white-noise. So this method cannot be feasible in case of practical situations.

2.3 Constant Reinforcement Method

Ivan and Lakshminarayanan (2009) introduced a new compensation method called constant reinforcement (CR) approach. The CR method is similar to knocker method (Hägglund, 2002) but the added signal is a constant quantity instead of pulse.

The compensating signal is a constant reinforcement, added to the controller output signal in the direction of the rate of change of control signal. The valve input signal, $m(t)$ is defined as follows:

$$m(t)=u_c(t)+\alpha(t)=u_c(t)+a\times sign(\Delta u_c) \quad (2.6)$$

Where, α(t) is the compensator signal. If the controller output is constant, the value of α(t) is zero. The recommendation for the constant reinforcement, a is to use the estimated amount of the stiction d in a sticky control valve.

The configuration of CR approach is as follows:

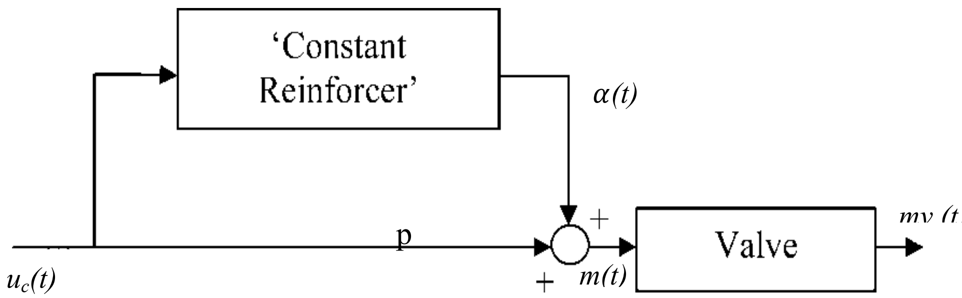


Figure 2.4: Block diagram for CR approach.

The constant reinforcement (CR) (Ivan and Lakshminarayanan, 2009) method is a noteworthy modification of the knocker method (Hägglund, 2002). But it cannot minimize causes of excessive valve movement. Excessive valve movement reduces the life time of control valve.

2.4 Modified Two Moves Method

Farenzena and Trierweiler (2010) proposed a novel methodology to compensate the stiction effects in control valve. They extended and modified the two moves method of Srinivasan and Rengaswamy (2008). In this modified two move method, traditional PI controller is modified instead of adding a compensator block. The method aims to adapt the traditional PI controller for scenarios where stiction is present.

The values for *du* and *dt* can be computed based on the desired closed loop performance (e.g. rise time (r_t)). Assuming a first order plant, these parameters are computed using the following relations:

$$du = \frac{0.95dy}{K(1-e^{-\frac{r_t}{\tau}})} \tag{2.7}$$

$$dt=r_t \tag{2.8}$$

Where r_t is the desired closed loop rise-time, K and τ the process gain and time constant respectively and dy the set point change. The user should tune also the window size Δt, which provides the distance between each pair of moves. Based on Δt, the user can adjust the valve demand - decreasing values imply infrequent valve actions. Depending on the stiction magnitude, or the desired closed-loop rise-time, the first movement *(du)* can be smaller than the minimum movement necessary to overcome the stiction. If there is a model mismatch or the process is constantly affected by disturbances, a modification of the previous relations should be posed. In this method, a small offset between set point and process variable is accepted to avoid constant valve movement.

2.5 Improved Two Move Method

Cuadros et al. (2012) suggested improved versions of the two move compensation method proposed by Srinivasan and Rengaswamy (2008) in order to overcome the drawback related to the set point tracking. In this improved method, the compensating signal is not added to the output of the *PID* controller signal. The compensating signal is directly used as a input signal to the control valve. The proposal consists basically in ensuring that the valve moves smoothly until the error *(SP–PV)* is around zero. The compensating signal $u_i(t)$ of the improved two move method is given by the Equation 2.9.

$$u_i(t) = \begin{cases} u_c(t_1) + \alpha S\left(1 - \frac{t-t_1}{kT_p}\right) sign\left(\frac{du_{cf}}{dt}\right); \;\; t_1 \le t < t_2 \\ u_c(t_1) + \frac{\alpha S}{2} sign\left(\frac{du_{cf}}{dt}\right); \;\; t \ge t_2 \end{cases} \tag{2.9}$$

where u_c is the controller output, u_{cf} is the filtered controller output, T_p is the period of oscillation, α is a real number greater than one, S is the stickband plus deadband.

In this improved two move method, compensator works satisfactorily to reduce process oscillation but cannot reduce the valve stem movement at the desired level. This method also required the actual stiction parameters in the control valve. The determination of actual stiction parameters are practically unfavorable. So a better compensation method is necessary to solve the stiction problem in control valve.

2.6 Three Move Method

Karthiga and Kalaivani (2012) proposed a stiction compensation method of control valve that involved three compensation movements. This approach exhibits a lower overshoot and settling time than two moves method of Srinivasan and Rengaswamy (2008). It imposes a smoother valve operation, which results in a longer valve life. The proposed method aims to obtain the desired closed loop performance with the reduced output variability. The wave shape of the proposed method is shown in Figure 2.5 which involves three movements.

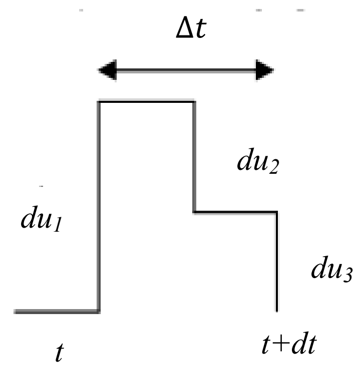


Figure. 2.5: Wave-shape of Karthiga and Kalaivani (2012) compensation method.

The values for each movement du_1, du_2, du_3 can be computed based on the desired closed loop performance. Assume a first order plant, these parameters can be computed using the following relations.

$$du_1=18(d) \tag{2.10}$$

$$du_2=18(u_{max}) \tag{2.11}$$

$$du_3=-(du_1-0.2) \tag{2.12}$$

Where, the d is the stiction parameter of one parameter stiction model (He, Pottmann and Qin, 2007) and u_{max} is the maximum signal from controller output. The sampling time is taken as 0.01.

Depending on the stiction magnitude, the proposed method aims to obtain the values of the first, second and third movements. By using the above relations, the compensator is designed which reduces the valve movements when compared to other compensating methods explained earlier. The compensator needs stiction parameters to estimate the first movement. It is not favorable in case of practical situation. Incorrect estimation of stiction parameter may hamper the compensation of stiction in control valve.

2.7 Improved Knocker and CR Method

Souza et al. (2012) proposed an improvement of the knocker method (Hägglund, 2002) and CR method (Ivan and Lakshminarayanan, 2009) of stiction compensation. The essence of the method lies in the fact that when the Knocker (Hägglund, 2002) or CR (Ivan and Lakshminarayanan, 2009) compensator is applied, after some time the absolute error is minimum and if there are no disturbances or set point changes, the compensating pulses are no longer needed. When these conditions are not met, the compensating pulses should be resumed. The minimum absolute error is sought and the derivative of the filtered error are used to detect it. If the maximum derivative of the filtered error during a given time interval is less than of a threshold, the minimum absolute error was found and the compensating pulses were stopped. A single pulse of considerable amplitude can make the valve move too far and can increase the error. The proposed flow diagram of Souza et al. (2012) stiction compensation method is shown below:

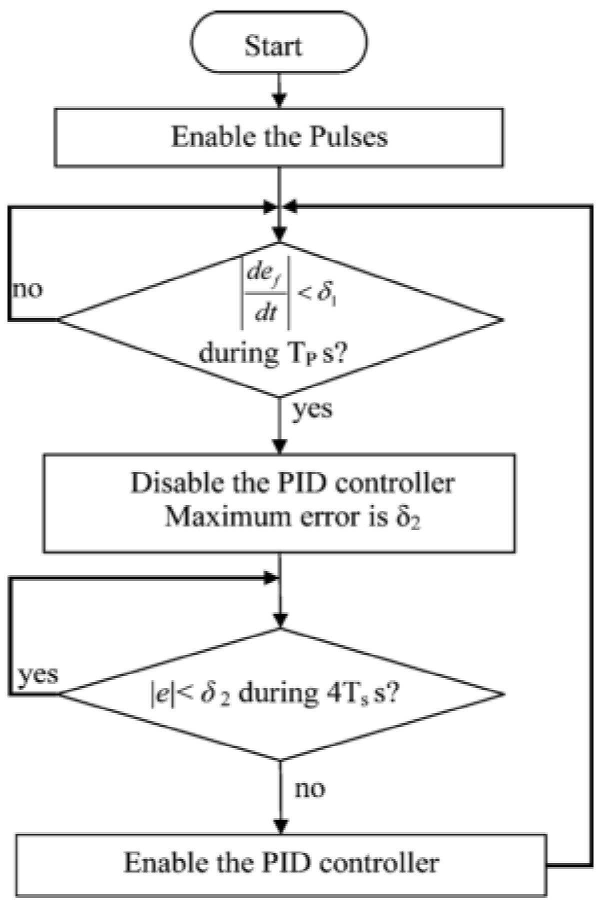


Figure 2.6: Decision making flow diagram of the improved knocker and CR method.

The compensating signal for this method is defined as follows:

$$u_k(t) = \begin{cases} a\ sign\left(u_C(t) - u_C(t_p)\right), t \le t_p + h_k + \tau \\ 0 \qquad\qquad , \ t > t_p + h_k + \tau \end{cases} \tag{2.13}$$

The error must be greater than or equal to δ_2 during the time interval *4Ts* to reactivate the *PID*. This short interval increases robustness to noise and avoid the addition of unnecessary pulse.

The implementation of the proposed compensator is shown in Figure 2.7. It is mainly comprised of Equation 2.13 and the algorithm shown in Figure 2.6.

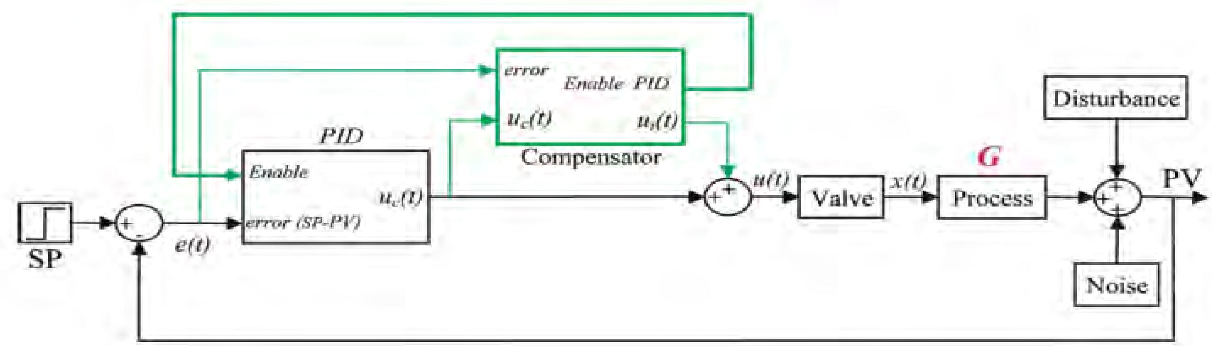


Figure 2.7: Block diagram illustrating the improved knocker and CR compensation method.

The compensator receives the error and the controller signal to produce the pulses and to enable the *PID* controller. The action of enabling and disabling the *PID* controller was performed, changing its mode to manual. This action is replaced by enabling and disabling the *PID* output deadband. When the process variable crosses the *SP* (error crosses zero and changes sign) and as long as *SP−PV* remains in the deadband, the controller output does not change. This strategy is used to reduce actuator wear resulting from controller signals in response to noise only. However, the proposed scheme is claimed to be more robust to outliers because the error signal is to be greater than a threshold value, δ_2 during four sample intervals for the *PID* controller to resume operation. The compensator reduces the output process oscillation as well as valve stem movement but takes longer time to track the set point or reject disturbance. It also requires prior process knowledge which is unfavorable to determine in case of practical situation.

2.8 Improved Knocker Method

Sivagamasundari and Sivakumar (2013) proposed a stiction compensation method based compensation technique similar to the method proposed by Hägglund (2002). But the difference is that the selection of amplitude and duration of the pulse. The waveform of the proposed method is given in Figure 2.8.

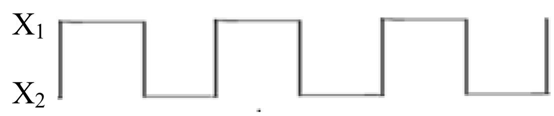


Figure 2.8: Waveform of the improved knocker method.

Instead of selecting the pulse of equal magnitude, here X_1 and X_2 are selected according to the stiction parameter. The pulse characteristics in Figure 2.8 are defined in the following equation.

$$X_1 = 3\times \text{ Stiction in \% of controller span} \tag{2.14}$$

$$X_2 = -\,(125\% \text{ of } X1) \tag{2.15}$$

$$\text{Pulse width} = \text{Sampling time} \tag{2.16}$$

Achieving a non-oscillatory output without forcing the valve stem to move faster and wider than normal is the most important characteristic of this algorithm. This method does not need extensive prior information about the process and the controller, and can track set point changes during operation.

This method reduce the process oscillation at a cost of increasing valve stem movement which is not acceptable in industry.

2.9 Model Free Stiction Compensation Method

Arifin et al. (2014) proposed a model free stiction compensation technique where the compensating signal is a function of the error. The error means the difference between set point and process output. A small error produces little or no valve movement. The proposed scheme is shown in Figure 2.9

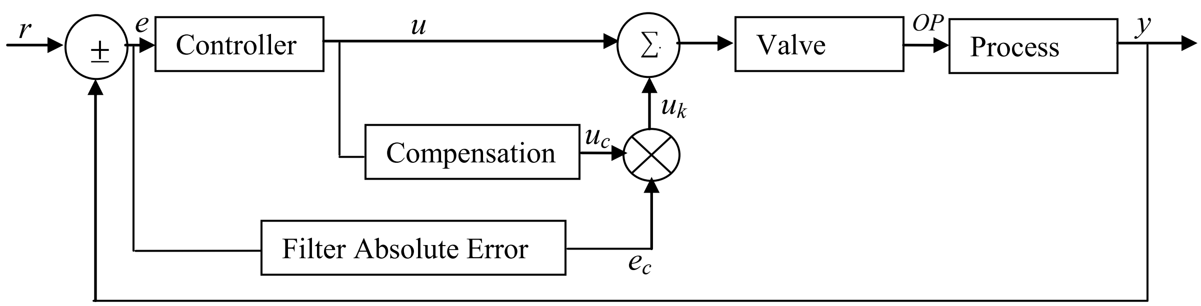


Figure 2.9: Stiction compensation strategies of Arifin et al. (2014) method.

Here, the signal $u_c(t)$ is calculated using the knocker (Hägglund, 2002) or CR (Ivan and Lakshminarayanan, 2009) method. The amplitude of the pulses is $a = \frac{S}{2}$ in order to overcome stiction. Where S is a deadband plus sticband. The signal $e_c(t)$ is the filtered absolute error multiplied by a constant γ, which is between 0 and 1. The compensating signal is the product of e_c and u_c.

$$u_k(t) = e_c(t)u_c(t) \tag{2.17}$$

The procedures to find $e_c(t)$ and $u_c(t)$ are shown in Figure 2.10 and 2.11.

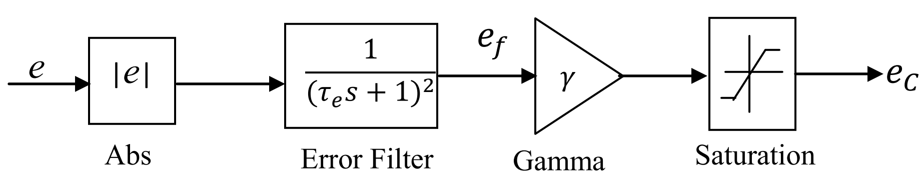


Figure 2.10: Signal flow path for computation of the error signal.

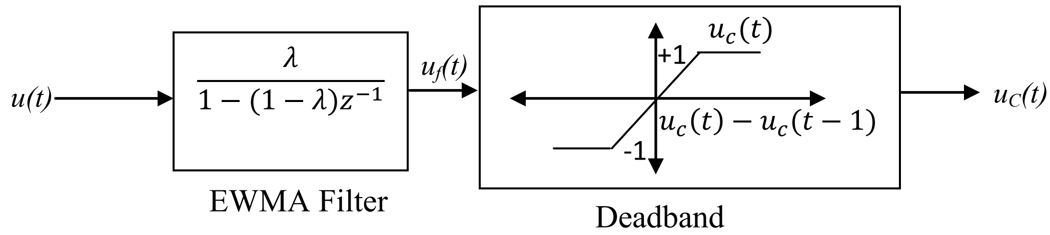


Figure 2.11: Signal flow path for computation of the control signal u_c.

The steps required for the design of the proposed stiction compensation scheme are as follows:

(a) Collect u and $e(= r - y)$ during some periods of oscillation and obtain A_{OP}, A_E and w_o. Where, A_{OP} is the amplitude of control signal. A_E and w_o are the amplitude and frequency of oscillation of error signal.

(b) Calculate $\gamma \geq A_{OP}/ A_E$.

(c) Calculate time constant for error filter, $\tau_e \geq 1/w_o$ [unit = sec /rad].

(d) Select λ in the EWMA filter to reduce noise. Values around 0.5 are a good choice in general. It depends upon how noisy the controller output signal u is.

(e) Calculate deadband for u_c. $\delta_u \leq 0.1A_{OP}$ is reasonable choice .

(f) Calculate the compensating signal u_k using Equation 2.17

(g) Calculate the valve input signal *OP* by adding compensating signal, u_k and controller signal, u.

As discussed, this method requires many parameters to be specified. They are different for different process. So it is cumbersome to implement this method in different industrial plants.

2.10 Method for Compensating Stiction Nonlinearity

Arumugam et al. (2014) proposed a new method for compensating stiction nonlinearity in control valve. This method is similar to the knocker method proposed by Hägglund (2002). The knocker method (Hägglund, 2002) used the square wave which contains harmonics and cause sudden changes in the manipulated variable which may affect the control valve subsequently. In this method, the sinusoidal signals were considered for stiction compensating purpose instead of pulse signals.

The proposed valve stiction compensating scheme using sinusoidal signal is shown in Figure 2.12.

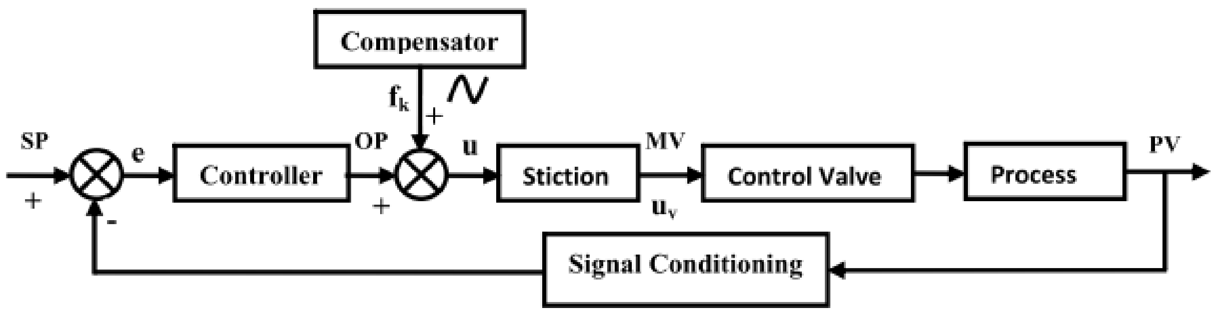


Figure 2.12: Closed loop control system with stiction compensator using sinusoidal signal.

In this Figure, the compensating signal f_k is a sinusoidal signal which is characterized by the following equation

$$f_k = A \sin \omega t \tag{2.18}$$

Where; A=Amplitude $=2\times a$; and $\omega=2\pi f$; $f = \frac{1}{T}$

The amplitude, A and frequency, ω of *sine* wave are calculated from oscillatory response of process output before compensation. The parameter '*a*' *and* '*T*' are the amplitude and period of oscillation of uncompensated process output respectively.

This proposed method can give better results than knocker method (Hägglund, 2002). It can reduce output process oscillation, track set point and reject disturbance satisfactorily. But this method cannot reduce the aggressiveness of the valve stem movement.

2.11 Six Move Method

Wang et al. (2015) presented a six move method to compensate valve stiction. Three consecutive implementations of the "standard" two move method (Srinivasan and Rengaswamy, 2008) are used. This technique allows to estimate the on-line steady-state value of valve input *(OP_{ss})*. Therefore, no a priori assumption on MV is required. However, this approach could take a very long time in real applications, since two extra open-loop step responses must be awaited to compute the final input OP_{ss}.

This method imposes six open- loop movements to the valve as follows:

$$OP(kT_S) = \begin{cases} OP_{max}, & kT_S < T_o \\ OP_{min}, & T_o \leq kT_S < T_o + T_1 \\ OP_{\max,} & T_o + T_1 \leq kT_S < T_o + T_1 + T_2 \\ OP_{min}, & T_o + T_1 + T_2 \leq kT_S < \cdots T_o + T_1 + T_2 + T_{sw} \\ OP_{sw}, & T_o + T_1 + T_2 + T_{sw} \leq kT_S < \cdots T_o + T_1 + T_2 + T_{sw} + T_h \\ OP_{ss}, & Otherwise \end{cases} \tag{2.19}$$

Where $T_o = t_o + \theta_o$, and T_S is the sampling time. When OP is increasing, close to its peak, the controller is switched into open-loop mode at time t_o, and valve input is set to OP_{max} to make the valve move away from the current sticky position. Then, after time interval θ_o, OP is enforced in the opposite direction to OP_{min}. Afterwards, OP switches once again between these two extreme values for times T_1 and T_2. Note that T_1 corresponds to the time interval between the second-last peak and the valley and T_2 corresponds to the time interval between the valley and the last peak, both measured on the oscillation of OP before the compensation starts. Then, after time interval T_{sw}, OP is switched to:

$$OP_{sw} = OP_{ss} - \beta_{sw}(OP_{max} - OP_{min}) \tag{2.20}$$

Where, T_{sw} does not have to be specific, but only to ensure that PV has changed direction. Likewise, β_{sw} is a coefficient (≥ 1) that enables the valve to overcome the stiction band. Finally, after time interval T_h, OP is held to a value so that PV is expected to approach SP at the steady state. The desired steady- state valve position is estimated, according to:

$$MV_{SS} = \frac{OP_{min}T_1 + OP_{max}T_2}{T_1 + T_2} + f_d \frac{T_1 - T_2}{T_1 + T_2} \tag{2.21}$$

If OP is increased first and decreased afterwards, its steady- state value can be computed, by making use of He et al. (2007) stiction model as following:

$$OP_{ss} = MV_{ss} + f_d \tag{2.22}$$

where f_d is the dynamic friction in the valve. In reverse, if the method is implemented in opposite direction, i.e., OP is decreased first and increased afterwards:

$$OP_{ss} = MV_{ss} - f_d \tag{2.23}$$

The interval T_h should be as small as possible, to avoid that PV deviates much from SP value.

However, being a fully open-loop approach, set point tracking and disturbance rejection are still not ensured. The measurement of stiction parameter required to determine steady state valve position is cumbersome. Incorrect measurement of steady state valve stem position may hamper the compensation results.

2.12 Four Move Method

The Revised stiction compensation technique proposed by Capaci et al, (2016) is based on the approach of Wang et al. (2015) by developing some practical simplifications. Only four open-loop movements are now required:

$$OP(kT_S) = \begin{cases} OP_{max}, & kT_S < T_o \\ OP_{min}, & T_o \le kT_S < T_o + T_{sw} \\ OP_{sw}, & T_o + T_{sw} \le kT_S < T_o + T_{sw} + T_h \\ OP_{ss}, & Otherwise \end{cases} \tag{2.24}$$

Where $T_0 = t_o + \theta_o$, and Ts is the sampling time. The first two moves are same as Wang et al. (Wang et al., 2015). When OP is increasing, close to its peak, the controller is switched into open-loop mode at time t_o, and OP is set to OP_{max}. Then, after time interval θ_o, OP is enforced to OP_{min}. If one chooses to impose symmetrical movements to OP that is $T_1 = T_2$, the steady state valve stem position becomes:

$$MV_{ss} = \frac{(OP_{min} + OP_{max})}{2} \tag{2.25}$$

Equation 2.20 and 2.22 or 2.23 are used to compute OP_{SW} and $OP_{SS.}$

Time interval T_{sw} in Equation 2.24 does not need to be specific. A safe choice is $T_{sw} \approx T_{op}$, where T_{op} is the average half-period of oscillation of OP. In this case too T_h should be as short as possible. Overall, by using the proposed method, two valve movements can be avoided, and a significant time (equal to T_1+T_2) can be saved. The compensation process may hamper due to inaccurate measurement of steady state valve position.

2.13 Variable Amplitude Pulses Method

The latest stiction compensation method is the Variable Amplitude Pulse Method proposed by Arifin et al. (2018). The main idea behind the method is to perform a unidirectional search for the amplitude of pulses that brings the error within specified limits. The proposed algorithm is represented by the flowchart shown in Figure 2.13.

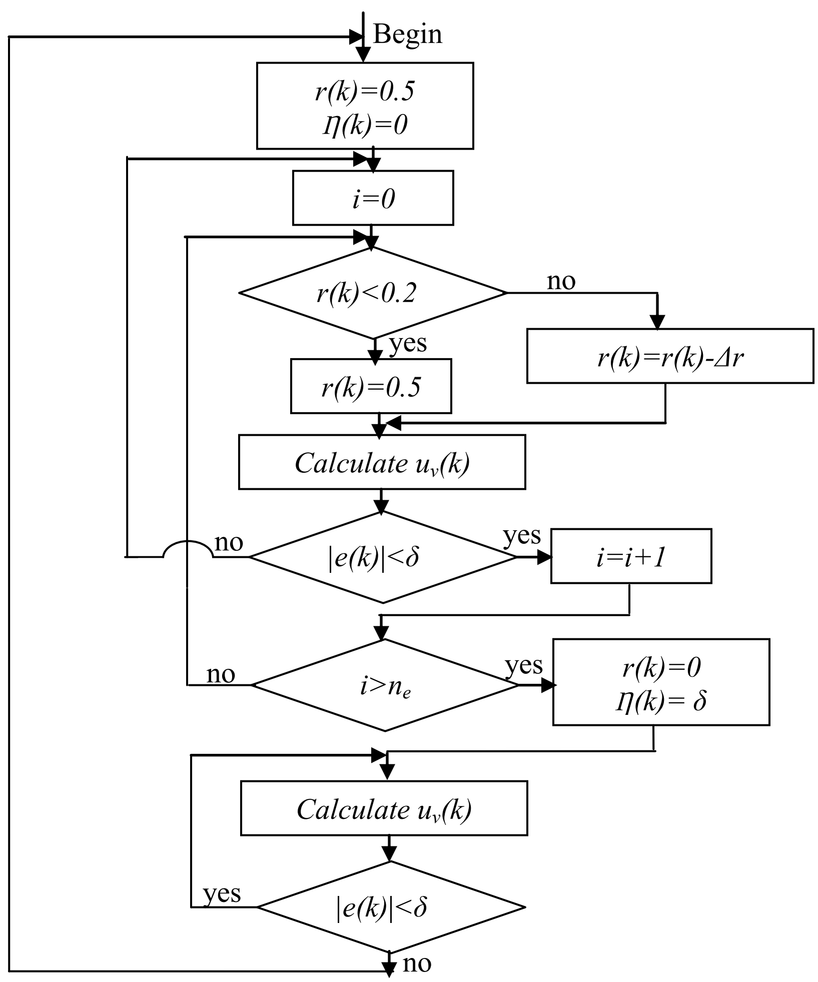


Figure 2.13: Flow chart of actuator stiction compensation via variables amplitude pulse.

When compensation becomes active, compensation pulses are computed using

$$u_c(k)= sign(u(k)- u(k - d))p(k)r(k) \tag{2.26}$$

The addition of a ramp signal, $r(k)$ that starts with a value of 0.5, ends with a value of 0.2, whose slope is given by

$$\Delta r = \frac{0.3}{n_r} \tag{2.27}$$

$$n_r = floor(\frac{4\tau}{T_S}) \tag{2.28}$$

Where τ is the approximate value of time constant of the control loop and T_S is the sampling interval of the loop. Smaller values of n_r reduce the chance for the pulses to be effective, while larger values increase the time for the compensation to work, reducing the performance indexes. A good variation for this choice of n_r is allowed. In the flowchart in Figure 2.13, at every sample time k, the amplitude of the pulses is reduced according to the new value of $r(k)$ and the signal $u_v(k)$ is applied to the valve. The signal $u_v(k)$ is the sum of PI signal $u(k)$ and compensating signal $u_c(k)$ given by Equation 2.26. If the absolute value of the error becomes smaller than the specified limit δ, this event is counted and after n_e number of counts, the pulses are ceased. A suggested value for n_e is given by

$$n_e = floor(\frac{\tau}{T_S}) \tag{2.29}$$

To cease the pulses, the amplitude of the ramp $r(k)$ is set to zero. Also, the dead band for the integral action of the *PI* controller, represented by signal $\eta(k)$ in Figure 2.13, is set to δ. This action is required to prevent the integral action of the controller from bringing the oscillations back, and is common in stiction compensating schemes. If the error becomes greater than the threshold, the pulses are resumed making $r(k)$= 0.5 and setting the deadband of integral action of PI controller to zero, i.e., $\eta(k)$= 0.

This method requires many parameters to be specified for the proper compensation of stiction. They are different for different processes. So, it will be difficult to implement it in different industrial process plants.

2.14 Summary of the Available Stiction Compensation Methods:

Table 2.1 summarizes the main features of the reviewed compensation methods. Four criteria have been used. They are 1) reduction of *PV* oscillation, 2) reduction of valve movement, 3) process knowledge requirement and 4) set point tracking and disturbance

Table 2.1: Table for Various features of available stiction compensation methods.

Sec.	Method	Features			
		Reduction of PV Oscillation	Reduction of Valve Movement	No priori of Process Knowledge Requirement	Set point Tracking and Disturbance Rejection
2.1	Hägglund (2002)	√	×××	√	√
2.2	Srinivasan and Rengaswamy (2008)	√	√	×	××
2.3	Ivan and Lakshminarayanan (2009)	√	××	√	√
2.4	Farenzena and Trierweiler (2010)	√	√	×	×
2.5	Cuadros et al. (2012)	√	×	√	√
2.6	Karthiga and Kalaivani (2012)	√	√	×	√
2.7	Souza et al., (2012)	√	√	×	√
2.8	Sivagamasundari and Sivakumar (2013)	√	××	×	√
2.9	Arifin et al. (2014)	√	×	×	√
2.10	Arungum et al. (2014)	√	××	×	√
2.11	Wang et al. (2015)	√	√	×	√
2.12	Capaci et al. (2016)	√	√	×	√
2.13	Arifin et al. (2018)	√	√	×	√
Symbols: "×" no/low; "××" bad; "×××" very bad; "√" yes/good					

From Table 2.1, it is worth noting that all methods exhibit good capacity in reducing PV oscillation, but most of them also show some drawbacks regarding other issues such as excessive valve movements and process knowledge requirement. So a good stiction compensator should be developed which can reduce PV oscillations without increasing valve stem movement and can work without much process knowledge requirement. A good compensator should satisfy all four criteria. In this study, a novel compensator has been developed which satisfies all four criteria listed above.

2.15 Chapter Summary

This chapter reviews different methods available in the literature for compensating the effect of control valve stiction. It was seen that all methods have some limitations. Finally, all methods with their success and drawbacks have been summarized. It was noted that though there are numbers of stiction parameters, the process industry still needs a good compensator which will be simple yet powerful satisfying all performance criteria.

CHAPTER 3

DEVELOPING A COMPENSATION TECHNIQUE FOR CONTROL VALVE STICTION

Stiction is one of the most adverse nonlinearities that can affect a control valve. It impacts both valve longevity and product quality. The reduction of stiction nonlinearity in control valve is achieved normally at the cost of aggressive valve stem movement. The aggressive valve stem movement may damage the control valve and may lead the process variables to instability. A new stiction compensation scheme has been developed taking into consideration the drawbacks of the various existing compensation methods. The proposed method reduces both oscillation amplitude and frequency, obtains good set point tracking and disturbance rejection.

3.1 Compensation Scheme

The proposed stiction compensator consists of a sequence of pulses with relatively small energy contents. They are added to the control signal for a short period of time when the error, *(SP-PV)* crosses a threshold limit. Thus the compensation is performed by adding short pulses of equal amplitude and duration to the detuned control signal. Once the valve is stuck due to stiction, the valve stem cannot move quickly for a certain step disturbance or set point change. If the valve stem remains at a fixed position, the error keeps increasing. When the error crosses a threshold limit, the compensating pulse signal is activated. Thus the air pressure on the diaphragm increases gradually until the valve slips. The pulse signal is added to the detuned controller signal to start the movement of the valve stem from its stuck position. The pulse should be small enough so that valve does not travel too much. At the same time it should be large enough so that the valve can slip easily from its stuck position. After a small interval of time, the loop is switched back to the standard *PI(D)* controller with the reduced control action. When the valve stem position close to the desired steady state position, the process variable tracks the set-point. The main idea behind the new stiction compensation is illustrated below:

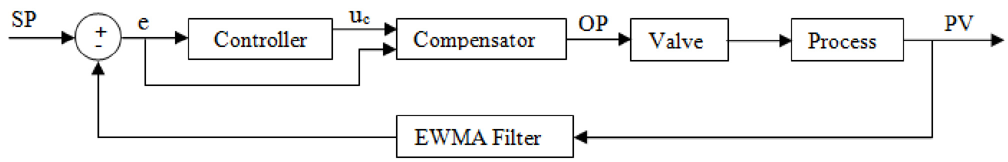


Figure 3.1: Block diagram illustrating the proposed model.

The compensator output, *OP* can be designed as follows:

$$OP(kT_S) = \begin{cases} \dfrac{u_C(kT_S)}{\alpha} - A_P\, sign(error), & if\ T_o < kT_s < T_1 \\ \dfrac{u_C(kT_S)}{\alpha} & ,\ Otherwise \end{cases} \tag{3.1}$$

Where, $T_1 = T_o + T_G$, and T_s is the sampling time. The pulse is started at time T_o when error crosses the threshold, ϵ and continued up to time T_1. T_G is the specified time duration, when the pulse generator, A_P is kept switched on. 'α' is a detune parameter, which is used to reduce the controller action.

To add the pulse with detuned controller signal, the change of the direction of the error signal is taken into consideration. The 'sign' function of the error gives the following

If the error signal is positive, the sign returns '+1'.

If the of error signal is negative, the sign returns '-1'.

If the of error signal is zero, the sign returns '0'.

Therefore, when the sign (error) is '-1', the pulse is added to the detuned controller signal and vice versa.

Suppose A FOPTD process is simulated for 15000 s. A soft stiction block is introduced after 5000 s. The proposed compensator is applied for a process at 10000s. The controller output (upper panel of Figure 3.2) is oscillatory from 5000 s to 10000 s due to stiction. The pulse generator (second last panel of Figure 3.2) is started working at T_o = *10001*s and the pulse is added to the controller signal up to T_1=10100 s. This results in compensator output as shown in lower panel of Figure 3.2. After time T_1, the pulse generator is turned off thus the compensator output is only the detuned controller signal.

The scenario of the process variable output and set-point, controller output, pulse generator output and compensator output is shown in Figure 3.2. The zoomed version of the Figure 3.2 is shown in Figure 3.3.

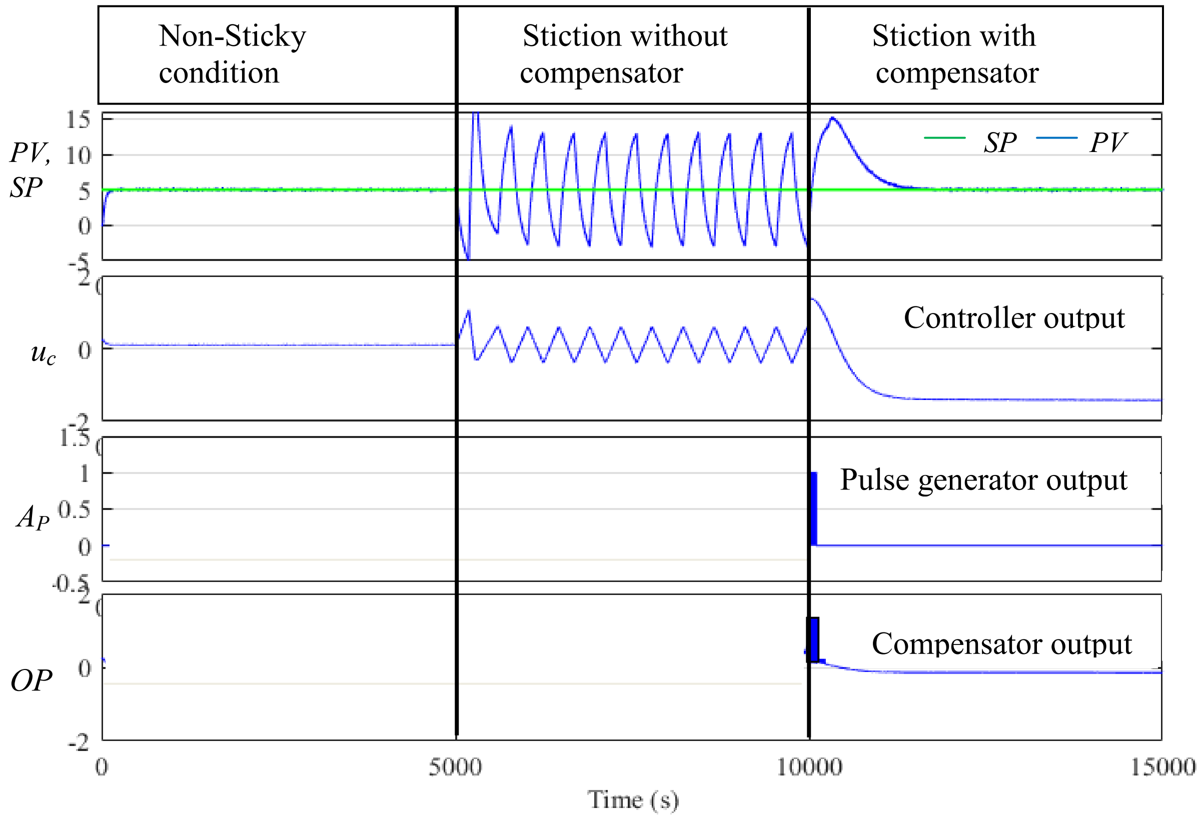


Figure 3.2: Signal from controller, pulse generator and compensator.

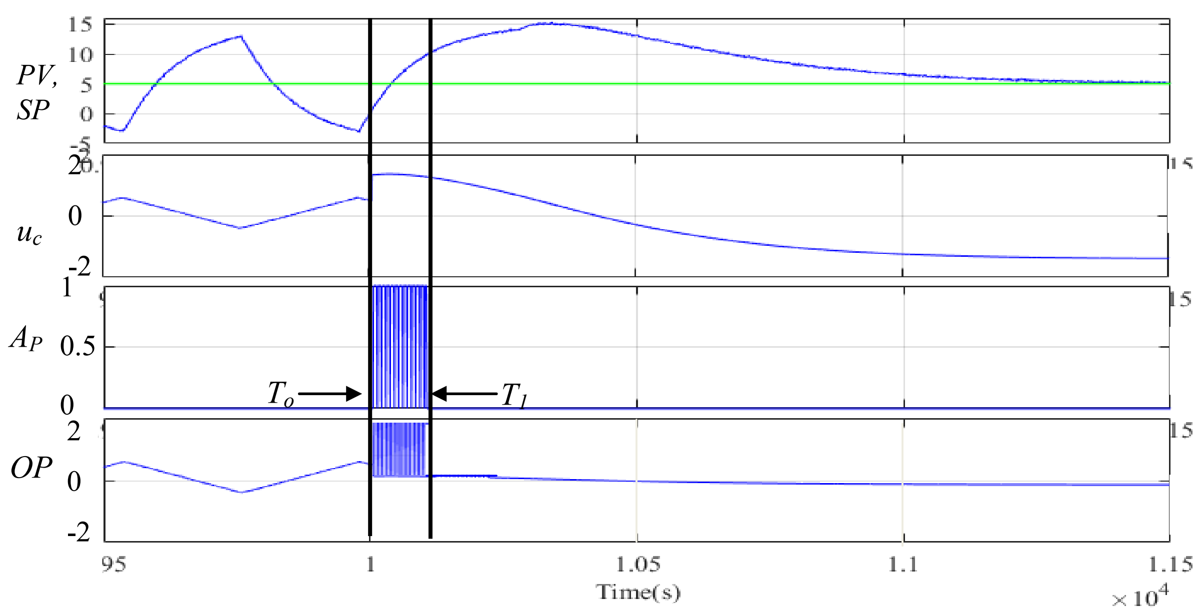


Figure 3.3: Zoomed version of Figure 3.2.

The pulse generator is started from T_o s and continued for T_G s. The time T_G is counted as T_P when the pulse generator is switched on. This value of T_P is kept in memory and add sampling time with the previous value in every loop until $T_P = T_G$. After that the pulse generator is deactivated and T_P is set to zero till the pulse generator active again. The proposed compensator is simulated using the MATLAB Simulink software. The Simulink block diagram of the proposed compensator is shown below:

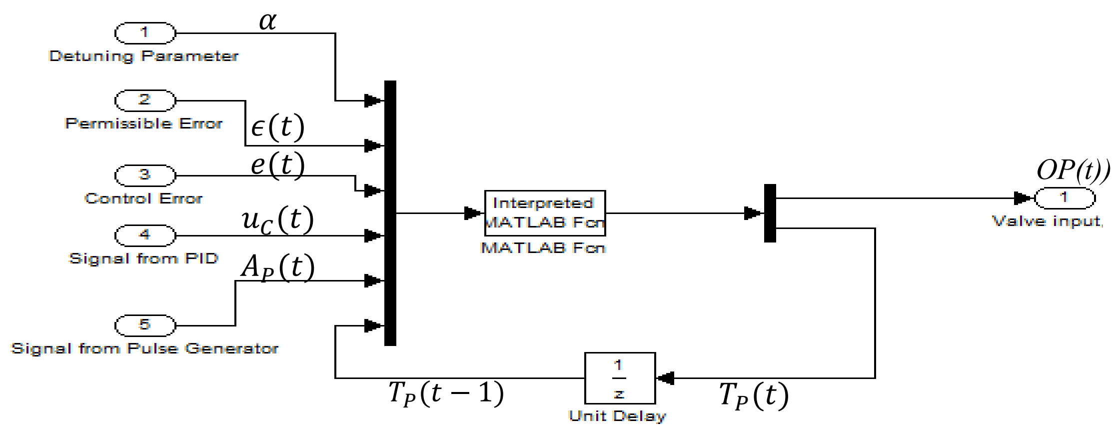


Figure 3.4: Simulink block diagram of the proposed compensator.

The flowchart describing the algorithm in the Matlab function is shown in Figure 3.5.

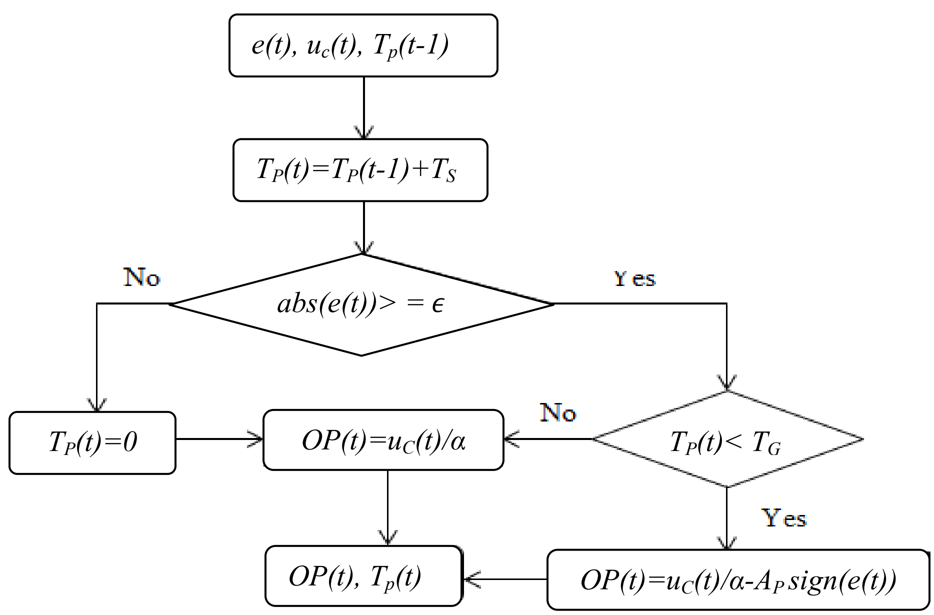


Figure 3.5: Flowchart for the proposed compensator model.

The steps of the proposed algorithm in Figure 3.4 are as follows:

1. First the controller output and the error signal are fed an input to the compensator
2. The controller signal is divided by a detune parameter, α to reduce the controller action.
3. If the error signal greater than a threshold, ϵ then a pulse generator is added to the reduced controller output for a specified period of time, T_G to push the valve from its initial position.
4. Otherwise, the pulse generator is turned off and the reduced controller action is taken as the compensator output, *OP*.
5. The sign of error signal is taken into consideration to add the pulse signal. For the negative value of error signal, pulse is added to the detuned controller signal and vice versa.
6. Finally, the output signal from compensator, *OP* is fed as an input to the control valve.

3.2 Dealing with Stochastic or Noisy control Signals

In order to handle a noisy or stochastic control signal, a time domain filter, e.g. an exponentially weighted moving average (EWMA) can be used before the control error to filter the noisy process variable signals. The filter can be described as follows:

$$G_f(z) = \frac{\lambda Z}{Z-(1-\lambda)} \tag{3.2}$$

The magnitude of λ will depend on the extent of noise used in the simulation. A typical value of λ can be chosen as 0.01.

3.3 Choice of Compensator Parameters

It would be the most convenient if the compensators could be used without any requirement of specifying parameters by the users. Unfortunately, all compensators need some parameters to be specified. The proposed compensator requires a simple pulse generator and a detuning constant, α. This study found that the pulse generator parameters and specified time limit, T_G can be kept same for all processes. Only the detuning parameter, α needs to be specified each time depending on the controller and the process.

3.3.1 Parameters for Pulse Generator

The proposed compensator techniques require a pulse generator to push the valve stem from its initial stuck position. The pulse sequence is characterized by three parameters: pulse period, h_k, pulse amplitude, Ap and the pulse width, τ_d. The pulse sequence characterization is shown in Figure 3.6.

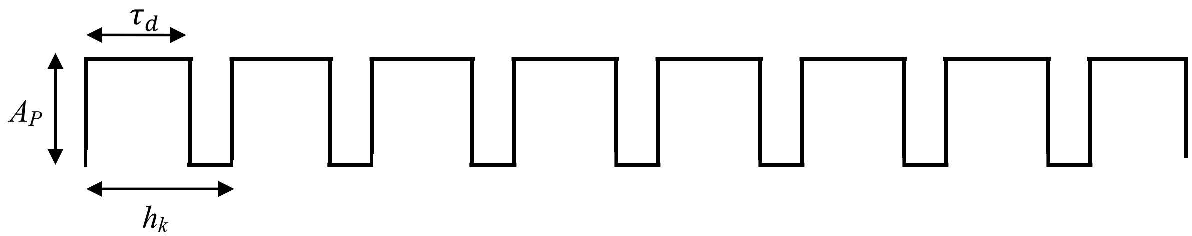


Figure 3.6: Pulse sequence characterization of the proposed method.

To characterize the pulse sequence, three parameters are to be chosen suitably. The pulses should be sufficiently large, so that the valve slips quickly. At the same time they must be small enough so that they do not cause any extra slip. Usually stiction varies from 0 to 20% for most industrial cases. On a normalized scale, it is suitable to choose pulse amplitude in the interval of $1\% < A_P < 2\%$.

It is important not to feed too much energy into the positioner at the moment when the valve slips. Therefore, it is desirable to use a relatively short pulse width. The pulse width can be chosen as 7 to 8 times of the sampling time, T_S.

It is desirable to keep pulse period $1T_S$ to $2T_S$ larger than pulse width so that two successive pulses cannot occur in one sampling time.

3.3.2 Specified Time Limit, T_G

T_G is the specified time limit where pulse generator is kept switched on. Pulse generator is added to the controller action to push the control valve from its stuck position. Pulse generator is not needed when the control valve starts moving from its stuck position. Only the reduced controller action is used to move the valve stem to the desired steady state position. So it is not necessary to add an extra pulse after the valve slips. A Large value of T_G means adding an extra pulse after the valve slips. It causes extra valve movements that is not desired. Usually, 50 s$<T_G<$100 s is suitable for all type of sticky valve (0-20%).

3.3.3 Permissible Error, ϵ

Permissible error is the error that could be acceptable. Generally it is taken as 1.5 times of the average amplitude of oscillation of the process variables for the non-sticky valve (Capaci et al., 2016).

3.3.4 Detuning Parameter, α

In the proposed compensator model, a detuning parameter, α is needed to reduce aggressiveness of the controller. If the controller is aggressive, then the value of α should be high and vice versa. A reliable estimation of this parameter is an important prerequisite for this method. The detune parameter, α is different for every different process. Incorrect estimation of α cannot mitigate the effect of stiction satisfactorily. The detune parameter, α can be selected by evaluating the integral absolute error, *IAE* using Equation 3.3. For a given process, *IAE* is different for different value of α. The value of α which gives minimum *IAE* for a process would be the actual detune parameter, α for that process.

The *IAE* is defined as:

$$IAE = \frac{1}{T_b - T_a} \int_{T_a}^{T_b} |(SP - PV)|\, dx \tag{3.3}$$

Where, *SP* is the desired set point and *PV* is process variable. T_a and T_b are the time interval over which the *IAE* is calculated.

The importance of correct estimation of α is illustrated using an example. Assume a first order process given as in Equation 3.4. The compensator was introduced after 4000s for three different cases of detune parameter, α. The process is defined as follows:

$$G_p(s) = \frac{50e^{-10s}}{100S+1} \tag{3.4}$$

The controller parameter for this process based on *IMC* are: $K_C = 0.046$, $\tau_I = 100$.

The controller parameters were kept same for all three cases. It can be seen from Figure 3.7 (a) that the oscillation of the process variable is increased after installing compensator at 4000 s for small detune parameter, α (α=5) and gets a large *IAE* value of 4.13. A large value of α (α=30) can reduce the oscillation but takes much time to track the set-point point as shown in Figure 3.7(c).

Figure 3.7(b) shows that compensator can mitigate the oscillation for the process successfully and gets a minimum *IAE* for a detune parameter, α=10. So, α should be selected as 10 for this process. Therefore, the values of α should be determined correctly. It is to be noted that values of α is always greater than 1.

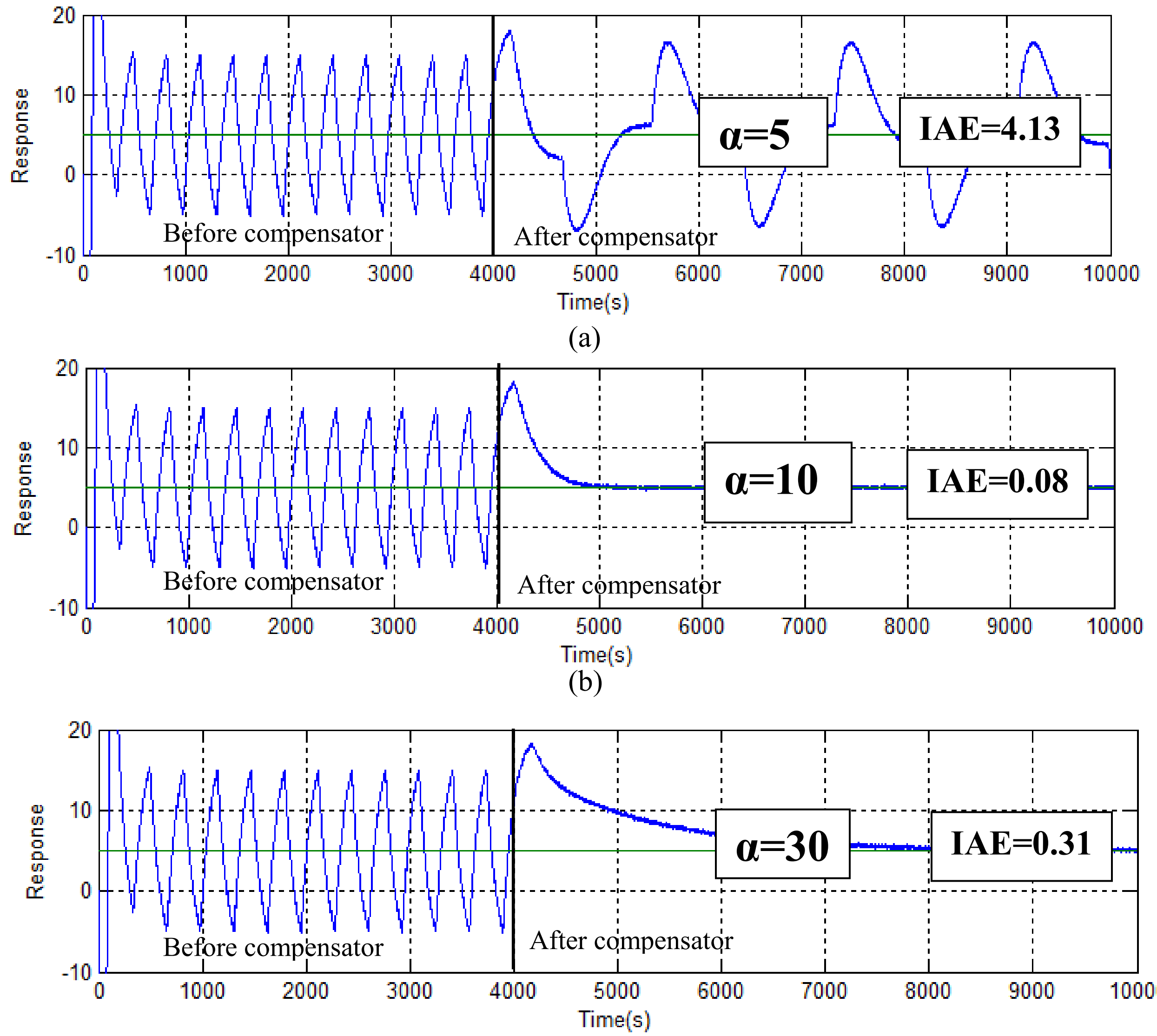


Figure 3.7: Process output response curve of a single first order process for three different cases of α.

3.3.4.1 Estimation of Detune Parameter, α

As shown in the above example, the detune parameter, α should be chosen properly for the successful implementation of the compensator. The relationship between controller parameter and detune parameter should be found to implement the compensator. It is difficult to find such relationship theoretically. In order to find an empirical relationship between α and the controller parameters, the proposed compensator is applied for different First Order Plus Time Delay (FOPTD) processes. The general model for a FOPTD process is described as:

$$G_P(s) = \frac{K_P e^{-\theta s}}{\tau s+1} \tag{3.5}$$

Where, K_P is process gain, τ is time constant and θ is time delay. The process gain (K_P), time constant (τ) and time delay (θ) were varied in the range of [1:5:51], [1:10:201] and [0:3:21] respectively. These combinations produced 1848 different FOPTD processes. Valve stiction was introduced using the widely used Choudhury et al. (2005) stiction model setting the parameters: $S=5$ and $J=3$. A white noise with zero mean and standard deviation σ =0.01 was added to the process output. Before starting the compensator, the process output was oscillatory because of introducing stiction to the valve. But some combinations may not be able to generate limit cycle because they do not fulfill the criteria of producing limit cycles. The detail of such criteria is described in Choudhury et al. (2005).

Out of 1848 different FOPTD processes, 1431 FOPTD processes could generate limit cycle oscillations in presence of stiction. This is not problem because they are being used to find a relationship of finding α. The relationship will be validated to see its reliability.

The proposed stiction compensator was applied to mitigate the stiction effect of these 1431 FOPTD process. For all cases, the following parameters were used: T_s= 1, T_G= 100 and ϵ = 0.5. Every process model was run up to 10000 s and the compensator started at 4000 s for each case. The detuning parameter α was varied in the range of [1:0.5:200] for each process and the corresponding integral absolute error *(IAE)* for each FOPTD process was calculated using the last 3000s data. The *IAE* values were different for different detune parameters for each FOPTD process. The optimum detune parameter was selected for which the *IAE* was minimum. So, 1431 FOPTD model have 1431 optimum detune parameter where the compensator worked satisfactorily and process variable remained steady with minimum oscillation. It is to be noted that the minimum *IAE* was found to be less than 0.10 for all cases whereas the *IAE* was greater than 4 before the compensator was used. The Figure 3.8 shows that the proposed compensator reduces the process oscillation in presence of stiction and gives small value of *IAE* for optimum detune parameter of each 1431 process.

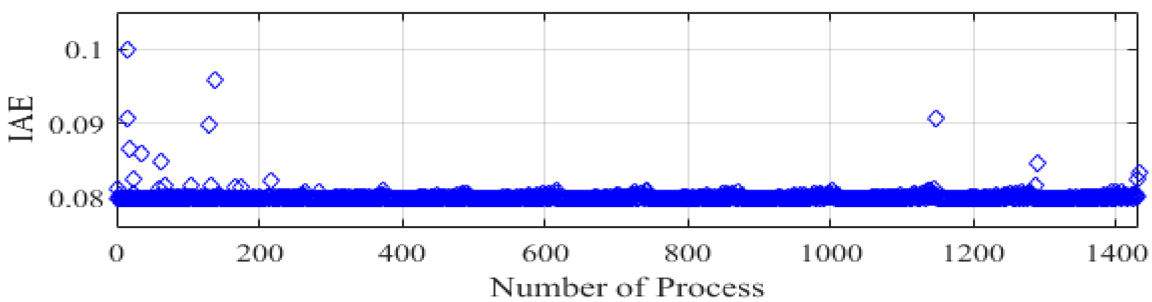


Figure 3.8: Minimum IAE of different process for selected detune parameter, α.

Now data for 1431 process models were fitted using 14 different empirical model by trial and error method using statistical regression analysis. Regression analysis is a set of statistical processes for estimating the relationships among variables. It helps to understand how the typical value of the dependent variable changes when any one of the independent variables is varied, while the other independent variables are held fixed. For the 1431 *FOPTD* processes, the process variables K_P, τ, θ and *PI* controller parameters(K_C, τ_I) using *IMC* technique are known.

The following empirical models were fitted:

[1] $\alpha = AK_P^x + B\tau^y + C\theta^z + \mathrm{D}$

[2] $\alpha = AK_P^x + Be^{\tau/y} + C\theta^z + \mathrm{D}$

[3] $\alpha = AK_P^x + B(\frac{\theta}{\tau})^y + \mathrm{D}$

[4] $\alpha = AK_C^x + B\tau_I^y + \mathrm{D}$

[5] $\alpha = A(\frac{K_C}{\tau_I})^x + D$

[6] $\alpha = A\, e^{x(K_C/\tau_I)} + \mathrm{D}$

[7] $\alpha = A(K_C^x/\tau_I^y) + \mathrm{D}$

[8] $\alpha = Alog(K_C^x/\tau_I^y) + \mathrm{D}$

[9] $\alpha = Alog(K_C^x \times \tau_I^y) + D$

[10] $\alpha = \frac{AK_P^x}{(\tau\theta)^y} + \mathrm{D}$ $[if\ \theta\ is\ zero\ then\ set\ to\ 0.01]$

[11] $\alpha = AK_C^x + \mathrm{D}$

[12] $\alpha = \frac{A}{K_P^x+\tau^y+\theta^z} + \mathrm{D}$

[13] $\alpha = \frac{A}{K_C^x+\tau_I^y} + \mathrm{D}$

[14] $\alpha = AK_P^x + Be^{-\tau/y} + C\theta^z + \mathrm{D}$

The values of power indices x, y and z in these above combinations were varied each in the range of [0: 0.1: 5].

For example, for the first combination, it can be written as

$$Y = X\beta + \in \tag{3.6}$$

Where,

$$Y = \begin{bmatrix} \alpha_1 \\ \alpha_2 \\ - \\ - \\ - \\ \alpha_n \end{bmatrix}_{1431\times1} \quad X = \begin{bmatrix} 1 & K_{P1}^x & \tau_1^y & \theta_1^z \\ 1 & K_{P2}^x & \tau_2^y & \theta_2^z \\ - & - & - & - \\ - & - & - & - \\ - & - & - & - \\ 1 & K_{Pn}^x & \tau_n^y & \theta_n^z \end{bmatrix}_{1431\times4} \quad \beta = \begin{bmatrix} D \\ A \\ B \\ C \end{bmatrix}_{4\times1}$$

The coefficient, β were calculated as follows:

$$\beta = (X^TX)^{-1}(X^TY) \tag{3.7}$$

These variation of power x;y;z produce [51×51×51=132651] number of equations for the first equation. For each empirical equation, $\hat{Y}$ can be found using $\hat{Y} = X\beta$. The Sum of Square Error, *SSE* can be found as

$$SSE = \in^T \in \tag{3.8}$$

Here, $\in$ is the difference between Y and $\hat{Y}$. So, Equation 3.8 becomes

$$SSE = \left(Y - \hat{Y}\right)^T (Y - \hat{Y}) \tag{3.9}$$

From the 132651 different simulation cases, the final equation was selected where the *SSE* was minimum. Similar procedure had been done for all other 13 empirical models. The minimum *SSE* for 14 empirical models are shown in Figure 3.9.

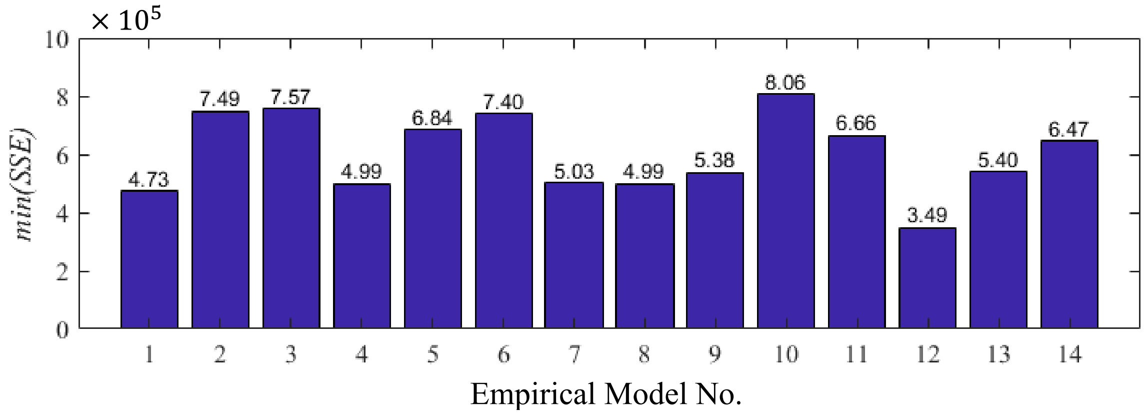


Figure 3.9: Minimum SSE for different number of Empirical Models

Figure 3.9 shows that empirical model no. 12 gives *min(min(SSE)*. So, it can be said that empirical model no.12 is the best fitted model where the deviation between predictive and actual detune parameter was minimum. The parameters corresponding to this model are:

x= 0.3; y=0.7;z =0.9; A=834; D=-12.

Therefore, the empirical relation is

$$\hat{\alpha} = -12 + \frac{834}{K_P^{0.3} + \tau^{0.7} + \theta^{0.9}} \tag{3.10}$$

The controller parameters were found based on *IMC* method

The controller parameters depending on *IMC* are,

$$K_C = \frac{\tau}{K_P(\frac{\tau}{3} + \theta)} \;;\quad \tau_I = \tau \tag{3.11}$$

Replacing the process parameters by controller parameters in Equation 3.10, the final relationship for detune parameter is found to be:

$$\hat{\alpha} = -12 + \frac{834}{\left\{\frac{3\tau_I}{(\tau_I + 3\theta)K_C}\right\}^{0.3} + (\tau_I^{0.7} + \theta^{0.9})} \tag{3.12}$$

3.4 Initialization of Compensator

As it is mentioned earlier that the proposed compensator works by reducing the controller action and adding an extra pulse. The output of the controller is divided by a constant detune parameter. So at the switching time to the starting of the compensator, the input signal to the valve may cause some process upsets. The process output response curve for a typical case is shown in Figure 3.10, where the compensator starts working at 4000s. The output of the process goes to lower value than it was before as shown in the enclosed the circle in Figure 3.10.

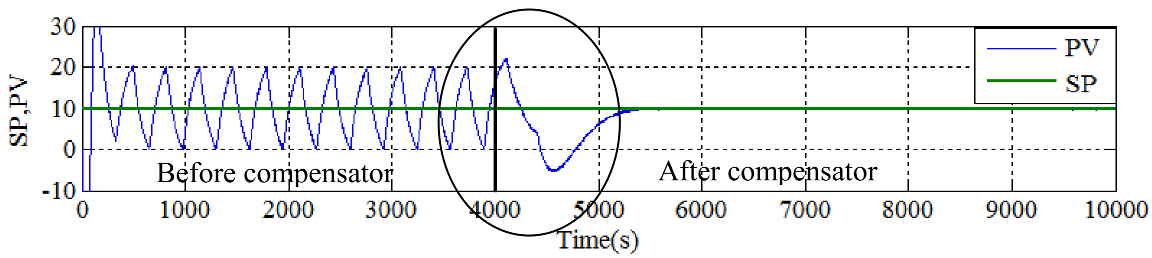


Figure 3.10: Process output response for the proposed model without initialization.

To overcome this problem, the compensator output needs initialization. The initialization can be done using the steady state value of the controller output before starting the compensator. Figure 3.11 shows the performance of compensator after such initialization. The average signal of the controller output was set as initial output of the compensator at 4000s. Then the reduced controller action with pulse was added with the initial compensator signal at 4001s. The compensator could mitigate the stiction effect without creating much upset as shown in Figure 3.11.

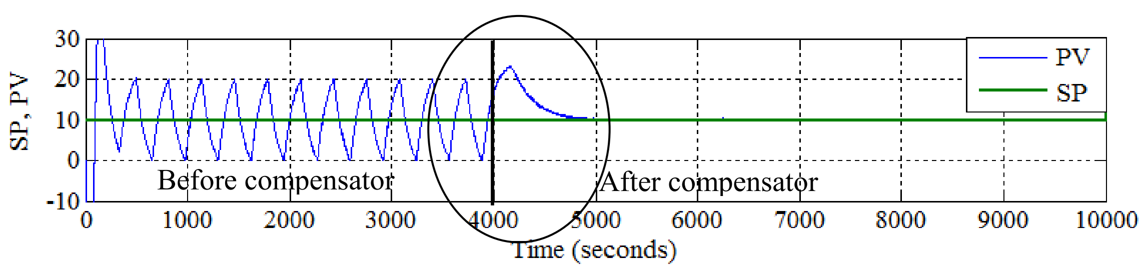


Figure 3.11: Process output response for the proposed model with initialization.

3.5 Chapter Summary

The proposed stiction compensation method has been described in this chapter. The procedure for specifying the required parameters for this method is also discussed. The relationship between detune parameter and process parameters is developed using regression analysis. This chapter has focused on the development of the proposed stiction compensation method and determination of the required parameters.

CHAPTER 4

PERFORMANCE EVALUATION OF THE PROPOSED COMPENSATION METHOD

Stiction is most commonly found problem in control valves. There are some compensation methods available in literature. The performance of the compensator is evaluated using different criteria such as set-point tracking, disturbance rejection, number of valve reversals and sensitivity to noise. This chapter discusses the evaluation of the performance of the proposed compensator.

4.1 Evaluation of Performance of Compensator for FOPTD Process

The performance of the proposed compensator has been evaluated extensively for many number of First-Order Plus Time Delay (FOPTD) processes, which will be discussed in next section. In this section, a particular case of FOPTD process will be presented elaborately. Consider a FOPTD process as follows:

$$P(s) = \frac{50}{100\,s+1} e^{-10\,s} \tag{4.1}$$

The PI controller based on IMC was:

$$C(s) = 0.046(1 + \frac{1}{100\,s}) \tag{4.2}$$

Valve stiction was introduced using the widely used Choudhury et al. (2005) stiction model setting the parameters: $S=5$ and $J=3$. A white noise with zero mean and standard deviation of σ =0.01 was added to the process output.

For the proposed compensators, the following parameters were used: T_s= 1 s, T_G = 100 s, ϵ = 0.5 and α=10.

The process model was simulated using MATLAB Simulink software environment. The process model were run for 10000 s and the compensator started at 4000 s.

The Simulink block diagram is given below:

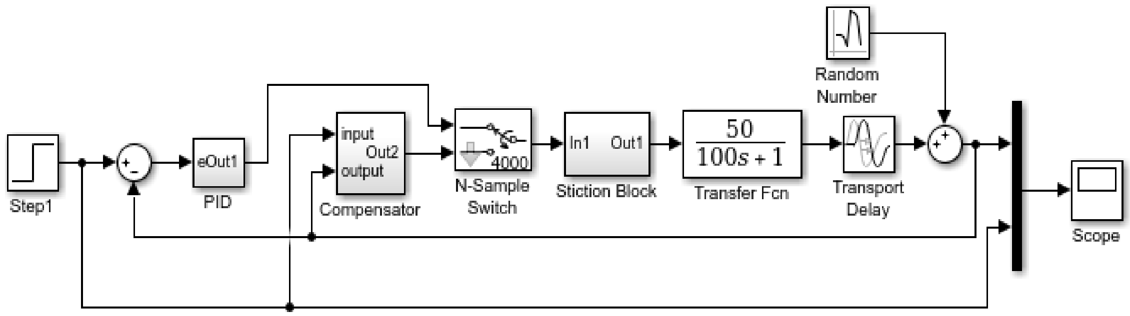


Figure 4.1: Simulink block diagram for evaluating the performance of proposed compensator.

The output process variable, *PV* along with set-point, *SP* (top panel) and valve stem movement, *MV* along with valve input signal, *OP* (bottom panel) are shown in Figure 4.2.

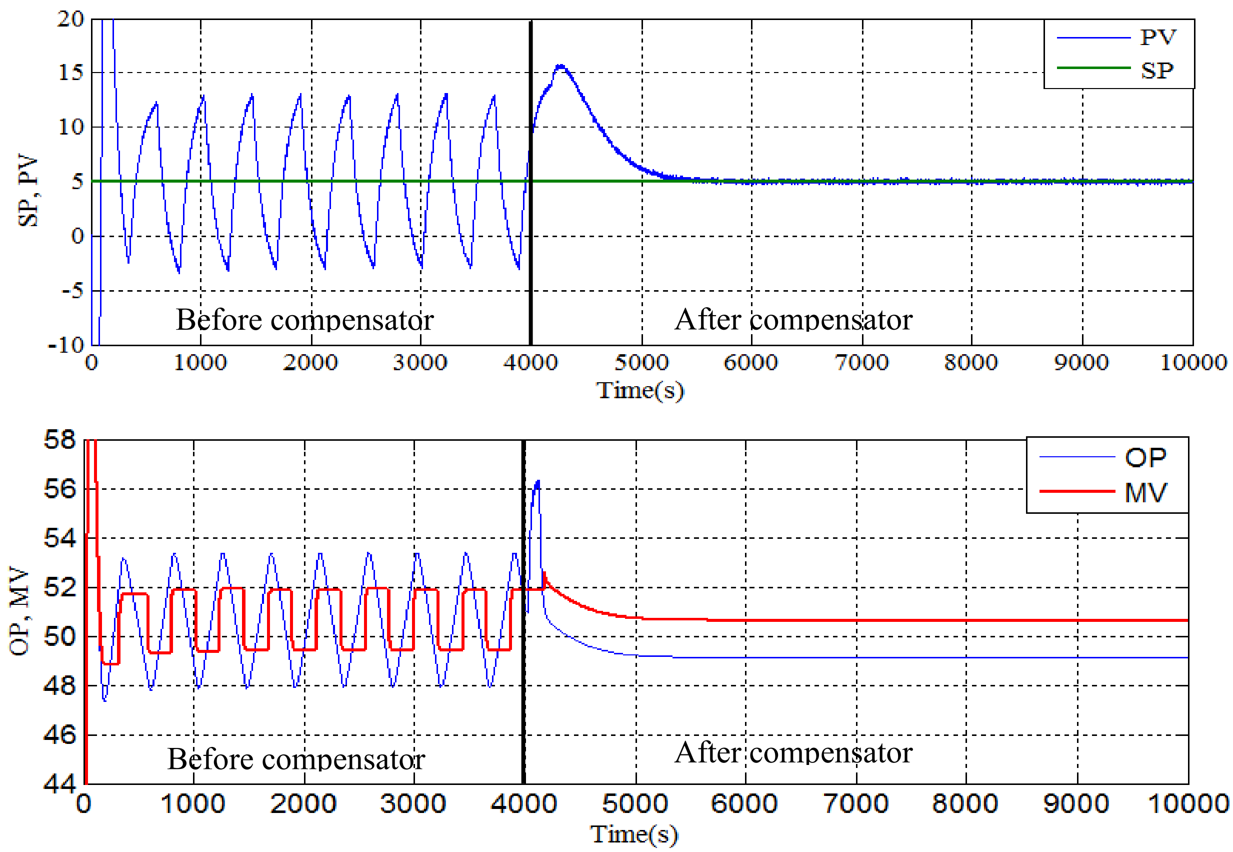


Figure 4.2: Simulation results for the proposed compensator of a single FOPTD process.

Figure 4.2 shows that the process variable, *PV* was oscillatory up to 4000 s due to valve stiction. At 4000 s, the compensator was started and the compensator could eliminate the oscillations completely in about 1200 s time. Therefore, it can be said that the proposed compensator can mitigate the stiction effect satisfactorily by reducing process oscillation as well as valve stem movements.

4.2 Evaluation of the Performance of the Compensator for different FOPTD process

The proposed method was also evaluated for different First Order Plus Time Delay (FOPTD) processes. The general model for FOPTD processes can be written as:

$$G_P(s) = \frac{K_P e^{-\theta s}}{\tau s+1} \tag{4.3}$$

Where, K_P is process gain, τ is time constant and θ is time delay. The process gain (K_P), time constant (τ) and time delay (θ) were varied in the range of [1:5:51], [1:10:201] and [0:3:21], respectively. These combinations produced 1848 different FOPTD processes. The controller parameters were calculated based on *IMC* rules for different processes. The parameters of the proposed compensator were kept same as before. Only the detuning parameter α was different for every model which was dependent on the process parameters. Every process models were run for 10000 s and the compensator started at 4000 s for each case. For the sake of brevity, it is not possible to show all results. Some compensation results are shown in Figure 4.3 for different FOPTD processes.

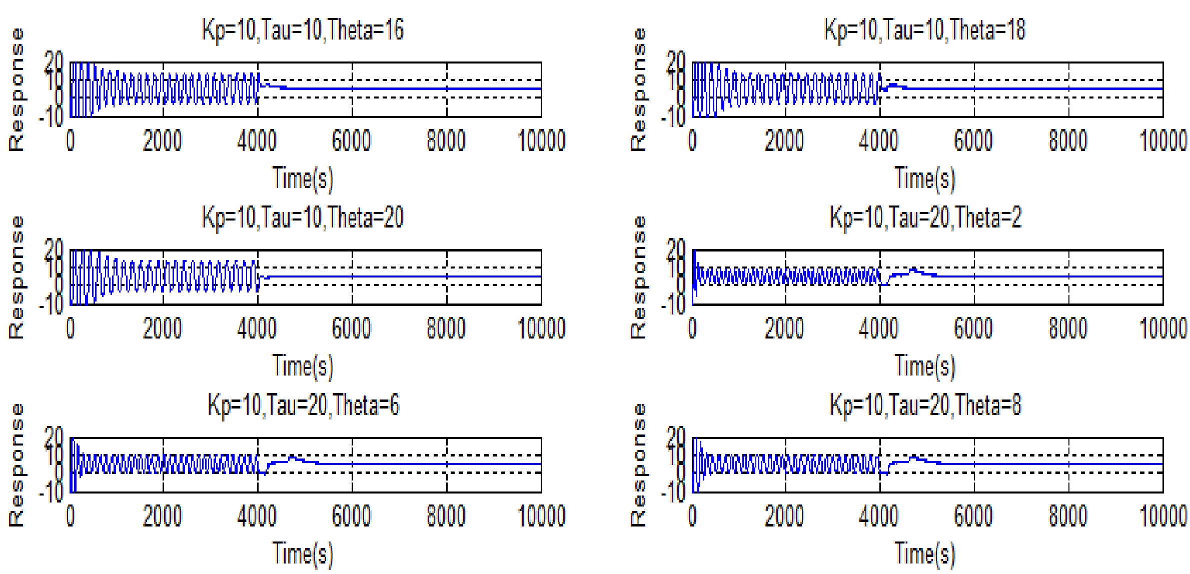


Figure 4.3 (a): Sample results for different FOPTD model.

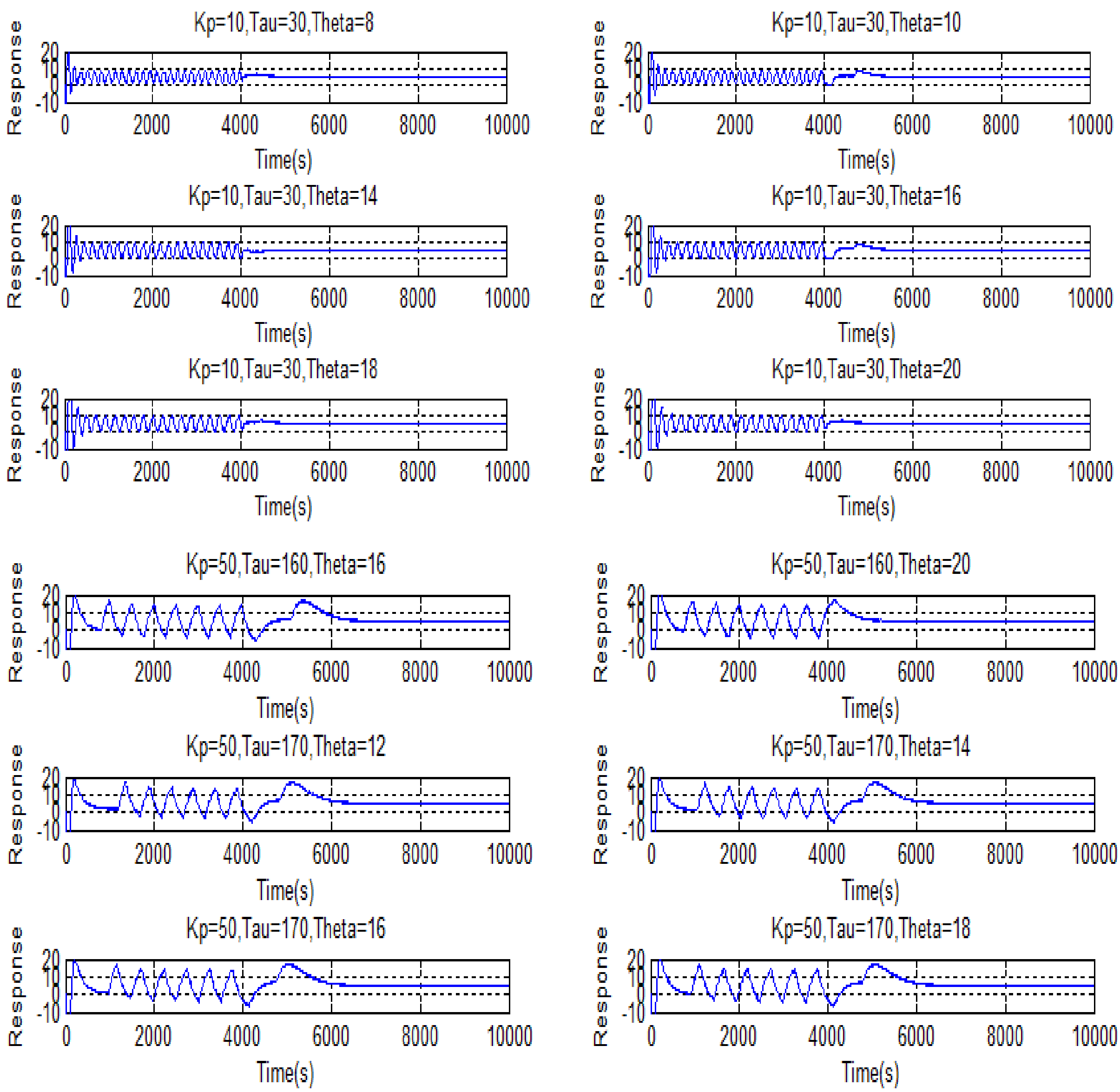


Figure 4.3 (b): Sample results for different FOPTD model.

Figure 4.3 shows that the process output is oscillatory before starting the compensator. After the compensator was switched on at 4000 s, the oscillations in the process output, *PV* were gone completely. Therefore, it can be said that the proposed compensator works satisfactorily for different FOPTD processes.

4.3 Evaluation of the Compensator Performance for Different Perturbations

The proposed compensation scheme was applied for different scenarios on the same process and the controller described by Equation 4.1 and 4.2, respectively. Stiction was introduced by using the widely used Choudhury et al., (2005) stiction model. The stiction parameters (*S=5, J=3*) were used for first three cases. The compensator was evaluated for four cases such as 1) set-point tracking 2) disturbance rejection 3) various noise level and 4) varying amount of stiction.

4.3.1 Set Point Tracking

The proposed compensator model was tested to check its performance during set-point tracking control. The compensator was triggered ON at time instant, t=4000 s. The top panel in Figure 4.4 shows that the oscillations have been eliminated and process variable can track the different set point changes successfully. The bottom panel shows that there is no excessive valve movement and the number of valve reversals is also minimum.

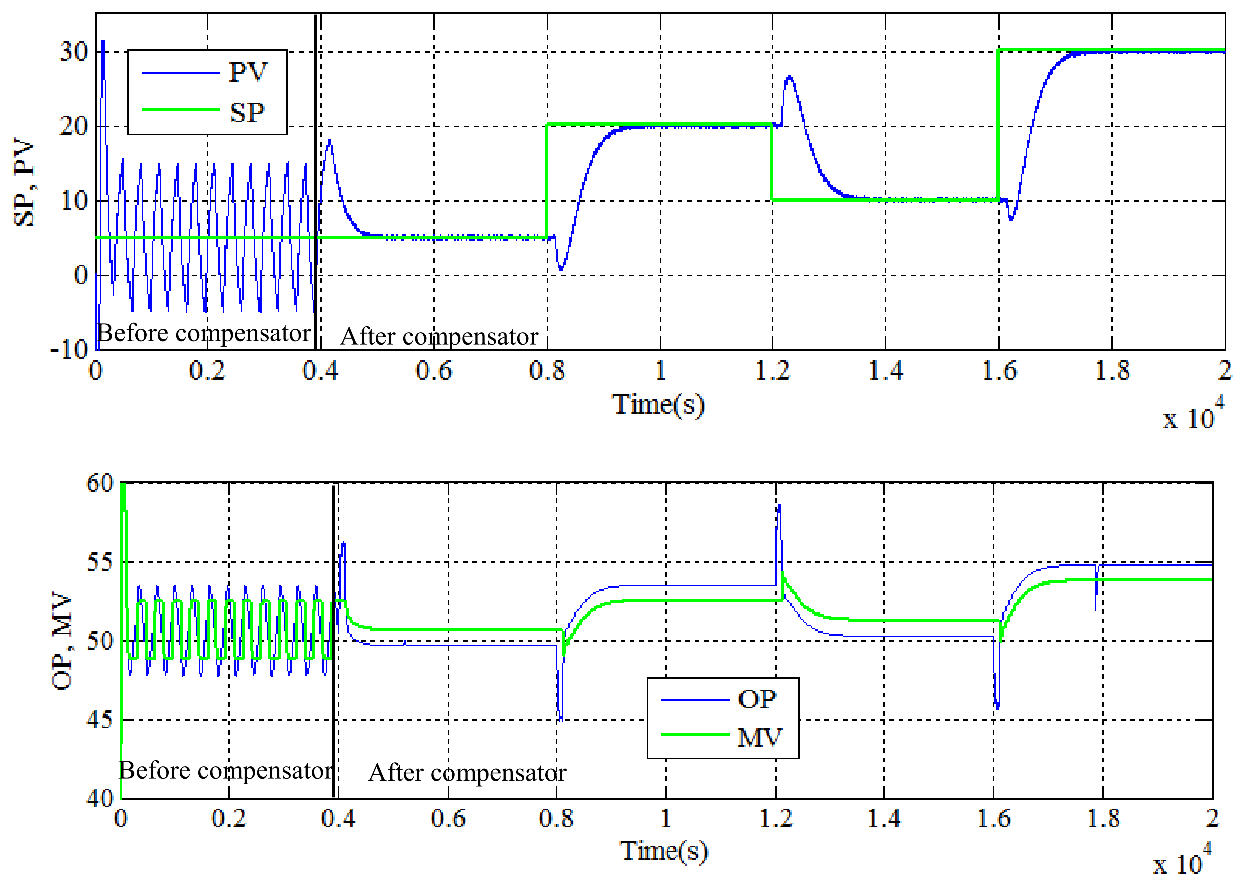


Figure 4.4: Stiction compensation for set-point tracking.

4.3.2 Disturbance Rejection

Any step disturbance tends to deviate the process output from its set point. It is desirable to remove the effect of disturbance within a short time. The compensator was applied at t=4000 s. It could eliminate the oscillation and bring the process at steady state around 4800 s. Then a unit step type disturbance was applied at 7000 s. The proposed compensator could eliminate the step disturbance effectively within a short period of time. The bottom panel of Figure 4.5 shows that the valve stem returns quickly at the desired steady state position. Therefore, it can be said that step type disturbance could be eliminated effectively by the proposed compensator.

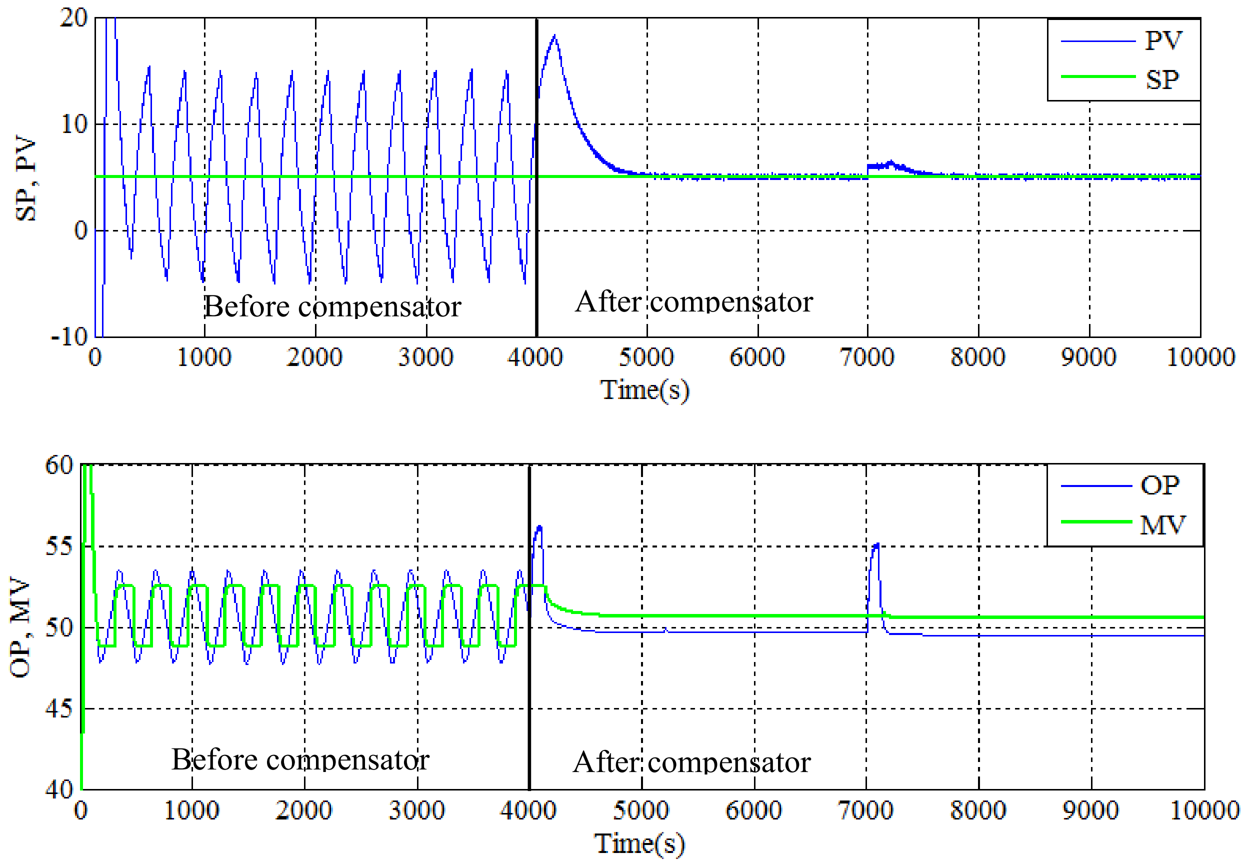


Figure 4.5: Results for proposed compensator when a step disturbance is added at 7000 s.

4.3.3 Sensitivity to Noise

The performance of the proposed compensator was evaluated for different noise levels. The white noise sequence of different variance levels was added to the process output. The integral absolute error *(IAE)* was computed for each case. Table 4.1 summarizes the results.

Table 4.1: Effect of noise on stiction compensation

Variance of noise (σ)	IAE_{before}	IAE_{after}	% $IAE_{reduction}$
0	4.60	0.0002	99.99
0.01	4.60	0.0809	98.24
0.02	4.61	0.1138	97.53
0.03	4.61	0.1396	96.97
0.04	4.61	0.1620	96.49
0.05	4.61	0.1807	96.08
0.10	4.61	0.2572	94.42
0.20	4.62	0.3628	92.15
0.50	5.41	0.5709	89.44

For each case, the model was run for 10000s and the compensator was applied at t=4000 s. The integral absolute error, *IAE* for every case was calculated for 1000 to 4000 s before starting the compensator. After starting the compensator, the *IAE* was also calculated for the last 3000 s when the process reached the steady state. For all noise levels, the compensator could eliminate the oscillation completely. Time trend for the case of $\sigma=0.5$, is shown in Figure 4.6. It clearly shows that the compensator is working satisfactorily in the presence of significant amount of noise. Table 4.1 shows that the compensator performed very well. Even for the last case of high level of noise, it could reduce *IAE* by 89.44%.

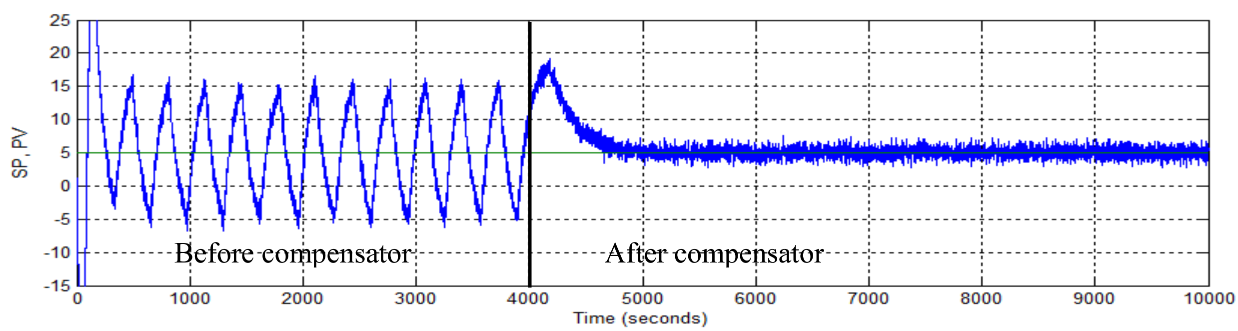
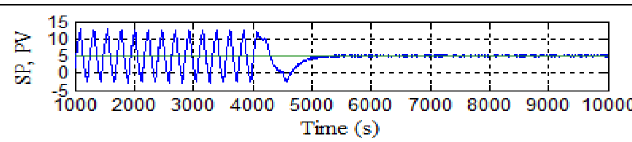
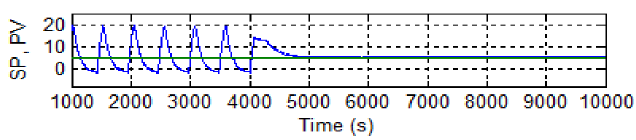
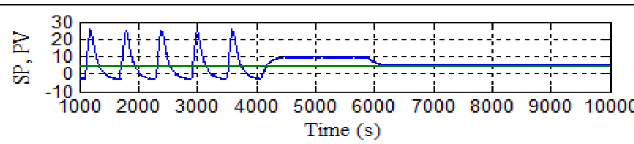
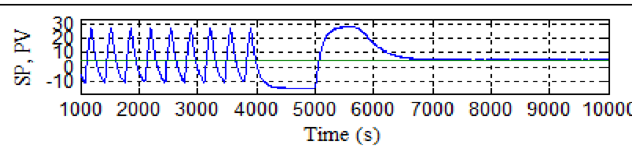


Figure 4.6: The output response of a process when high variance level of noise is added.

4.3.4 Sensitivity to Extent of Stiction

The compensator developed in this study does not require the knowledge of stiction parameters. The measurement of stiction parameters is cumbersome. The sticband plus deadband for upward and downward direction of valve travel could be different. A modified version stiction model appeared in Choudhury et al., (2005) was used to simulate asymmetric stiction. The modified version was capable of specifying different amount of sticband plus deadband in the upward and downward direction of valve travel. The deadband plus sticband parameter, S is denoted as 'SU' for upward and as 'SD' for downward direction. The other parameter slip jump is denoted by, J. Various stiction cases are simulated for stiction parameters listed in the first column of Table 4.2. The performance of the proposed compensator for these different cases were evaluated by calculating IAE_{before} from 1000 s to 4000 s and IAE_{after} from 7000 s to 10000 s. The compensator performed well in all cases as shown in Table 4.2.

Table-4.2: Impact of stiction parameters on stiction compensation

SU,SD, J	Detuning parameter, α	Time trend	IAE_{before}	IAE_{after}
3,3,3	10		4.05	0.081
7,5,3	10		5.70	0.080
10,6,4	40		7.45	0.083
10,8,6	10		10.20	0.084

4.4 Chapter Summary

The performance of the proposed compensator has been evaluated in this chapter. The proposed compensator can mitigate stiction effect satisfactorily. It can track set point and reject disturbance for various combination of FOPTD processes. The performance of the proposed compensator under various perturbations has also been discussed.

CHAPTER 5

COMPARISON OF PERFORMANCE OF DIFFERENT COMPENSATION METHODS

A good compensator should have oscillation reduction, set point tracking, and disturbance rejection capability with a minimum number of valve reversals. This chapter compares the performance of the proposed compensator with other compensators appeared recently in the literature. For this purpose, the integral absolute error *(IAE),* valve travels *(VT),* variance of process variable (σ^2) and performance indices ( ) are calculated and compared with each other.

5.1 Stiction Compensators

Recently appeared compensators such as: (a) Hägglund (2002) (b) Srinivasion and Rengaswamy (2008) (c) Ivan and Lakshminarayanan (2009) (d) Aurunugum et al. (2010) (e) Arifin et al. (2014) (f) Capaci et al. (2016) stiction compensators have been programmed and used in simulation to compare with the proposed compensator. The following FOPTD process model was used:

$$P(s) = \frac{50}{100\,s+1} e^{-10\,s} \tag{5.1}$$

The PI controller based on *IMC* was:

$$C(s) = 0.046(1 + \frac{1}{100\,s}) \tag{5.2}$$

Valve stiction was introduced using Choudhury et al. (2005) stiction model setting the parameters: *S=5, J=3*. A white noise sequence with zero mean and standard deviation, σ = 0.01 was added. The compensator was started for every model after 5000 s and set-point was changed from 10 to 20 at 10000 s. The simulation was run for 15000 s.

5.2 Comparison of Trend Analysis

The trends of process variable, *PV* and set-point, *SP* for different compensator models are shown in Figure 5.1. As can be seen from this figure, all compensator are able to reduce the process oscillations and track the set-point successfully. However all of them except Capaci et al. (2016) and the proposed compensator could only reduce the amplitude of oscillations at the price of increasing the frequency of oscillations. As a result, the number of valve reversals has been increased significantly.

The Capaci et al. (2016) model could reduce the oscillations successfully but oscillation returns for a certain period of time for a set-point change at 10000s. But for the proposed compensator, oscillations have not come back for the set-point changes. Thus, the proposed compensator outperforms all other compensator appeared in the literature.

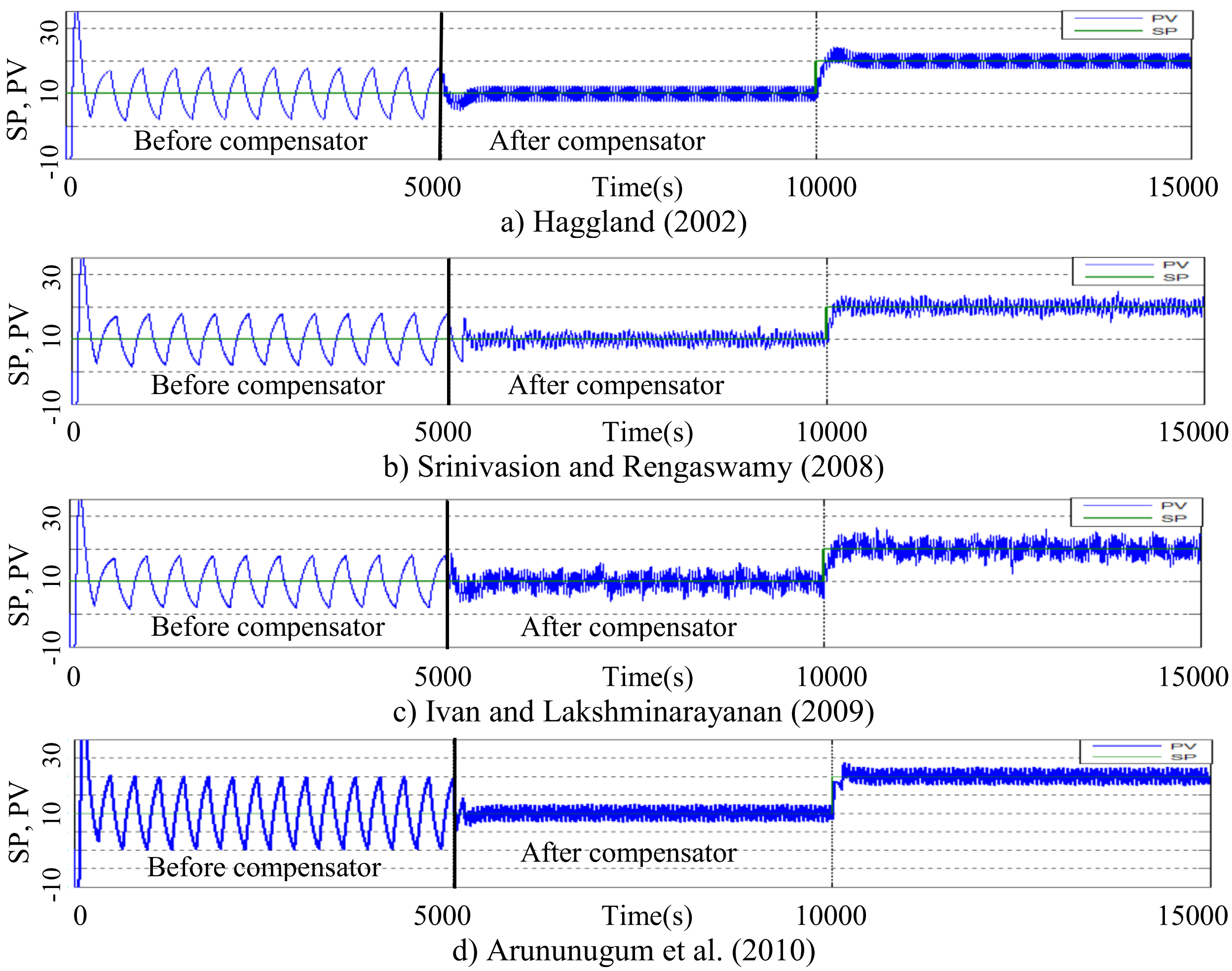
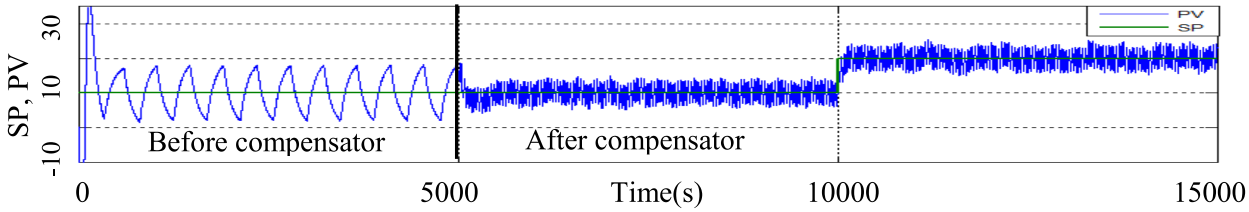
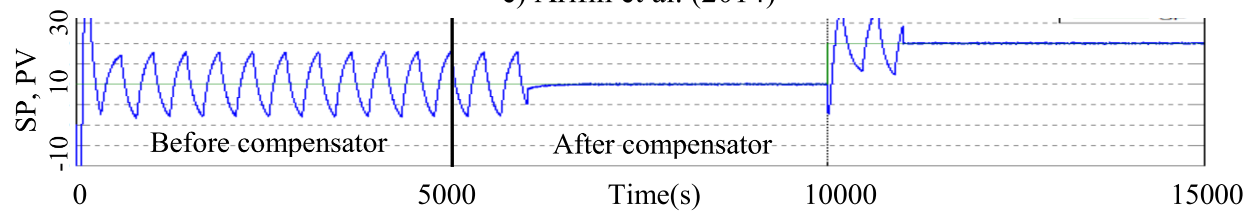


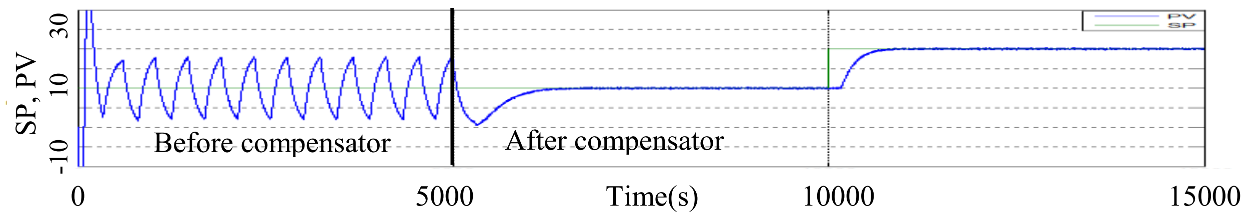
Figure 5.1 (a): Response curve of output process variable, *PV* and set point, *SP* for different compensator model.



e) Arifin et al. (2014)



f) Capaci et al. (2016)



g) Proposed Model

Figure 5.1 (b): Response curve of output process variable, *PV* and set point, *SP* for different compensator model.

The valve stem position, *MV* along with controller output, *OP* are plotted in Figure 5.2. From this Figure, it is clearly shown that the first five methods have increased the valve movements and caused a significant increase in valve reversals after the compensator was switched on at 5000 s. This will certainly reduce the longitivity of control valve. A good compensator is expected to mitigate stiction effect and reduces process oscillation without increasing the number of valve revarsals. Figure 5.1 and 5.2 show that the only last two method namely Capaci et al. (2016) and the proposed method can successfully track set-point and reduce oscillation of process variables without increasing number of valve reversals.

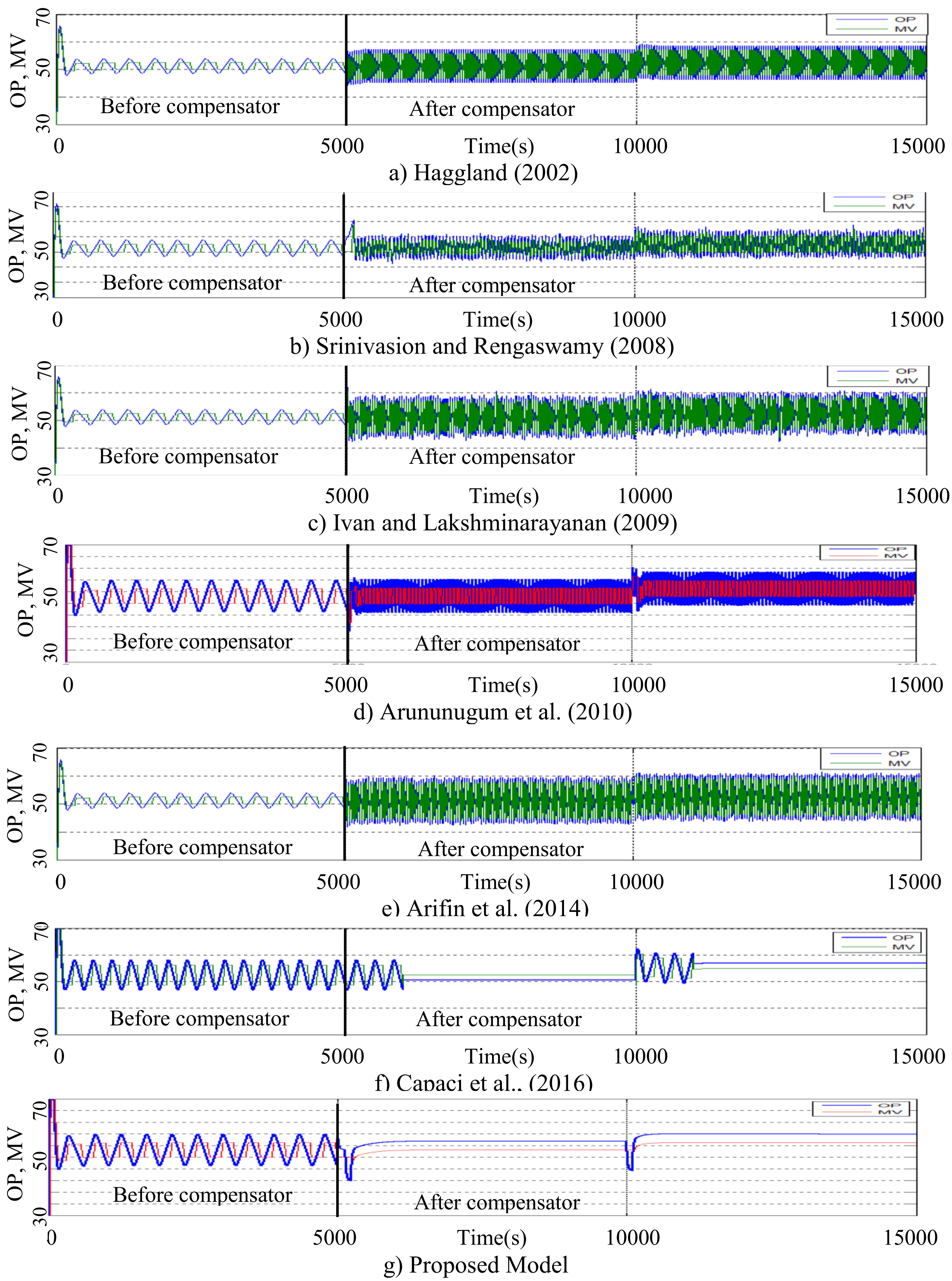


Figure 5.2: Response curve of valve position, *MV* and controller output, *OP* for different compensator model.

5.3 Comparison of Performance

In this section, the performance of different compensators are quantified and compared with each other. The performance can be quantified using the integral absolute error, *IAE*, process variable variance, σ^2 and valve travel, *VT*. Smaller values of *IAE* and σ^2 mean that compensator can reduce process oscillation, track set-point and reject disturbance satisfactorily. The valve reversals can be analyzed by calculating valve travels, *VT*. Minimum number of valve reversals results smaller *VT*. So a good compensator have minimum *IAE*, $\sigma^2$and *VT*. These parameters are calculated as follows:

$$IAE = \frac{1}{T_b - T_a} \int_{T_a}^{T_b} |e(t)|\, dt \tag{5.3}$$

$$\sigma^2 = \frac{\sum\{PV(t) - \mu\}^2}{N} \tag{5.4}$$

$$VT = \int_{T_a}^{T_b} |x(t) - x(t-1)| dt \tag{5.5}$$

Where, *e (t)* is the difference between set-point and process variable and *x(t)* represents the valve position at time t, μ is the mean value of process variable and N is the total number of data points.

The compensator for every model was activated at 5001 s. The *IAE,* σ^2 and *VT* before compensator were calculated using data for 1000 s to 5000 s and those values after compensator were calculated using data for 6000 s to 10000 s. The values of *IAE* before and after the compensator are shown in Table-5.1. The percent decrease of integral absolute error, *IAE* is also shown in the right-most column of this Table. The proposed method has the minimum *IAE* among all compensators. The proposed compensator could reduce *IAE* by 97.2%, which is maximum among all of them.

Table-5.1: Integral Absolute Error, IAE data for different compensator model.

Name of the Model	IAE_{before}	IAE_{after}	% decrease of IAE
Hägglund (2002)	4.657	1.525	67.3
Srinivasion and Rengaswamy (2008)		1.316	71.7
Ivan and Lakshminarayanan (2009)		2.118	54.5
Aurunugum et al. (2010)		0.854	81.7
Arifin et al. (2014)		2.268	51.3
Capaci et al. (2016)		0.407	91.3
Proposed Compensator		0.1303	97.2

The minimum process variable variance indicates the smaller oscillation in the output process variable. As shown in Table 5.2, before the compensator was started, the PV variance was 27.193 for all cases. All of the compensators can reduce the oscillations but the proposed compensator outperforms all other methods.

Table-5.2: Variance of output process variable, σ^2 data for different compensator model.

Name of the Model	$\sigma^2{}_{before}$	$\sigma^2{}_{after}$	% decrease of σ^2
Hägglund (2002)	27.193	2.873	89.4
Srinivasion and Rengaswamy (2008)		2.318	91.5
Ivan and Lakshminarayanan (2009)		6.016	77.9
Aurunugum et al., (2010)		0.969	96.4
Arifin et al. (2014)		6.496	76.1
Capaci et al. (2016)		0.436	98.4
Proposed Compensator		0.044	99.8

Depending on the valve stem position, valve travel, VT was measured. Actually VT cannot be measured in actual industrial process plants because MV data is not generally available. Thus the measurement of VT is possible for simulation study to evaluate the performance of the compensator. The quantified results of VT for different compensator are summarized in Table- 5.3. The last column shows the percentage change of VT before and after compensator was applied. Actually, VT has increased after starting the compensator than it was before for most cases. The last column of Table- 5.3 shows that only the proposed and Capaci et al. (2016) compensator could reduce the valve travels. In these two methods, valve reached to a steady state position quickly after starting the compensator. Then they could reduce the valve travels by 95% and 76% respectively, whereas other compensators have increased the valve travels in manifolds. Excessive valve movements reduce the longevity of the valve. So, the implementation of these compensators in real industry would damage the control valve within a short period of time. The implementation of the proposed compensator in real industry would be beneficial because it can mitigate the stiction effect without increasing the valve stem movements.

Table-5.3: Valve Travel, VT, data for different compensator models.

Name of the Model	VT_{before}	VT_{after}	% change of VT
Hägglund (2002)	7.561	356.4	4595
Srinivasion and Rengaswamy (2008)		124.5	1547
Ivan and Lakshminarayanan (2009)		425.3	5524
Aurunugum et al. (2010)		69.30	817
Arifin et al. (2014)		473.8	6166
Capaci et al. (2016)		1.799	-76
Proposed Compensator		0.351	-95

The results in Table 5.1, 5.2, 5.3 show that the last two methods are the best stiction compensators among the widely used compensation method available in literature. Between the last two, the proposed method gives better results than the Capaci et al. (2016) method.

5.4 Comparison of the Proposed Method and Capaci et al. (2016) Method.

The analysis in section 5.1, 5.2 and 5.3 show that the proposed compensator and Capaci et al. (2016) method outperforms all other available compensation methods. The performances of both compensators with respective to set-point tracking, IAE, σ^2 and VT

are very close. However, the proposed compensator performs better than the Capaci et al. (2016) method. The performances of these two methods are investigated more elaborately below.

5.4.1 Disturbance Rejection Capability

Disturbance is very common for every process industry. A good compensator should have quicker disturbance rejection capability. A unit step disturbance was applied at 10000s for two different compensator models applied on the same process model described by Equation 5.1. The trends of the process variables for the two cases are shown in Figure 5.3 and 5.4. In Figure 5.3 (Capaci et al., 2016), oscillation returns and stays for about 1000 s before the disturbance can be eliminated. One the other hand, the proposed compensator can quickly reject the unit step disturbance within a short period of time as shown in Figure 5.4. It is noteworthy to mention that when compensator was switched on at 5000 s, the proposed compensator could remove oscillation earlier and perform better than Capaci et al. (2016). (See Figure 5.3 and 5.4)

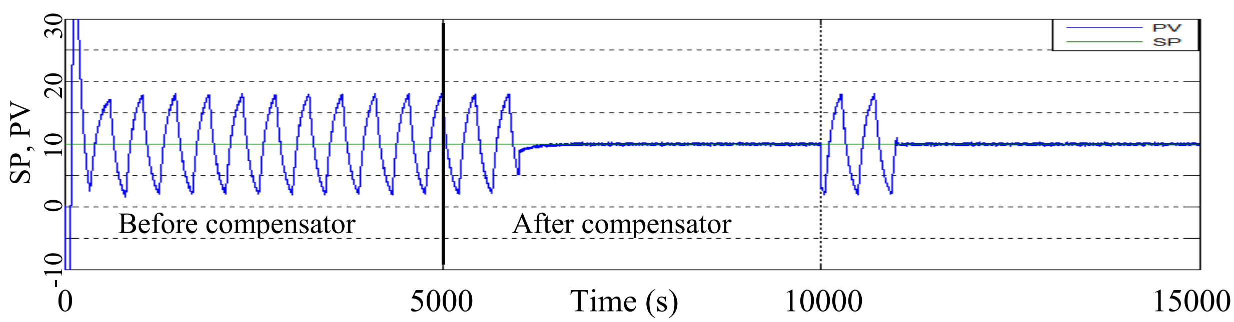


Figure 5.3: Unit step disturbance rejection capability for Capaci et al. (2016) method.

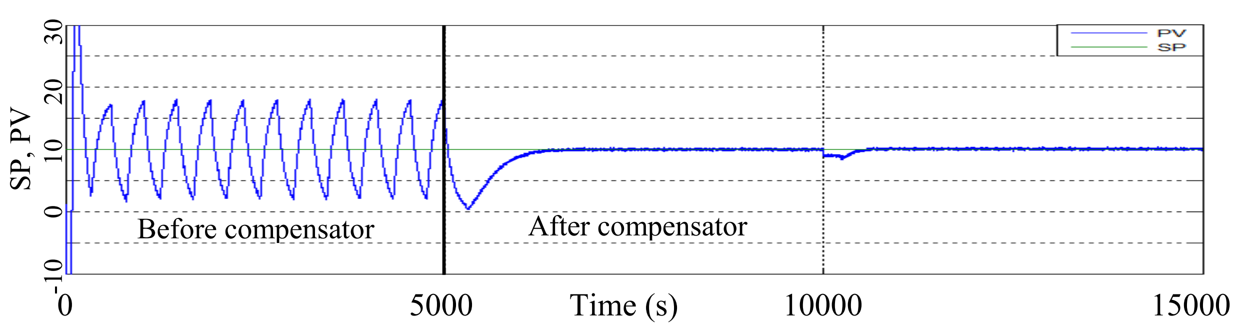


Figure 5.4: Unit step disturbance rejection capability for proposed compensator method.

5.4.2 Uncertainty in Stiction Amount

The Capaci et al. (2016) method is strictly dependent on the estimation of stiction parameter. Accurate stiction detection is a priori in their work. Incorrect detection of stiction parameters can produce wrong results. It can reduce the oscillation but cannot track the set point. As shown in Figure 5.5, there is an offset between *PV* and *SP*.

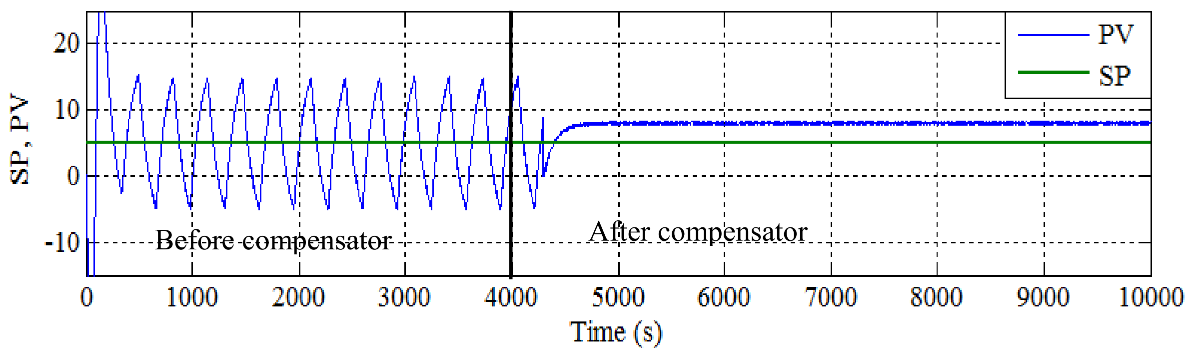


Figure 5.5: Process response curve of unknown stiction parameter for Capaci et al. (2016) method.

On the other hand, for the proposed compensator, there is no requirement of knowing values of stiction parameters. As shown in Figure 5.6, the proposed compensator can remove oscillations and at the same time, can track the set-point accurately.

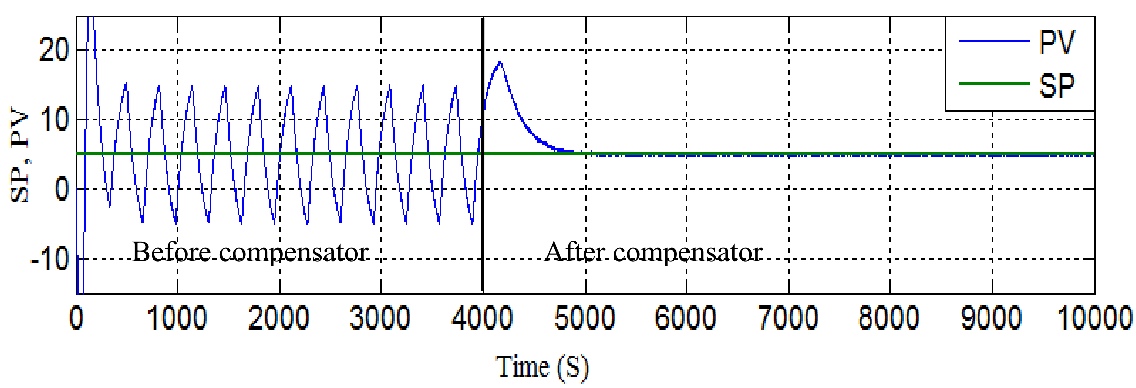


Figure 5.6: Process response curve for proposed method for the case of unknown stiction parameters.

Accurate measurement of stiction parameters for a real industrial plant are difficult. Therefore, the implementation of the Capaci et al. (2016) compensation technique in real plant will be cumbersome. The proposed compensator is simple and can be implemented in a real plant without knowing the extent of stiction in control valve.

5.4.3 Non-Homogeneous Stiction Case

The Capaci et al. (2016) compensation technique works only for constant parameter stiction model. It cannot be used for the case of non-homogeneous stiction model. Non-homogeneous stiction means that the values of stickband plus deadband for upward, SU and downward, SD direction of valve travel could be different. The proposed compensator can also work for non-homogeneous stiction parameters. A sample results for non-homogeneous stiction parameters is shown in Figure 5.7. A modified version of stiction model *(SU=7,SD=5,J=3)* appeared in Choudhury et al. (2005) was used to evaluate the performance of compensator for non-homogeneous cases of stiction.

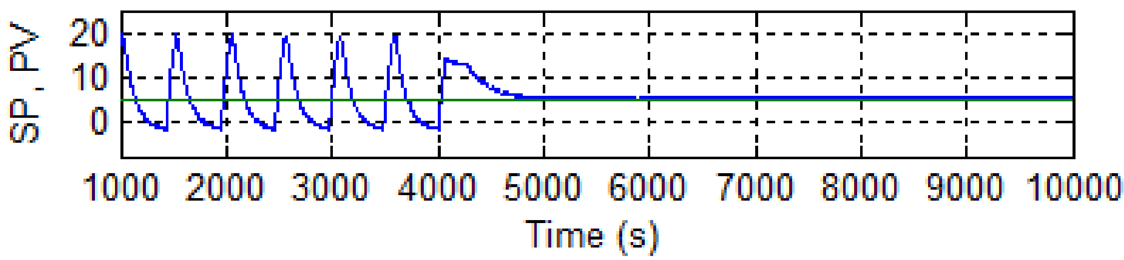


Figure 5.7: Simulation results of proposed model for non-homogeneous stiction parameters.

5.4 Chapter Summary

In this chapter, the performances of the available stiction compensators along with the proposed compensator are compared. The process output trends, the integral absolute error, process variable variance and the valve travel values were compared for seven different compensation methods. It was found that the proposed compensator outperforms all other compensation methods available in the literature.

CHAPTER 6

EXPERIMENTAL EVALUATION OF THE PROPOSED COMPENSATION TECHNIQUE

The industrial chemical processes generally contain a thousands of process control loops. The product quality greatly depends on performance of such control loops. But control loop suffers from poor performance due to nonlinearity or oscillation of the final control element, control valve. Among the many types of non-linearities in control valves, stiction is the most common in the process industry. Compensation of control valve stiction thus has become an important topic of research and development. This chapter discusses the experimental evaluation of the proposed stiction compensation method applied to a level control loop of a computer interfaced pilot plant located in the process control laboratory of Chemical Engineering Department, BUET

6.1 Description of the Plant

The pilot plant set up located in Chemical Engineering Laboratory, BUET consists of four tank level plus temperature control system. The small tank with a level control loop was used to carry out the experiment. A photograph of experimental set-up is shown in Figure 6.1 and a schematic diagram of the level control loop is shown in Figure 6.2.

Figure 6.1: Experimental set-up of the water level control system.

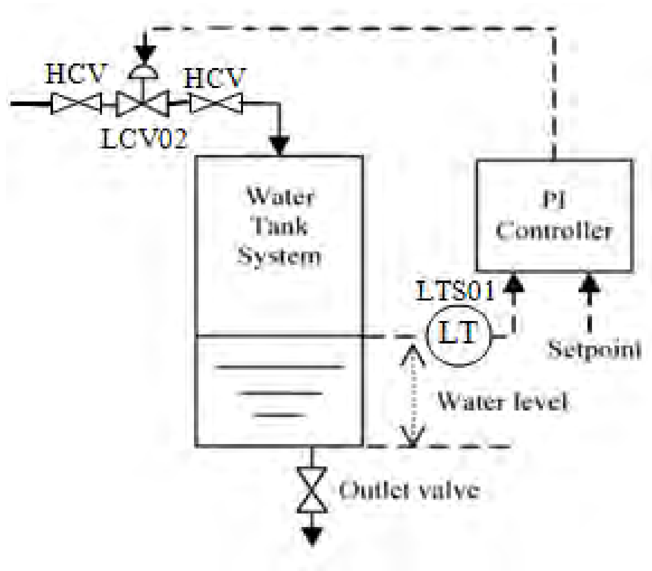


Figure 6.2: Schematic diagram of the water level control system.

A brief description of the various components of the system is given below:

i. Small Tanks

A small tank of cylindrical shape is located in the pilot plant. The tank was attached with a stand. Material of construction of the tank is glass. It was 2 ft long with a 4 inch in diameter.

ii. Pneumatic Control Valve

The pneumatic control valve in the pilot plant is fail close type. Material of construction of the valve is SS, internal threaded. Size of each valve is 1/2 inch in diameter.

iii. Transmitter

There is a level transmitter with the tank. The transmitter is of Model 1151 Smart Pressure Transmitter manufactured by Fisher-Rosemount Inc.

iv. Manual Valves

There are two types of manual valves in the set-up. These are ball valves and needle valves. Material of construction for all type of valves is SS, internal threaded. Size of the needle valves is 1/2 inch and ball valves are of 1 inch, 1/2 inch and 1/4 inch.

v. Compressor

A centrifugal compressor is used to supply instrument air to the pilot plant. Compressed air is necessary to operate the pneumatic control valve. The compressor was used to supply the required compressed air to the valve.

vi. Water Supply Line

All pipe line for transporting water are made of SS. The size of main pipe line is 1/2 inch in diameter, Schedule 10 and the size of the lines that goes to the transmitters is 1/4 inch diameter, Schedule 10.

vii. Instrumental Air Supply

From the instrument air header, air is supplied to operate the pneumatic control valves. Minimum air pressure required for the system is 4 bar. The material of construction of the header is SS. Size is 1 inch, Schedule 10. 6 mm PVC tubes are used as pneumatic cables.

The pilot plant has SISO (single input single output) configuration containing a level control loop. Water was coming through the inlet pipeline of the tank. There was a flow control valve named LCV02 in the water inlet line. The process objective was to maintain the level of the tank at a desired value. To maintain the level of the tank, the inlet flow rate of water must be regulated using the control valve LCV02. Level of the tank was measured and transmitted using a level sensor LTS01. For automatic control configuration, this would make feedback control loop for the process.

The process was interfaced with MATLAB Simulink software through OPC server connectivity.

The block diagram is shown in Figure 6.3.

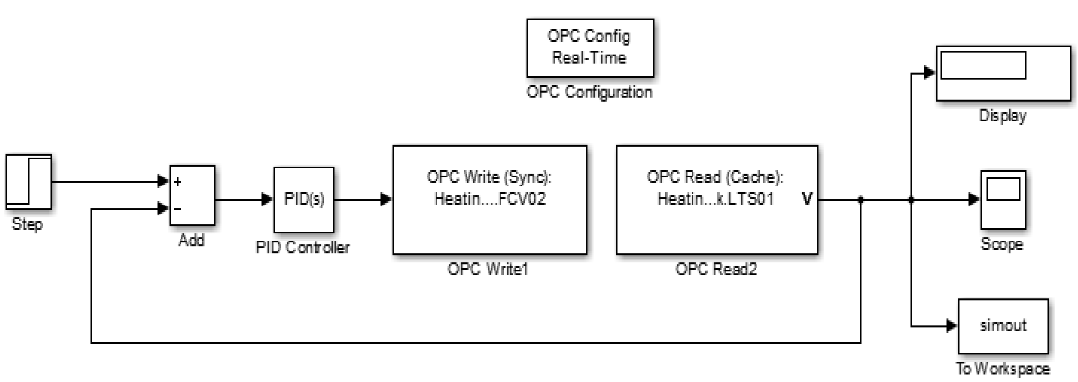


Figure 6.3: Simulink block diagram of a level control feedback loop.

6.2 Operating Procedure

The following is the operating procedure of the pilot plant

1. The computer connected to set up was turned on
2. Power was supplied to the Adam 5000TCP/IP data acquisition module
3. The computer and the module were connected by launching the Adam software
4. The compressor was turned on.
5. The HCVs along the pneumatic control valve were opened
6. The Modbus_TCP-Server_V2_1_4_001 software was launched. The software reads and displays level of tank and valve opening.
7. A library of Simulink file was prepared for this work. This can be used to build new Simulink files for running experiments.

6.3 Model Identification

For the development of a model for the small tank level control system, the simulink block diagram shown in Figure 6.4 was used. MATLAB Simulink and OPC toolbox were used to build it.

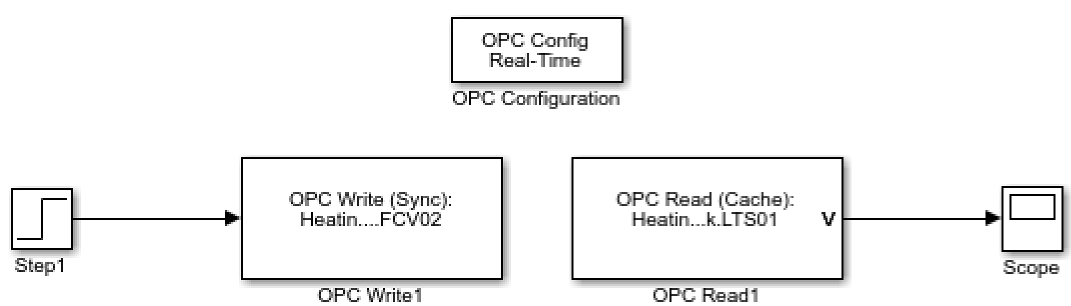


Figure 6.4: Simulink block diagram of open loop configuration.

During the experiment, at first the system was allowed to come to a steady state with respect to the level of water for an initial valve input signal of 13.5 mA to LCV02. The tank level data is shown in Figure 6.5. After steady state was reached, the valve input signal was changed from 13.5 mA to 14.0 mA at 630 s. As a result, the level of the tank reached to a new steady state value of 57% from the previous steady state of 15%. The sampling time for the open loop step test was 1 s.

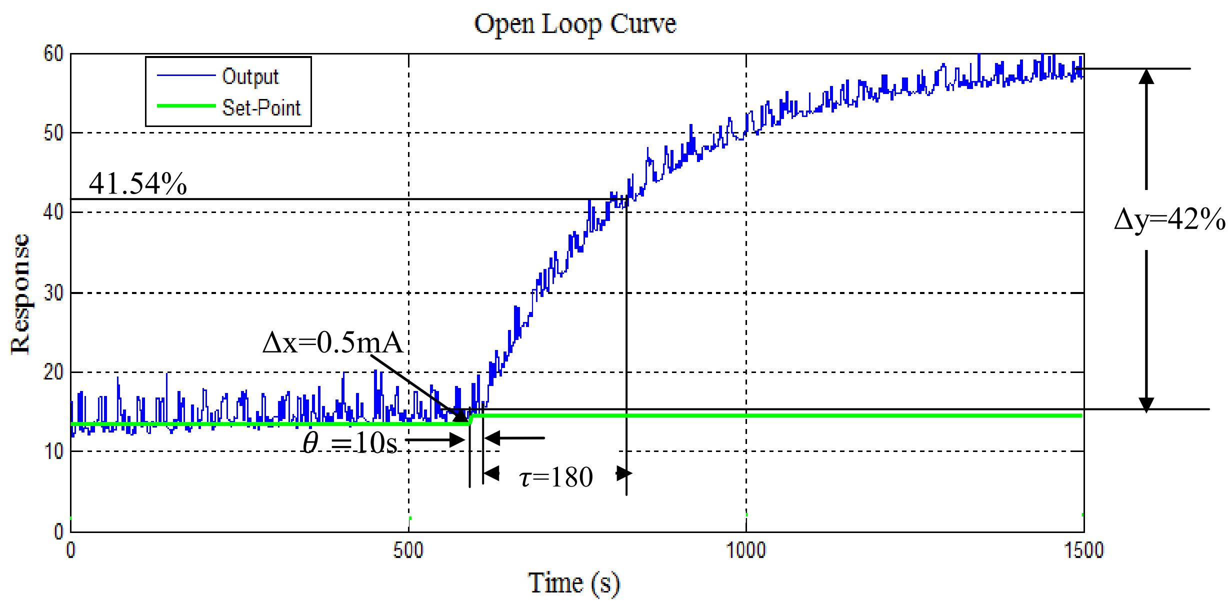


Figure 6.5: The open loop response curve for the pilot plant.

Figure 6.5 shows that it can be modeled using a First Order Plus Time Delay (FOPTD) model. The general transfer function for such a process is:

$$G_P(s) = \frac{K_p e^{-\theta s}}{\tau s+1} \tag{6.1}$$

The model parameter can be calculated as follows,

$$\text{Process gain, } K_P = \frac{Change\ in\ output}{Change\ in\ input} = \frac{\Delta y}{\Delta x} \tag{6.2}$$

$$\text{From Figure 6.5, } K_P = \frac{(57-15)\%}{(14-13.5)} = 84\ \%/mA$$

The time delay, θ is the time lag between input and output response. From Figure 6, it can be estimated as:

$$\theta = (640\text{-}630)\ \text{s} = 10\ \text{s}$$

For a first order process, time constant, τ is the time when the system reaches 63.2% of its ultimate value. The 63.2% of the ultimate response is 15+ (57-15) × 0.632 = 41.54% level for this system. This has been shown in Figure 6.5. The corresponding time to reach 41.54% value for the level can be read from the x-axis of Figure 6.5, which is 820 s.

Therefore,

$$\tau = (820\text{-}\theta\text{-}640)\ \text{s} = (820\text{-}10\text{-}630)\ \text{s} = 180\ \text{s}$$

Thus from the above open loop test, the transfer function of the process was obtained as:

$$G_P(s) = \frac{84e^{-10s}}{180s+1} \tag{6.3}$$

6.4 PI(D) Controller Design

The controller parameters, K_C and τ_I were calculated using the Internal Model Control (*IMC*) rules. The controller parameters were obtained by fine tuning the PI(D) controller after applying it to the process without any compensator and without any stiction.

The final controller settings were obtained as:

$$C(s) = 0.05(1 + \frac{1}{50\ s}) \tag{6.4}$$

The output process variable along with set-point for the final controller settings is shown in Figure 6.6. In this figure, the set-point was changed from 20% to 30% at 1000 s and 30% to 40% at 2000 s. The process output could track the set-point successfully for all cases without producing any oscillations. So, it can be said that the controller settings in Equation 6.4 is a good controller for this process.

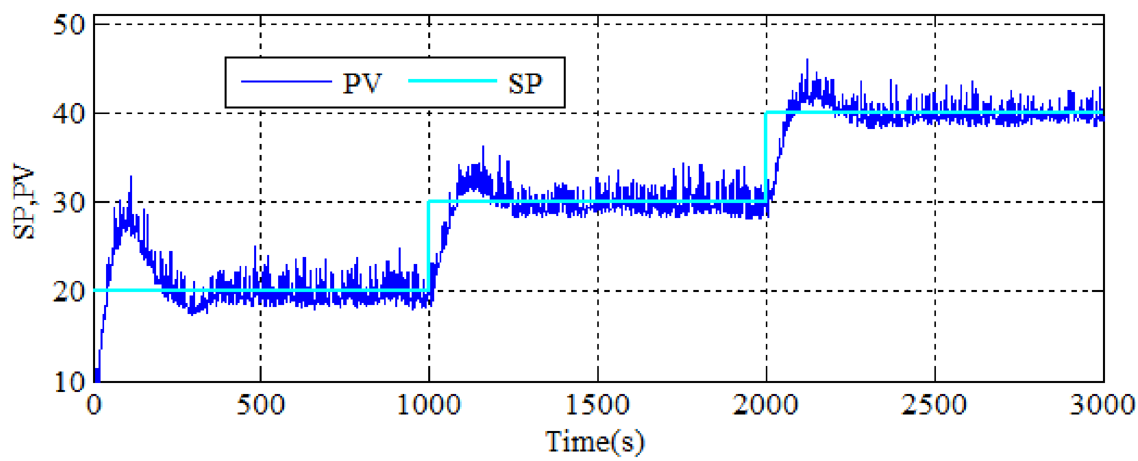


Figure 6.6: The process output results for the non-sticky condition using the final controller settings.

6.5 Experimental Evaluation

The control valve used for this study was non-sticky. A soft stiction block using two parameter Choudhury et al. (2005) stiction model was used to introduce stiction in the control loop. The Simulink block diagram for the evaluation of performance of the proposed compensator is shown in Figure 6.7.

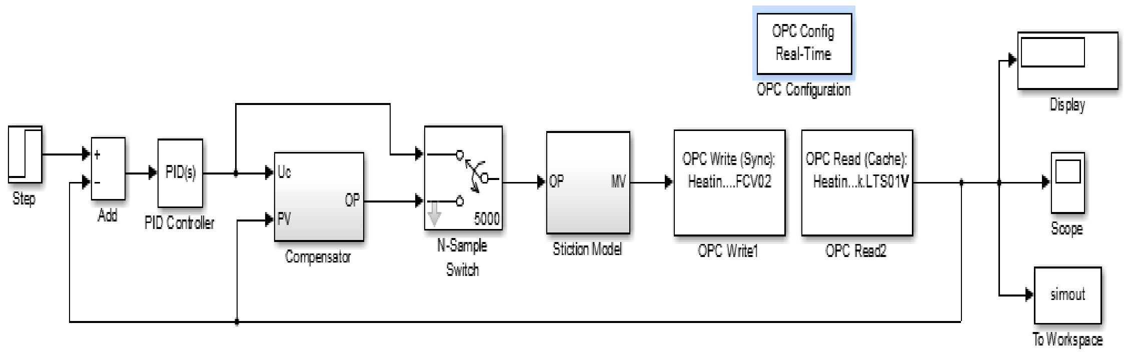


Figure 6.7: Simulink block diagram for the implementation of proposed compensator.

Stiction parameters were set as *S=5* and *J=3.*

The PI(D) controller was as in Equation 6.4.

The parameters of the compensator were set as: $T_s = 1\ s$, $T_G = 100\ s$, $A_P = 1$, $\epsilon = 3\%$, $\alpha = 25$

The process variable, level measurement was very noisy. The variance of it was 2.01% without stiction. Considering this variance, the threshold limit was set at $\epsilon = (1.5 \times 2.01\%) = 3\%$.

The detune parameter, α was calculated from the emperical equation developed previously in Chapter 3. (Equation 3.12)

The emperical equation for α was as follows:

$$\alpha = -12 + \frac{834}{\left\{\frac{3\tau_I}{(\tau_I+3\theta)K_C}\right\}^{0.3} + (\tau_I^{0.7} + \theta^{0.9})} \tag{3.12}$$

The values of controller parameters (K_C =0.05 and $\tau_I = 50$) were obtained from the controller setting of Equation 6.4. The value of time delay for this control loop as calculated from the open loop step test was found to be $\theta = 10\ s.$

Therefore, the detune parameter, α was found to be 22 from the above equation.

Figure 6.8 shows the results of the implementation of stiction compensation scheme. As shown in Figure 6.8, from 0 to 1000 s, there was no stiction compensation. The lower panel of this Figure shows that the valve stem moves ups and down in between 61% to 67% as well as controller output oscillates from about 60% to 68% due to valve stiction. Therefore, the level of water in the tank oscillates from 20% to 40%. At 1000s, the proposed compensation scheme was applied. After starting the compensator, the valve became almost stack at desired position till the set point changes. The valve stem would stack again at new desired position for corresponding set-point changes. So, the compensator could remove the oscillation of the output process variables satisfactorily. The set-point was changed from 30% to 40% at 3000 s, 40% to 50% at 6000 s and 50% to 60% at 8500 s. The compensation quality was not deteriorated due to change in set-point. Therefore, the proposed compensator could track the set-point change and could also remove oscillation completely.

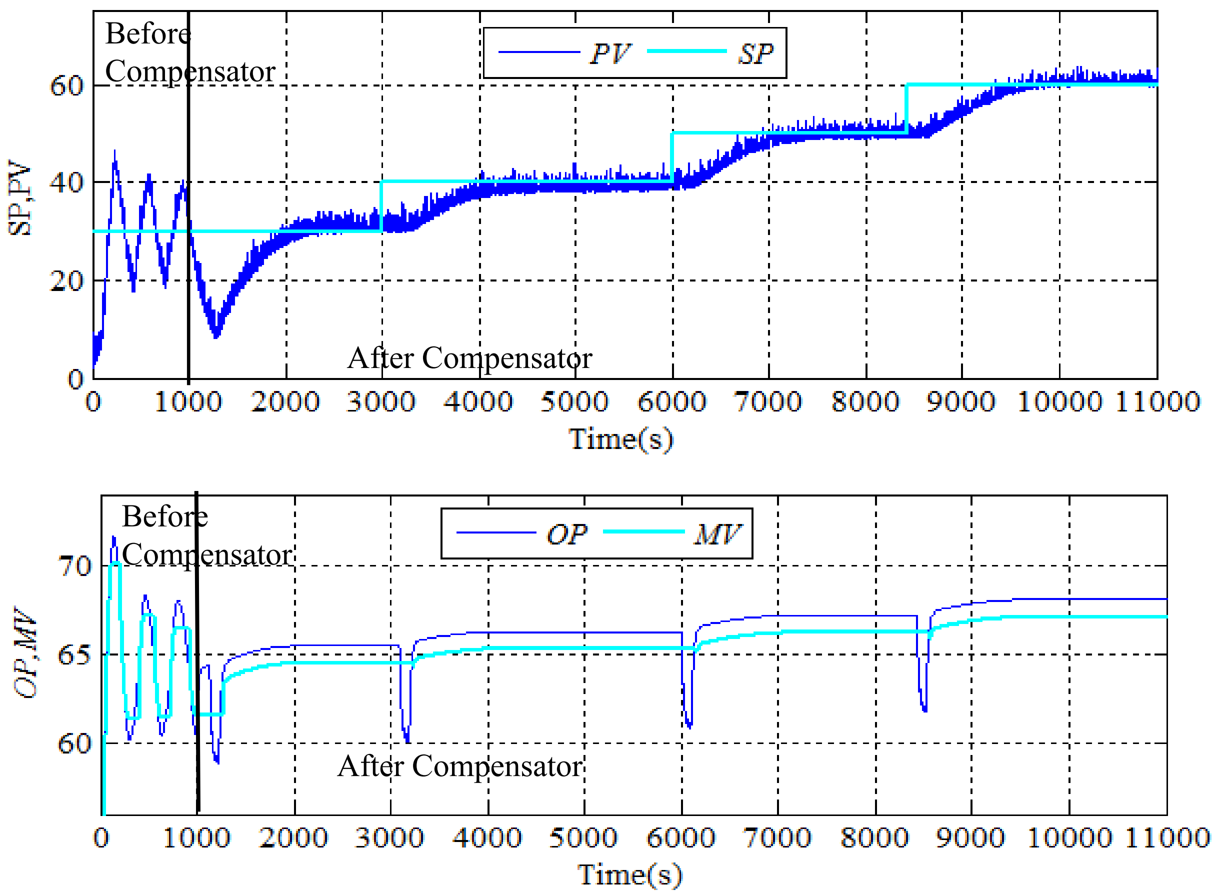
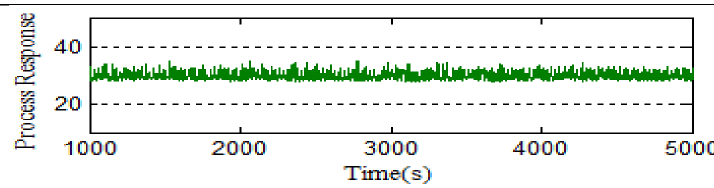
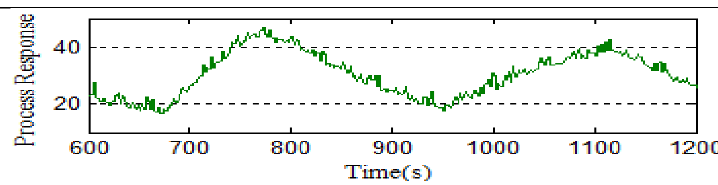
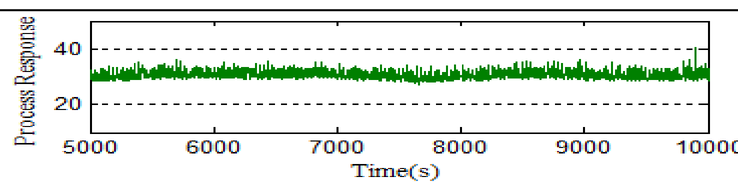


Figure 6.8: Experimental results of the proposed compensator.

Three cases namely 1) Non-sticky valve 2) Stiction without compensator and 3) Stiction with compensator are considered for further analysis. The variance of the process variable for three cases was calculated and listed in Table 6.1. For non-sticky valve, the process variable was mainly noisy and does not show any oscillation. This results in a variance of 2.01% as shown in the top row in Table 6.1. For the case of sticky valve, the process variable was oscillatory due to stiction before starting the compensator. Consequently, the variance of the process variable was very high. It was 58.72% as shown in the middle row of Table 6.1. After the compensator was switched on, it eliminated the oscillation completely and the process variable has almost returned to the previous state of non-sticky valve case. The variance of the process variable is almost same as the variance of the process variable for the case of non-sticky valve. Therefore, it can be said that the proposed compensator was able to mitigate the stiction effect quite effectively.

Table 6.1: Process Variable variance for different conditions.

Sl. No.	Conditions	Time trend	Process variable variance
1	Non-Sticky valve		2.01
2	Stiction without compensator		58.72
3	Stiction with compensator		2.11

6.6 Chapter Summary

This chapter provides a description of the tank level control set-up located at the process control laboratory of the Department of Chemical Engineering, BUET. A brief operating procedure for this system is provided. This chapter shows the identification result of the FOPTD process. It also evaluates the performance of the proposed compensator through experimental validation.

CHAPTER 7

CONCLUSION AND RECOMMENDATION

7.1 Conclusions

Chemical industries are very sophisticated and consist of hundreds or thousands of control valves. Stiction in control valve produces oscillation in process variables and reduces the product quality. If a control valve is found to be sticky, the permanent solution is to send it to workshop for maintenance. In most cases, it cannot be done without shutting down the plant. Therefore, it is a challenging task to develop a compensator to mitigate stiction effects while the plant is running. This thesis has developed such a stiction compensator. The main contribution of this study can be summarized as follows:

1. A new stiction compensator has been developed to mitigate the adverse impact of valve stiction. It can be applied online without stopping the process plant.
2. Methods for estimating the parameters of the proposed stiction compensator have also been developed.
3. The performance of the proposed compensator has been extensively evaluated for different First Order Plus Time Delay (FOPTD) processes through simulation study.
4. The performance of the proposed compensator was compared with other compensators available in the literature such as Hägglund (2002), Srinivasion and Rengaswamy (2008), Ivan and Lakshminarayanan (2009), Aurunugum et al. (2010), Arifin et al. (2014), Capaci et al. (2016). It was found that the proposed compensator outperforms all of them.
5. Finally, the proposed compensator was applied to a level control loop in a pilot plant experimental set up and was found to be successful in removing oscillation due to valve stiction.

The developed compensator has the desired capability of minimizing the process variability with a minimum number of valve reversals. It has good set-point tracking and disturbance rejection properties. The compensation scheme is simple yet powerful. It requires minimum process knowledge. It would not be difficult to implement it in process plants.

7.2 Recommendations for Future Work

There is a scope of improvement and extension of this research work. The following recommendations show some directions to future work.

1. The proposed compensator was developed based on FOPTD process. Though most real processes can be approximated using FOPTD processes, the compensator should be evaluated and modified as necessary for other higher order processes and non-linear processes.
2. The proposed compensator should be applied to a real industrial processes using DCS system.
3. Currently, all available compensators require user-defined parameters based on the process. New research can be directed to automate this process so that the user requires minimum effort to use a compensator.

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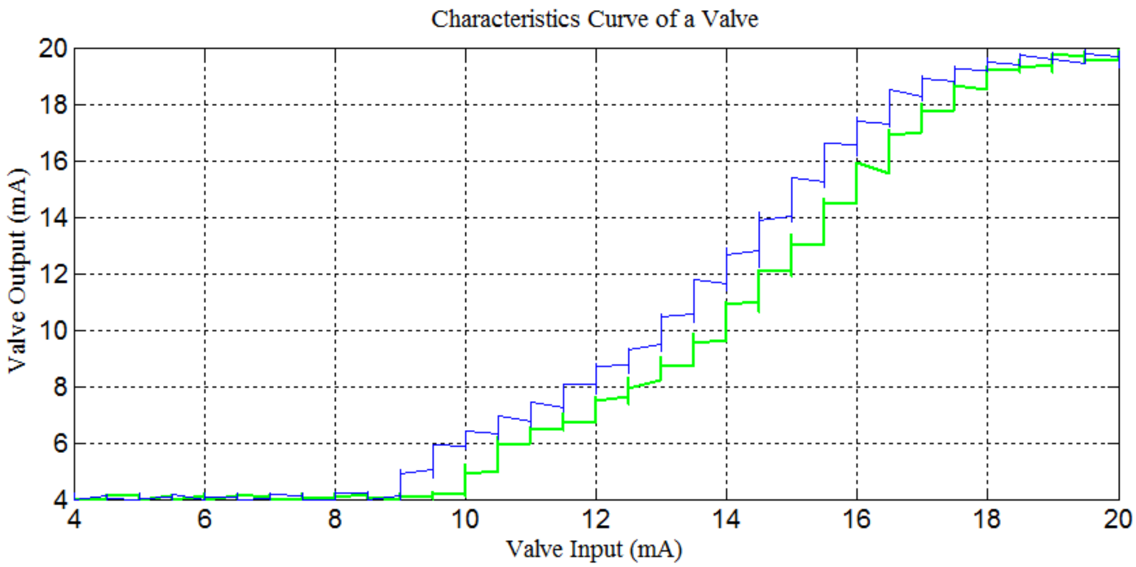
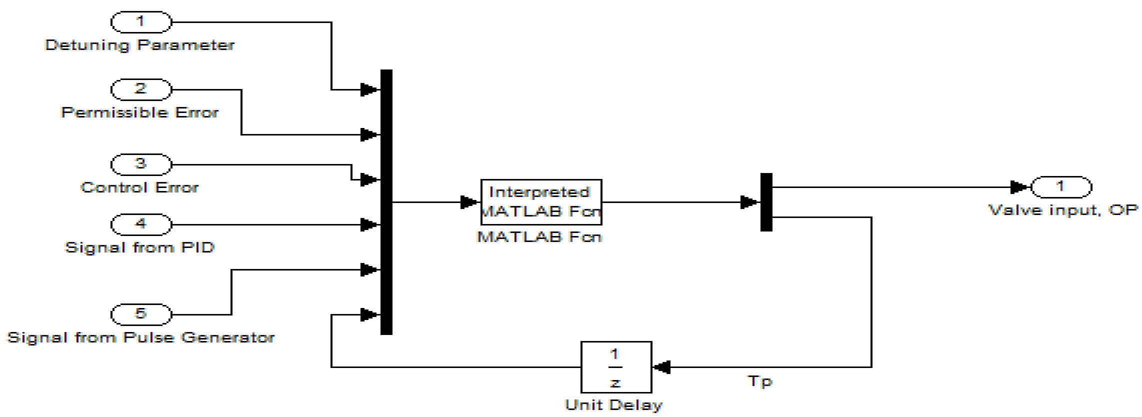
Appendix-A

Characteristics curve of small tank inlet control valve (FCV02) in Process Control Laboratory of Chemical Engineering Department, BUET



Appendix-B

Matlab Algorithm of Proposed Stiction Compensator

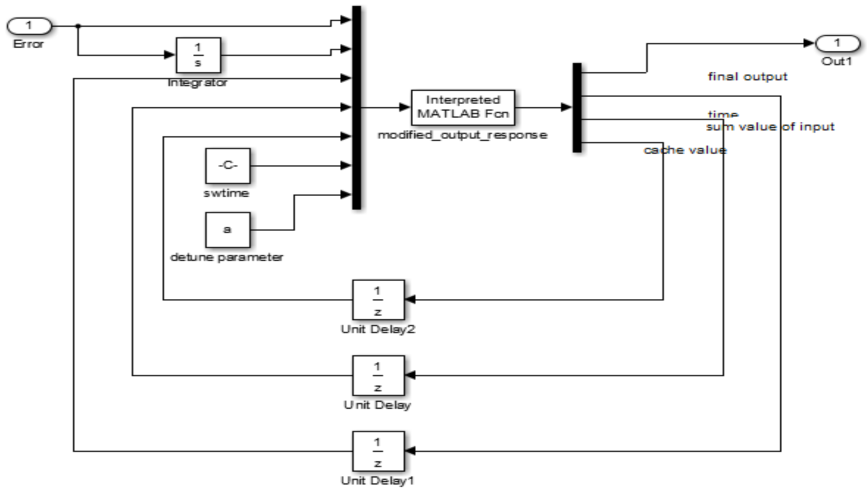



```
function y=CompensationModel(x)
pid=x(4);%signal from controller
detuner=x(1);%detune parameter
error=x(3);%error
diff=x(2);%acceptable deviation
pulse=x(5);%signal from pulse generator
TG=x(6);%for countng total time
TG=TG+1;

if abs(error)<=diff;%check whether the control is acceptable or not
    TG=0;
valve=pid/(detuner);% reduce the aggressiveness of the controller
else
    if TG<100 && error>diff
        valve=(pid/detuner)-pulse;
    elseif TG<100 && error<diff
       valve=(pid/detuner)+pulse;
    else
      valve=pid/(detuner);
    end
end
y=[valve TG];
```

Appendix-C

Matlab Algorithm for Compensator Initialization




```
function Y = integrator(X)
source = X(1);
integrated_source = X(2);
time = X(3);
source_sum = X(4);
cache = X(5);
swtime = X(6);
a=X(7);
if( time<=(swtime - 1))
    source_sum = source_sum + source;
end

final_output = integrated_source;
if (time==swtime)
    final_output = source_sum*a/swtime;
    cache = integrated_source;
elseif(time>swtime)
    final_output = (final_output - cache + source_sum*a/swtime);
end

tp = time + 1;

Y = [final_output tp source_sum cache];
end
```

Appendix-D

Matlab Algorithm to find detune parameter

```
clc
clear all
close all
Ts=1;
e=0.5;
m=[];
q=0;
swtime=4000;% the compensator start
for k=1:5:51 % Process gain
for t=1:10:201 % Process time constant
for d=0:3:21 % Time delay
p=0;
for a=1:0.5:200; % detune parameter
sim('Simmodel'); % MatLab Simulink
p=p+1;
w=(sum(IAE(:,1)))/3000;
A(p) = a
m(p)=w;
end
index1=find (m==min(m));

for ID = 1:length(index1)
    q=q+1;
    min_a = A(index1(ID));
    n4(q,1:5)=[k t d min_a min(m)]
    break;
end
end
end
end
```

Appendix-E

Matlab Algorithm to plot Compensation Results for Different FOPTD Process

```
clc
clear all
close all
Ts=1;
e=0.5;
q=0;
swtime=4000;
j=0;
load IAEVT
for i=1:length(IAEVT)

    if (IAEVT(i,5)<0.1 && IAEVT(i,6)>0)
   if j>=6
      j=0;
      figure
    end
        k=IAEVT(i,1);
        t=IAEVT(i,2);
        d=IAEVT(i,3);
        j=j+1;
        sim('Simmodel_witheq')

subplot(6,2,j)
plot(pv)
grid on
axis([0,10000,-10,20])
pp=['Kp=', num2str(k),',Tau=', num2str(t), ',Theta=', num2str(d)]
title(num2str(pp))
xlabel('Time(s)')
ylabel('Response')
    elseif (IAEVT(i,5)>0.1 && IAEVT(i,6)>0)
        out(i,1:3)=[IAEVT(i,1) IAEVT(i,2) IAEVT(i,3)];
    else
        notimc(i,1:3)=[IAEVT(i,1) IAEVT(i,2) IAEVT(i,3)];
  end
end
```

Appendix-F

Sample Matlab Algorithm to Fit the Detune Parameter

Model 1

```
clc
clear all
close all
load acvalidc
X0 = ones(length(acvalidc),1);
ii=1;
SSE = 10000000000;
for power_kp = 0:.1:5
    for power_tau = 0:.1:5
        for power_t = 0:.1:5
            Kp = acvalidc(:,1); Kp=Kp.^power_kp;
            Tau = acvalidc(:,2); Tau = Tau.^power_tau;
            Th = acvalidc(:,3); Th= Th.^power_t;
            X = [X0 Kp Tau Th];
           Y = acvalidc(:,4);
           theta1 = pinv(X'*X)*X'*Y;
           Y1 = X*theta1;
           nSSE = 0;
           n = length(Y);
           for i = 1:n
               nSSE = nSSE + (Y1(i,1) - Y(i,1)).^2;
           end
           model1(ii,1:4)=[power_kp power_tau power_t nSSE];
           ii = ii+1;
           if nSSE < SSE
               SSE = nSSE;
               power_KP = power_kp;
               power_TAU = power_tau;
               power_TH = power_t;
           end

           fprintf('%d %d %d\n',power_kp, nSSE, SSE);
        end
    end
end

ANS1 = [1 power_KP power_TAU power_TH 0 SSE]
save model1
save ANS1
```