The Knocker - A Compensator for Stiction in Control Valves

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1995

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ISSN 0280–5316
ISRN LUTFD2/TFRT--7539--SE

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November 1995

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Document name
INTERNAL REPORT

Date of issue
November 1995

Document Number
ISRN LUTFD2/TFRT--7539--SE

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Sponsoring organisation
NUTEK, The Swedish National Board for Industrial and Technical Development

Title and subtitle
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Classification system and/or index terms (if any)

Supplementary bibliographical information

ISSN and key title
0280–5316

ISBN

Language
English

Number of pages
15

Recipient's notes

Security classification

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Abstract. A procedure that compensates for static friction (stiction) in control valves is presented. The compensation is obtained by adding a pulse sequence to the control signal. The characteristics of the pulse sequence are determined from the control action. Industrial tests show that the procedure reduces the control error during stick-slip motion significantly compared to standard control without stiction compensation.

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1. Introduction

It has recently been realized that nonlinearities in the control valves are the largest source of process variability in process control. A Canadian paper-mill audit indicated that about 30% of all control loops suffer from these problems, see Bialkowski (1993). About the same numbers are obtained in other investigations as well, see, e.g., Ender (1993). The problems cause an increased energy consumption, waste of raw material and sometimes a less uniform end product, see Shinskey (1988) and Shinskey (1990).

Recently, a method that automatically detects stick-slip motion caused by stiction in the valve has been developed. See Hägglund (1995). When a valve has got too high friction, maintenance should be undertaken. However, it is often necessary to stop a major part of a production line in order to perform this maintenance. Therefore, the maintenance is often postponed, and the production is continued even after a detection of stick-slip motion.

For these reasons, it is desirable to improve the control when stick-slip motion is detected. Such an improvement will not only yield a more effective production. It may also delay the time until the next stop of the production, since production often has to be interrupted just because of valve problems.

This paper presents a procedure that compensates for static friction in control valves. The compensation is performed by adding a sequence of short pulses to the control signal. Industrial field tests show that the procedure manages to reduce the variations in the measurement signals significantly.

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† Patent pending – Swedish patent-application number 9503286-8.

The paper begins with a description of the control valve and the stiction problems in Sections 2 and 3. Different approaches to compensate for stiction in servo mechanisms are reviewed in Section 4. Unfortunately, these approaches are not directly applicable for stiction in control valves. The major reason for this is the property of the pneumatic positioner and actuator. In Section 5, the new stiction compensation procedure is presented. It is called *the knocker* because of its ability to overcome the friction by "knocking" on the valve using a sequence of pulses. Section 6 shows results from industrial field tests performed with the controller. The paper ends with conclusions, acknowledgements, and references.

2. The control valve

A control valve consists of three main parts: The valve, the actuator that forces the valve stem to move, and the positioner that controls the valve stem position so that it corresponds to the control signal.

In process control, there are many types of valves, actuators, and positioners. The most commonly used positioners and actuators are pneumatic. There are hydraulic and electric actuators, but in process control the pneumatic ones are used in far over 90% of the control loops. Throughout this paper, the pneumatic configuration is considered.

A schematic diagram of a control valve is shown in Figure 1. The *principle* described in this figure corresponds to almost all control valves. However, the details of the design vary between different products. For example, Figure 1 shows an actuator which uses air pressure to move the valve in both directions, whereas several positioners use air pressure on one side and a spring on the other. A crucial part of the configuration is the pilot valve in the positioner. It is interesting to note that the different manufacturers have a similar design of this valve.

The sizes of the valves and the actuators vary considerably, but normally the same positioner is used for all sizes of actuator. This is a feature that is advantageous for the stiction compensation procedure described in this paper.

A large actuator has larger air volumes than a smaller one. This means that with equal positioners it takes a longer time to change the air pressure in a large actuator. On the other hand, since larger actuators are stronger than smaller ones, a smaller pressure drop over the piston is required to move the valve.

3. Stiction in control valves

As mentioned in the introduction, nonlinearities in the control valve is the major reason for control impairment in process control. There is one type of nonlinearity that is related to the characteristics of the valve. The ratio between the flow change and control signal change, i.e., the valve characteristics, is normally not linear. This results in a varying loop gain if a controller with constant parameters is used. However, this nonlinearity can easily be compensated for using gain scheduling. See Hägglund (1991).

The most sever nonlinearities in control valves are static friction (stiction) and hysteresis (backlash). There is, of course, always some amount of

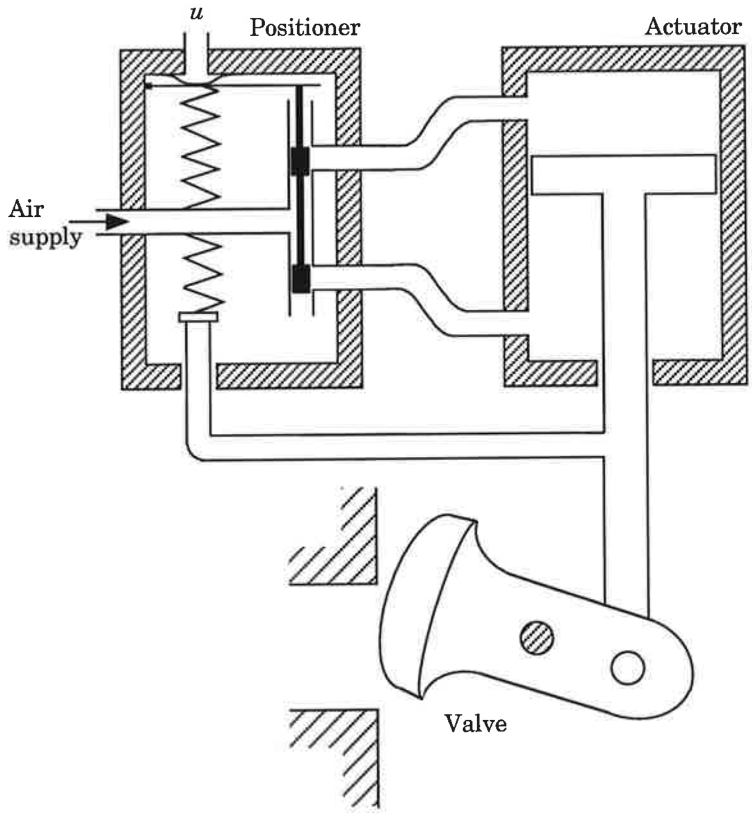

Figure 1. Schematic diagram of a control valve. When control signal u changes, the pressure on the membrane will change. The balance beam will then move the pilot valve so that the air pressure increases on one side of the actuator piston, whereas air is evacuated from the other side. The change in pressure over the actuator piston moves the piston to a new position. This movement is transferred to the valve, and also fed back to the positioner. This will result in a change in the spring force, and the membrane, balance beam, and pilot valve are returned to their equilibrium positions again.

both friction and hysteresis in mechanical configurations. The problem is that the stiction and hysteresis mostly increase gradually during operation, and after some time they give rise to oscillations in the control loop.

Figure 2 shows a recording from a flow loop in a paper mill. The figure demonstrates stick-slip motion caused by stiction in the valve. It is obvious from this figure that the standard PI controller is unable to handle this nonlinearity satisfactorily.

In all sorts of valves, friction appears in the packing boxes around the valve stem, especially when these are tightened hard. The packing boxes are often tightened after some period of operation in order to avoid leakage. In ball valves, ball segment valves, and throttle valves there is often also a significant friction between the ball/throttle and the seat. Hysteresis may appear at several places in the mechanical configuration due to wear and vibrations.

The stiction is varying, both in time and between different operating points. Temperature variations cause friction variations. A high temperature means that the material expands, and therefore that the friction forces increase. Some media give fouling that increases the friction. Particles in the media may cause damage on the valve. The wear is often nonuniform,

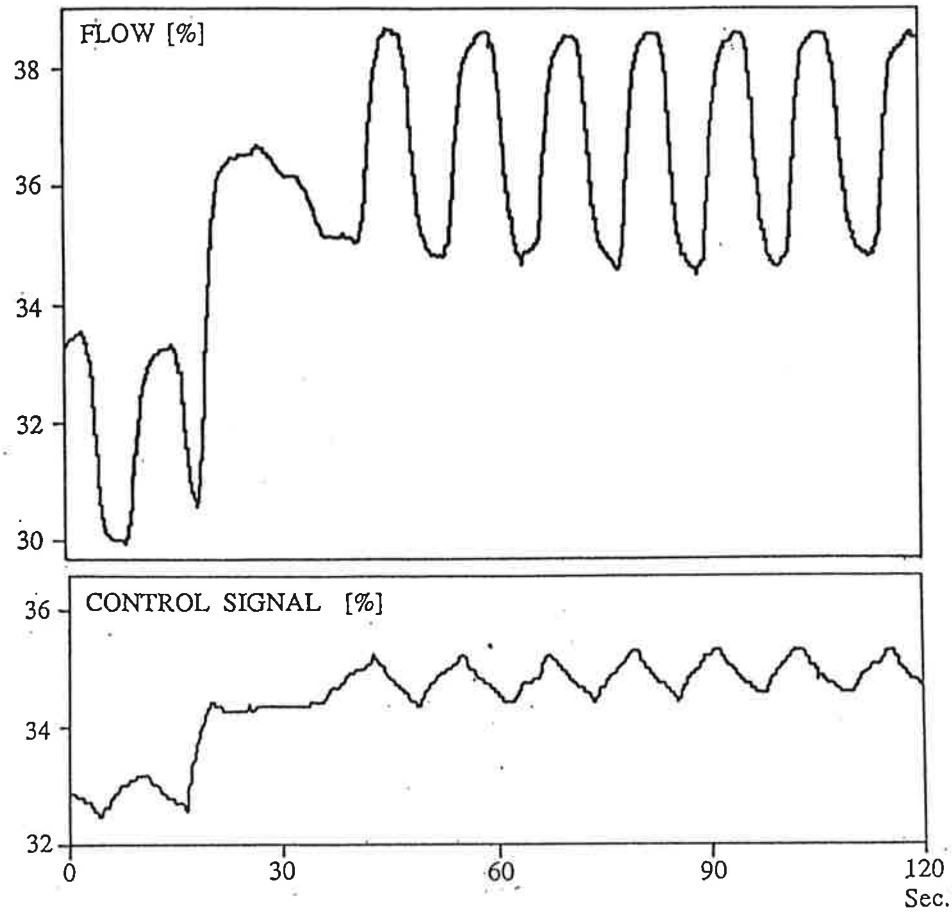

Figure 2. Stick-slip motion in a flow-control loop in a paper mill.

so that the friction is different at different valve positions. Experimental investigations has also shown that the force that is required to overcome the stiction is dependent on the rate at which this force is applied. See Richardson and Nolle (1976).

4. Stiction compensation

There are several approaches to stiction compensation in servo mechanisms. A good overview is given in Armstrong-Hélouvry *et al.* (1994). There are mainly two methods to compensate for stiction, namely dithering and impulsive control.

Dithering

Dithering means adding a high-frequency zero-mean signal to the control signal. The idea is that the amplitude of the dither should be so high that the stiction is overcome, and that the frequency should be high enough, so that the generated disturbance is above the interesting frequency range of the system.

Dithering is unfortunately not useful to overcome stiction in pneumatic control valves. The reason for this is the dynamics between the controller and the stiction point, i.e., the dynamics in the pneumatic positioner and actuator. Even if it were possible to use dithering, the solution would cause a significant wear on the valve.

A dithering signal may perhaps be generated by the positioner, since the pilot valve is rather fast. However, this high-frequency signal will be low-pass filtered (integrated) in the actuator. Since the output from the pilot valve furthermore is limited in amplitude, it is not possible to generate a dithering high-frequency pressure drop over the actuator piston.

Impulsive control

Another approach to compensate for stiction is impulsive control. In contrast to dithering, no stiction-compensation signal is *added* to the control signal, but the control signal itself is generated as a sequence of pulses. The pulses should be so large, that they overcome the stiction level. This approach is appealing for position control using servo motors, where there is an integrator in the process. This means that no pulses have to be generated in steady state.

There are several problems associated with impulsive control, when it is attempted to apply it on control valves. To obtain an accurate control signal that is insensitive to variations in the stiction level, the pulses have to be large with a short duration. Again, this high-frequency pressure drop over the actuator piston is impossible to generate due to the dynamics and limitations in the positioner and actuator. This is especially a problem at stick-slip motion in steady state, when it is desirable to obtain relatively small control actions.

One approach would be to use feedback in the following way. Apply an impulse, and wait for the slip. When the valve slips, remove the impulse immediately. Unfortunately, there are problems with dynamics once again, this time the dynamics *after* the point of stiction. The valve position is not measured. If it is a flow control loop, the flow is measured, but this flow signal is filtered through anti-aliasing filters and other filters. The consequence is, that the controller does not get the information about the valve slip in time to reduce the pulse. Therefore, it is most likely that this approach will increase the amplitudes at stick-slip motion instead of reducing them.

Stiction compensation in control valves

To compensate for stiction in control valves, other approaches than those previously used for servo mechanisms have to be used. The reason for this is the dynamics that surrounds the stiction points.

It is not possible to move the valve with a high-frequency input signal, and it is therefore impossible to avoid stick-slip motion. However, a faster transition between the different stiction positions can be obtained. This higher frequency of the oscillations may improve the control significantly. This is shown in Section 6.

Another question is *where* a compensation of the stiction should be made. There are several so called smart positioners available nowadays, and it is sometimes suggested that stiction compensation should be performed in these positioners. However, there is a major problem with this approach. The positioners measure and control the valve stem position. The problem is, that the stem position may not reveal the correct information about the valve position. Because of the elasticity of the valve stem, the valve may very well be stuck even though the stem moves. See Coughran

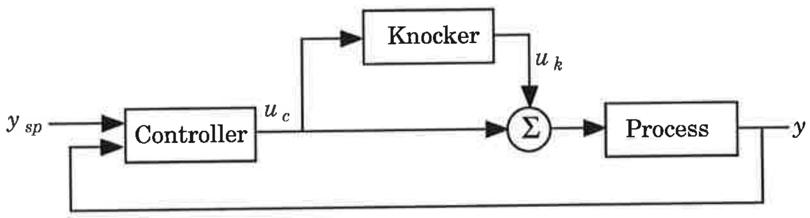

Figure 3. Block diagram illustrating the knocker used in a feedback loop.

(1994). This is a problem especially in rotary valves. The conclusion is that to really obtain the valve position it is necessary to measure the flow through the valve. This fact makes the controller the best candidate for valve supervision and stiction compensation, since the controller takes the flow, perhaps smoothed out, as input signal.

5. The knocker

During stick-slip motion in steady-state, the ideal stiction compensator would be a sequence of pulses added to the control signal, where each pulse has an energy content that exactly compensates for the stiction. With a lower energy content, the valve will remain stuck. With a higher energy content, the slip will be larger than desired. The problem is, that the stiction level is varying and therefore unknown.

Instead, the new stiction compensator consists of a sequence of pulses with a relatively small energy content. The idea behind the new stiction compensation procedure, the knocker, is the following:

> Add short pulses of equal amplitude and duration in the direction of the rate of change of the control signal to the control signal.

With an integrator in the controller, the basis level for the pulses will gradually change as long as the control error is nonzero. This means that the pressure drop over the actuator piston will increase gradually until the valve slips. When the valve slips, the measurement signal will cross the setpoint, and the rate of the control signal will be reversed. This means that pulses with the opposite sign will be added to the control signal.

It is likely that there will be several pulses of "wrong" sign shortly after the slip, since the measurement signal will not react to the slip immediately, and therefore not the control signal either. However, these extra pulses will not do any harm, since the valve is stuck at a new position where the pulses cannot overcome the stiction level. This is the advantage of having a small energy content in each pulse.

The principle of the new stiction compensation procedure is illustrated in Figure 3. Control signal $u(t)$ consists of two terms:

$$u(t) = u_c(t) + u_k(t) \tag{1}$$

where $u_c(t)$ is the output from a standard controller, and $u_k(t)$ is the output from the knocker.

Normally, $u_c(t)$ is the output from a PID controller with parameters gain K, integral time T_i, and derivative time T_d. See Åström and Hägglund (1995). The sampling period of the controller is h s.

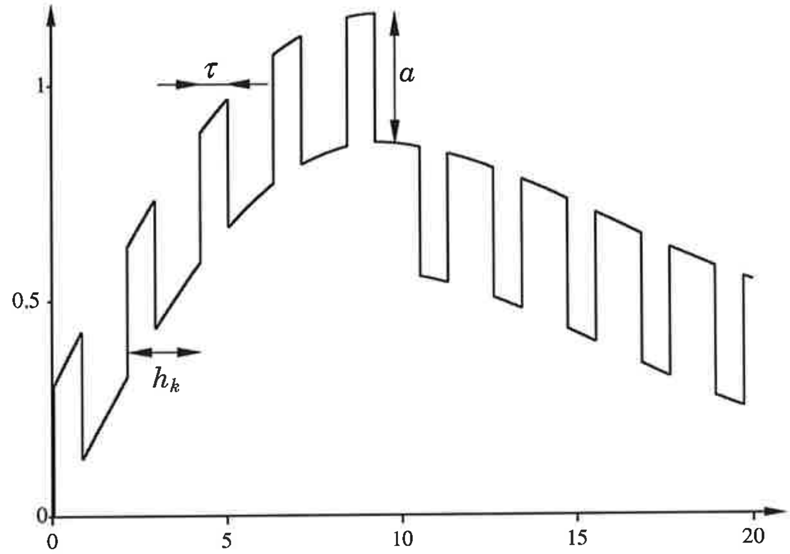

Figure 4. The control signal using the knocker.

Output $u_k(t)$ from the knocker is a pulse sequence that is characterized by three parameters: The time between each pulse h_k, the pulse amplitude a, and the pulse width τ. See Figure 4. During each "pulse interval", $u_k(t)$ is given by

$$u_k(t) = \begin{cases} a\, sign\,(u_c(t) - u_c(t_p)) & t \leq t_p + h_k + \tau \\ 0 & t > t_p + h_k + \tau \end{cases} \tag{2}$$

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

Choice of parameters

The knocker has three parameters, a, τ, and h_k, that determine the characteristics of the pulse sequence. These parameters have to be chosen suitably. The pulses should be sufficiently large, so that the valve slips at an earlier stage than without the knocker. The pulses must also be small enough, so that they do not cause any extra slip before the flow change is observed by the controller.

The transfer function between the knocker output $u_k(t)$ and the process output y is

$$Y = \frac{G_p}{1 + G_p G_c} U_k$$

where G_p is the process transfer function and G_c is the controller transfer function. See Figure 3. If $u_k(t)$ is a pulse with amplitude a and width τ, the process output becomes

$$Y = \frac{G_p}{1 + G_p G_c}(1 - e^{-s\tau})\frac{a}{s} \approx \frac{G_p}{1 + G_p G_c} a\tau$$

This means that the disturbances are proportional to the product $a\tau$. Hence, it is the product $a\tau$ that determines the energy of each pulse in the knocker.

Pulse amplitude a: If amplitude a is too large, the pilot valve will open so much that an uncontrolled evacuation from the low-pressure side of the actuator occurs. See Figure 1. Therefore, it is desirable to keep a relatively small. The field test have shown that it is suitable to choose a in the interval $1\% < a < 4\%$. There is probably no reason to have an adjustable a, but a can be fixed once and for all.

Pulse width τ: It is important to not feed too much energy into the positioner at the moment the valve slips. Therefore, it is desirable to use a relatively short value of τ. For practical reasons, τ cannot be smaller than the sampling interval h. In the field tests, τ was chosen to h or $2h$, where $h = 0.2$ s. These values are suitable for a 150 mm valve, but perhaps τ can be longer for very large actuators.

It might be desirable to have one adjustable knob that is related to the energy $a\tau$ in each pulse. Since it is desirable to keep a constant, this means that τ should be adjustable. Parameter τ may take continuous values, e.g. from one sampling period h up to a fraction of integral time T_i. It could also take a couple of discrete values, for example:

$$\tau_1 = 1h$$
$$\tau_2 = 2h$$
$$\tau_3 = 4h$$

Sampling interval h_k: The sampling period h_k must of course be larger than the sampling period h of the controller. It must also be larger than the pulse width τ. It is desirable to keep h_k short relative to integral time T_i, so that the basis level for the pulses does not change too much between successive pulses. (The effect of using a too long value of h_k is illustrated in Section 6).

It is reasonable to choose the sampling period h_k of the knocker in relation to the pulse width τ. A simple choice is

$$h_k = n\tau$$

where n is in the range $2 \le n \le 5$. It should also be ensured that h_k is reasonable compared to T_i.

Summary It would be most convenient if the knocker could be used without any parameters to be set by the user. This might be possible. One reason for this is that the pilot valves use to be the same for all actuator dimensions. Furthermore, the pilot valves that are found in positioners from different manufacturers are quite similar.

If it is not possible to fix all three parameters, it is suggested to make the pulse width τ adjustable, and to determine the sampling period h_k automatically as a function of τ.

Hysteresis compensation

The knocker is designed to compensate for static friction. However, it appears that it is also suitable to compensate for hysteresis in the valve. If the hysteresis is less than $2a$, the hysteresis band will be crossed when the rate of the control signal changes sign.

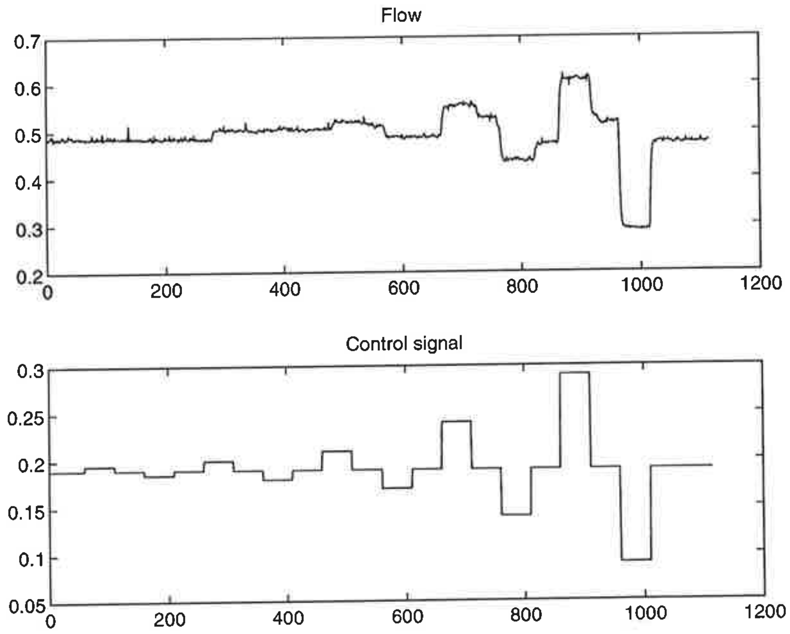

Figure 5. Manual step changes in control signal with magnitudes 0.5%, 1%, 2%, 5%, and 10% respectively. Because of friction and hysteresis there is no flow response at small control signal changes.

6. Field tests

The knocker has, of course, been tested in simulation studies. However, since the dynamics of the positioner and the actuator are complicated, it is not possible to rely only on these simulations. Therefore, the knocker was also tested in industrial field tests. The experiences obtained from these tests are provided in this section.

The set-up

The knocker was tested on a water flow system with a 150 mm ball-segment valve. The valve was in quite a bad shape, with damages on the valve ball. On the other hand, the resulting control was not worse than what is normally encountered in process control. The effects of the friction and hysteresis are demonstrated in Figure 5.

The controller was a PI controller without any dead-zone on the control error. The controller could run with or without the knocker procedure connected. The parameters of the PI controller were gain $K = 1$ and integral time $T_i = 5$ s throughout the experiments. The sampling period of the controller was $h = 0.2$ s.

The flow signal was fed through a first order low-pass filter with time constant $T_f = 5$ s before it entered the controller function.

The field tests

Several experiments were performed using the set-up described above. Some of these are presented below.

Experiment 1 Figure 6 shows 10 000 data points, i.e., about half an hour, from an experiment. A pure PI controller is used in the first phase, and the knocker is connected in the second phase. In this experiment, the knocker

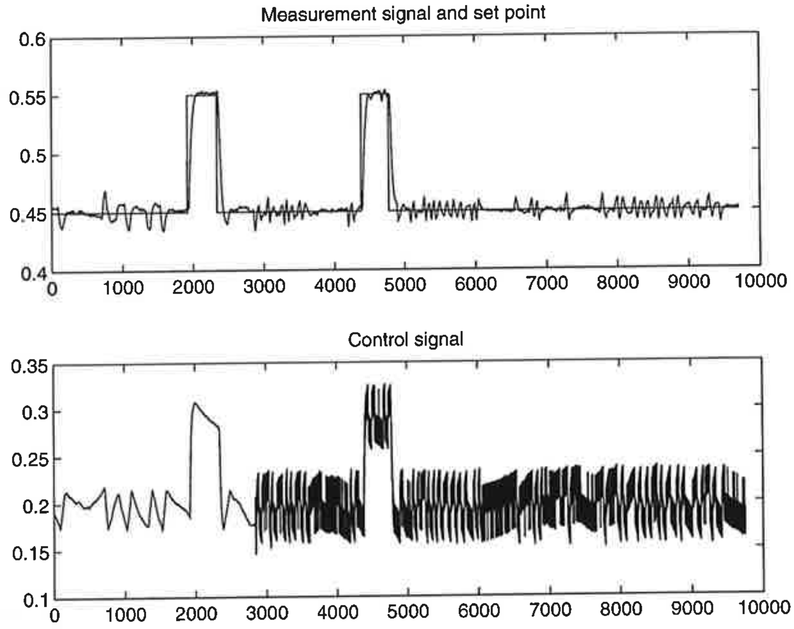

Figure 6. Data obtained in Experiment 1.

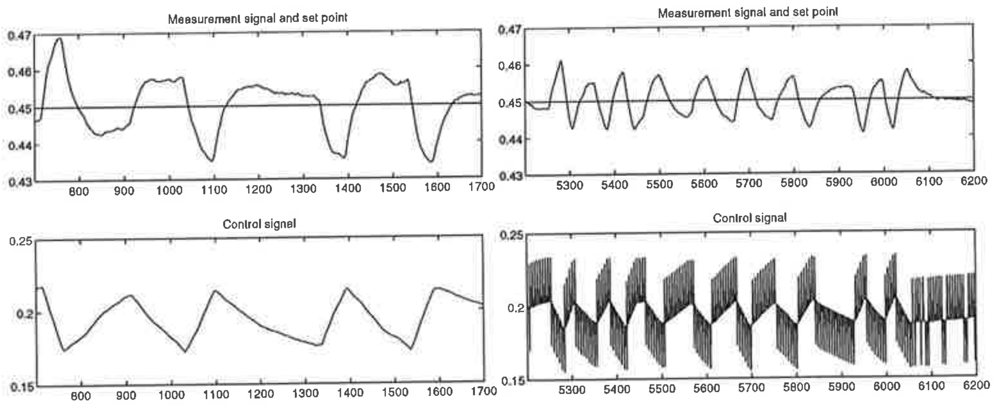

Figure 7. Control with a pure PI controller (left) and with the knocker (right).

had the following parameters:

$$a = 3\% \qquad \tau = 2h = 0.4s \qquad h_k = 5h = 1s$$

In Figure 7, 1 000 data points with pure PI control and with the knocker installed, respectively, are presented. The figure shows that the variations in control signal u_c, i.e., the component of the control signal that is generated by the PI controller, are much smaller when the knocker is applied.

It is also obvious from Figure 7 that the variations in the measurement signal decreases significantly when the knocker is used. Calculations of the integrated absolute error (IAE) and the integrated squared error (ISE), defined as

$$IAE = \frac{1}{T_2 - T_1} \int_{T_1}^{T_2} |e(t)| dt \tag{3}$$

$$ISE = \frac{1}{T_2 - T_1} \int_{T_1}^{T_2} e(t)^2 dt \tag{4}$$

also reveal this. The data series in Figure 7 gave $IAE_{PI} = 0.61\%$ and $IAE_{knocker} = 0.34\%$. The ratio between these two norms becomes

$$\frac{IAE_{knocker}}{IAE_{PI}} = 0.55$$

The integrated squared errors became $ISE_{PI} = 0.0055\%$ and $ISE_{knocker} = 0.0017\%$. This gives the following ratio between the norms:

$$\frac{ISE_{knocker}}{ISE_{PI}} = 0.31$$

The oscillation period using the knocker is about one third of the period obtained using pure PI control. The advantage of this increased oscillation frequency is discussed in subsection "Results" below.

The calculations have been performed during periods with a sustained stick-slip motion. In Figure 6, it is seen that there are periods when the measurement signal is quite close to the setpoint, and where therefore only a very slow drift in the control signal occurs. These periods could be expanded significantly if a dead-zone were used in the controller. Since the knocker generates a higher frequency in the oscillations, it will have the additional advantage of taking the system to this stationary state in a much shorter time than a pure PI controller. Therefore, using a dead-zone in the controller it is possible to reduce the long-term variations even more than indicated by the ratios given above.

Experiment 2 Figure 8 shows a similar experiment as Experiment 1. Here, the following parameters were used in the knocker:

$$a = 3\% \qquad \tau = h = 0.2s \qquad h_k = 2h = 0.4s$$

The difference from the previous experiment is that both the pulses and the sampling period between them are shorter.

Two sets of the data series in Figure 8 are shown in more detail in Figure 9. The date series in Figure 9 gave $IAE_{PI} = 0.42\%$ and $IAE_{knocker} = 0.31\%$. The ratio between these two norms become

$$\frac{IAE_{knocker}}{IAE_{PI}} = 0.75$$

The integrated squared errors are calculated to $ISE_{PI} = 0.0024\%$ and $ISE_{knocker} = 0.0013\%$. This gives the following ratio between the norms:

$$\frac{ISE_{knocker}}{ISE_{PI}} = 0.54$$

The period of oscillation was three times longer during pure PI control compared to control with the knocker.

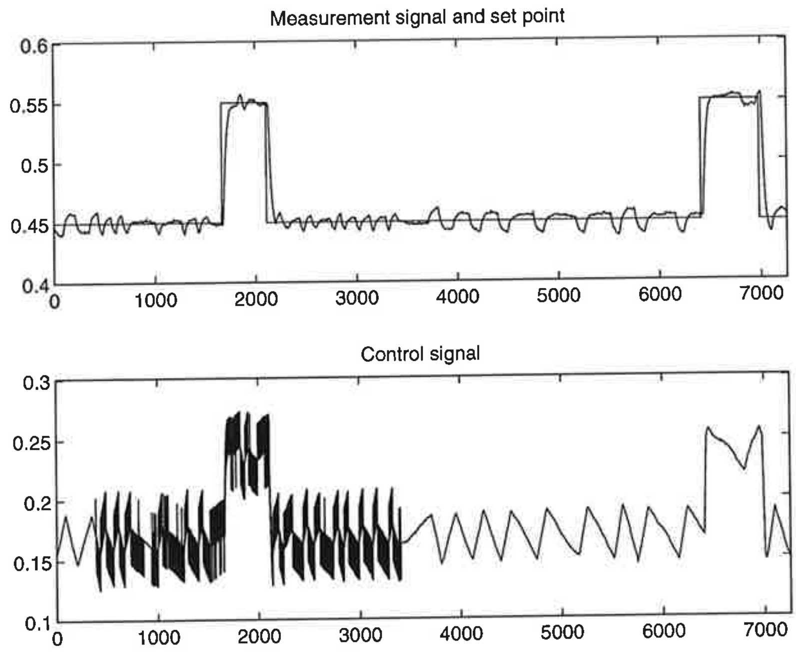

Figure 8. Data obtained in Experiment 2.

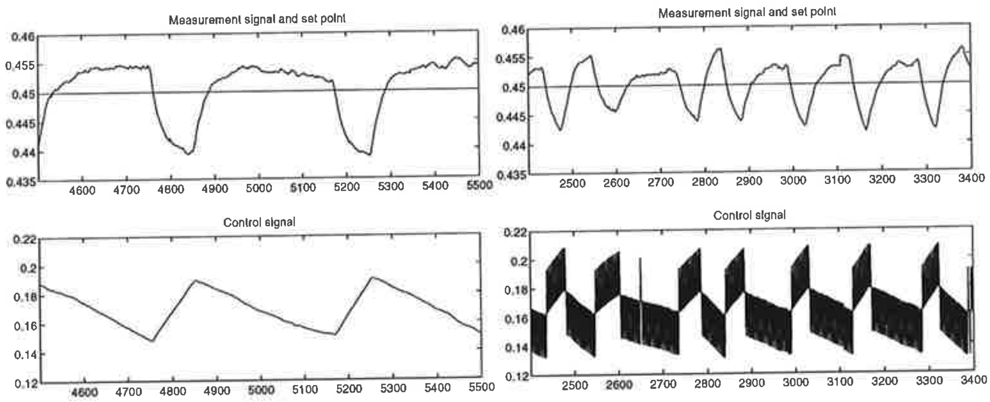

Figure 9. Results obtained with a pure PI controller (left) and with the knocker (right) in Experiment 2.

Choice of sampling period h_k **:** Both the pulse width τ and the sampling period h_k should be sufficiently short in order not to introduce too much energy at the time of slip. Figure 10 illustrates what may happen if sampling period h_k is too long. In this experiment, the knocker had the following parameters:

$$a = 3\% \qquad \tau = h = 0.2s \qquad h_k = 20h = 4s$$

Because of the long sampling period, $h_k = 4s$, control signal u_c will build up almost the pressure that is needed to generate a slip of the valve. When the pulse is applied, the pressure drop over the valve piston will be much larger than what is needed to overcome the stiction. The result is an oscillation that is significantly larger than what is obtained during pure PI control.

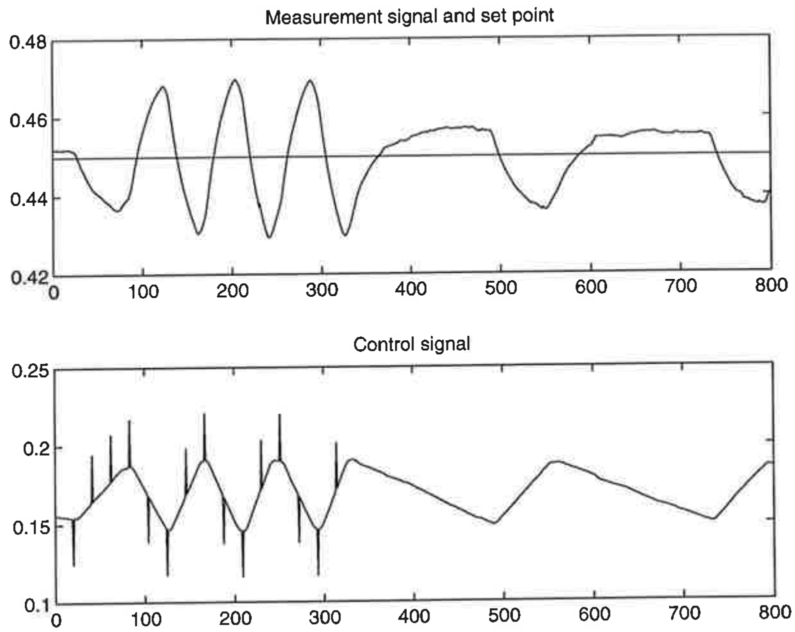

Figure 10. Too long periods between the pulses from the knocker deteriorates control. In the experiment, the sampling period was $h_k = 4s$.

Results

The field tests demonstrated that the knocker manages to improve the control during stick-slip motion in two ways. The knocker reduces the oscillation amplitude and increases the oscillation frequency. These two advantages are not independent, but the reduction of the oscillation amplitude is obtained mainly because of the increased oscillation frequency.

In the field tests presented above, the IAE decreased to 55% – 75% and the ISE decreased to 31% – 54% of the values obtained with pure PI control. It is likely that the improvements will be smaller in very fast flow control loops, where the measurement signal gives an immediate response when the flow changes. In these cases, fast control can be obtained with pure PI control. On the other hand, with filters and other dynamics between the point of stiction and the controller input, the improvements will be larger.

Flow controllers are often used in secondary loops in cascade control. Stick-slip motion in these inner loops form an oscillating load disturbance on the primary control loop. By increasing the frequency of these oscillations, their effect on the primary controller will be smaller, since the primary control loop has a lower band width, and therefore attenuates higher frequencies more than lower.

It is desirable to use a deadzone in the controller when stick-slip motion occurs, so that the controller stops integrating the control error when the control error is within an acceptable bound. Because of the higher oscillation frequency obtained with the knocker, the control system will reach the deadzone in a shorter time compared with a conventional controller. Therefore, if the controller is combined with a deadzone, the improvements obtained by using the knocker will be even more pronounced than what was obtained in the field tests.

Finally, the field tests have shown that the knocker is relatively unsensitive to parameter choices. It is probably possible to fix amplitude a once and for all to a value in the interval $1\% \leq a \leq 4\%$. Pulse width τ

can perhaps be fixed to one or a few sampling intervals. Perhaps it should be adjustable. This will be obvious when the procedure is applied to larger actuators. It is reasonable to choose sampling interval h_k in relation to τ, e.g. $h_k = n\tau$, where n is in the interval $2 \leq n \leq 5$. It should also be ensured that h_k always is less than a fraction of integral time T_i.

7. Conclusions

Control valve deficiences are the major source of process variability in the process industry. Until recently these problems have been unrecognized or, if recognized, often ignored.

The only way to eliminate stick-slip motion is to undertake a maintenance of the valve. This can normally not be performed without interrupting the production line. Therefore, even when the stick-slip motion is detected, it will continue until the next stop of production.

In this paper, a new control strategy is proposed. *The knocker* improves control-loop performance during stick-slip motion by adding short pulses to the standard controller output.

The intention is that the knocker should be introduced whenever it is detected that the friction level has increased so much that stick-slip motion results. It could be made automatically after an alarm from the stick-slip motion detection procedure presented in Hägglund (1995), but it is probably desirable to let the operator make this decision.

Use of the knocker will not cause any significant wear of the valve. During periods when the valve is stuck, the pulses will only affect the pilot valve. It is only those pulses that forces the valve to slip that will influence the valve.

Industrial field test have shown that the knocker manages to reduce the amplitude and the oscillation period significantly compared to conventional control. This will not only yield a more effective production. It may also increase the times between production stops, since production often has to be interrupted just because of valve problems.

8. Acknowledgements

This work was partially supported by the Swedish National Board for Industrial and Technical Development (NUTEK). The industrial experiments were performed in collaboration with Somas AB.

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