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Disturbance supervision in feedback loops

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Abstract : This paper treats the problem of disturbance supervision in feedback loops. Disturbances in feedback loops are unavoidable, but there are certain disturbances that can and should be eliminated. These disturbances can be introduced to the control loop from an external source, but they can also be generated inside the loop by friction in the valves or actuators. A procedure to detect these disturbances is presented, as well as those actions that should be taken to remove them. The detection procedure does not have any parameters to be tuned by the operator. It is intended to be included on a supervisory level in both adaptive and constant parameter controllers.

1. Introduction

There are always disturbances in feedback loops. If not, there would not be any reason to use feedback, but the process input signal could be given a constant value once and for all.

The controller is normally not able to eliminate the effects of the disturbances completely. The disturbances introduce control errors and consequently deteriorate the control. It is therefore of interest to try to eliminate the disturbances, if possible. This can be done in different ways depending on the origin of the disturbances. If the disturbances are introduced from an external source one should of course try to remove them at the source. If this is not possible, feedforward could be tried. If the disturbances are generated by friction inside the loop, valve maintenance should be performed.

The problem of load disturbances is of particular importance in adaptive control. Oscillating disturbances with frequencies in the neighbourhood of the ultimate frequency is one of the most common reasons for bad performance in adaptive control. See Hägglund (1991). These disturbances will detune most adaptive controllers, since the adaptive controllers interpret the oscillations as a result of a too high loop gain. It is therefore of interest to detect this situation automatically and stop the adaptation to avoid the detuning.

The paper is organized as follows. In the next section, the problem of load disturbances is further discussed, and the need for disturbance detection procedures is demonstrated. Section 3 treats the problem of friction in feedback loops, and it is shown that friction can conveniently be handled in the context of load disturbances. Section 4 presents a detection procedure, which provides a detection of oscillating disturbances. The diagnosis problem, i.e. how to isolate and eliminate the disturbances, is treated in Section 5. The detection procedure is analysed in Section 6, and conclusions are finally given in Section 7.

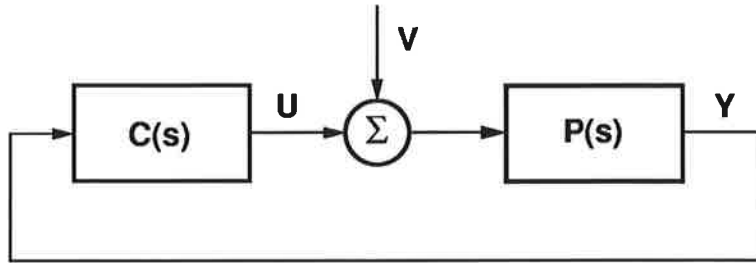


Figure 1. A simple feedback loop

2. Disturbances in feedback loops

By "disturbances" we mean all signals that are introduced to the feedback loop. It can e.g. be set-point changes introduced to the controller, high frequency noise added to the measurement signal or load changes, caused e.g. by changed operating conditions, introduced somewhere in the process. In the next section, it will be shown that friction can also be treated as a load disturbance added to the control signal.

The set-point change disturbances will not be treated in the following discussion. It differs from other disturbances in the sense that it is always known to the controller and can therefore be treated in special ways such as passing it through filters or ramping modules.

The disturbances should be treated by the controller in different ways depending on their frequency content. We will distinguish between high frequencies (HF), middle range frequencies (MF) and low frequencies (LF). These notations are of course coupled to the frequency response of the process. The frequency range around the process ultimate frequency ω_u , i.e. the frequency range where the process has a phase lag of about -180° , will be denoted middle range.

Consider the simple feedback loop in Figure 1, consisting of a process $P(s)$ and a controller $C(s)$. We will investigate how load disturbances $V(s)$, entering the control loop at the process input, are transferred to the measurement signal $Y(s)$. The transfer function between $V(s)$ and $Y(s)$ is given by

$$G_{vy}(s) = \frac{P(s)}{1 + P(s)C(s)} \quad (1)$$

We will determine the magnitude of $G_{vy}(i\omega)$ for frequencies in the LF, MF and HF regions respectively. To do so, we will first determine the magnitudes of $P(i\omega)$ and $C(i\omega)$ at these frequencies.

The process has normally a low-pass character, which means that the process gain at high frequencies, $|P(i\omega_{HF})|$, is small. The process gain in the middle range area has normally an order of magnitude equal to one. We shall say that $|P(i\omega_{MF})|$ is "moderate". The value of $|P(i\omega_{LF})|$ is normally moderate or, if it is an integrating process, high.

If the controller contains an integrator, the magnitude of $C(i\omega_{LF})$ is high. The controller has a moderate magnitude at middle range frequencies, i.e. $|C(i\omega_{MF})|$ is moderate. At high frequencies, $|C(i\omega_{HF})|$ is moderate or small, depending on whether the controller has high frequency roll off or not.

To summarize, in the normal situation the controller $C(i\omega)$ and the process $P(i\omega)$ have the magnitudes given in Table 1. From these observations,

Table 1. Process ($P(i\omega)$) and controller ($C(i\omega)$) gains for low frequencies (LF), middle range frequencies (MF) and high frequencies (HF)

	LF	MF	HF
$ P(i\omega) $	high/moderate	moderate	small
$ C(i\omega) $	high	moderate	moderate/small

we can draw the following conclusions concerning the frequency dependence of the load disturbance:

At low frequencies,

$$|G_{vy}(i\omega_{LF})| \approx \frac{1}{|C(i\omega_{LF})|}$$

Since the controller has a high gain at low frequencies, we can furthermore conclude that the effects of low frequency disturbances are eliminated effectively by the controller.

At high frequencies,

$$|G_{vy}(i\omega_{HF})| \approx |P(i\omega_{HF})|$$

Since the process has a low pass character, we can conclude that these disturbances are effectively filtered out by the process. High frequency components in the measurement signal are therefore normally not introduced in the process, but in the sensor or on the connections between the sensor and the controller. Since they do not contain any valuable information about the status of the process, they should be filtered out by the controller. It is also important not to transfer these signals to the controller output, since they may cause wear on the actuating equipment.

At middle range frequencies, both the process and the controller have moderate magnitudes. This means that these disturbances are neither eliminated by the controller, nor filtered out by the process. Since the phase shift of $P(i\omega_{MF})C(i\omega_{MF})$ is close to -180° , these disturbances may even be amplified because of the feedback.

The observations above are illustrated in Figure 2. The figure shows the controller and process outputs when the control loop is subjected to sinusoidal load disturbances of different frequencies and unit amplitude. The process has the transfer function

$$P(s) = \frac{1}{(s+1)^3}$$

and the controller is a PI controller tuned according to the Ziegler-Nichols rules. The controller parameters are $K = 3.2$ and $T_i = 2.9$ respectively. In Figure 2, the process output is almost unaffected by the low frequency disturbances because of the controller compensations. The middle range frequency disturbances are amplified because of the feedback, and it is obvious that the controller is unable to compensate for these disturbances. The high frequency disturbances are filtered out by the process. It means that both the process and controller outputs are unaffected by the disturbances.

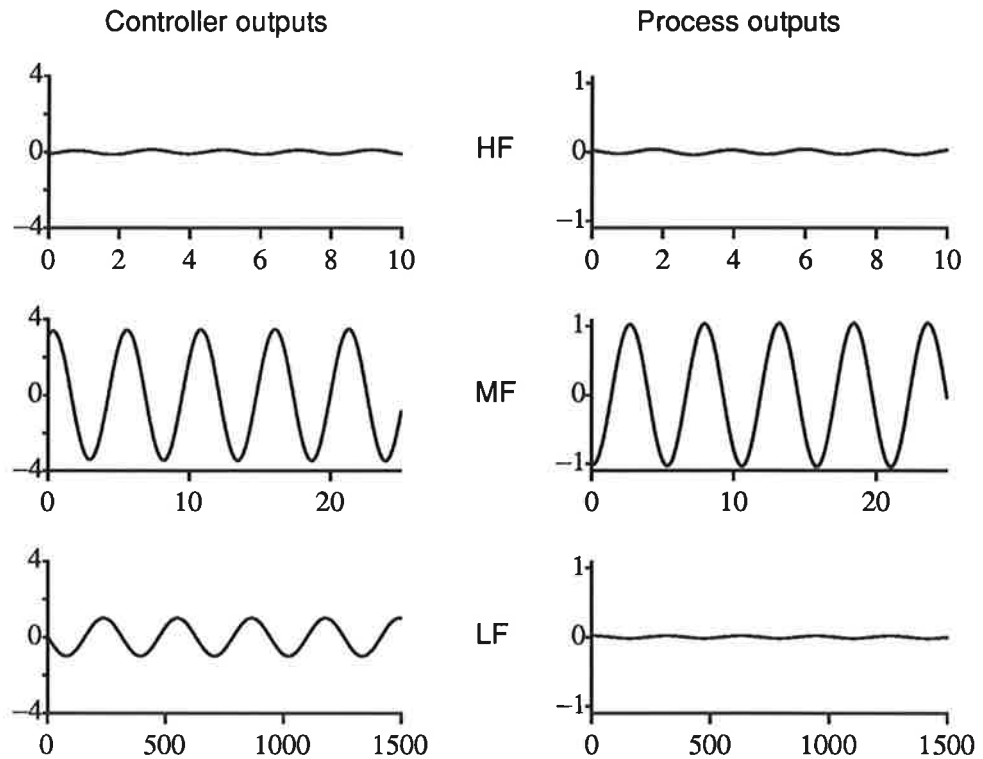


Figure 2. The process and controller outputs when the load disturbance is a sinusoid with low (LF) middle range (MF) and high frequency (HF)

The analysis above shows that the most severe disturbances are those with frequencies in the middle range. These disturbances are too fast to be treated efficiently by the controller, and they are too slow to be filtered out. The result is that the quality of the control deteriorates when the control loop is subjected to middle range frequency disturbances.

The disturbances are most severe in quality control loops, where they will result in a less uniform product. The *IAE* or the *ISE* criteria can often be used as economic-performance criteria of the plant. See Shinsky (1990). In other control loops, where the variations in the measurement signal perhaps can be accepted, the oscillations in the control signal will often result in a higher energy consumption. Even if one can accept disturbances like those in figure 2 in a certain control loop, these disturbances are often transferred to other control loops where the consequences are higher.

Therefore, it is recommendable to try to remove the middle range frequency disturbances from the control loops. To guide the operator in this work, a detection procedure for automatic detection of these disturbances is presented in Section 4.

3. Friction in valves

The previous section dealt with disturbances that were generated outside the feedback loop. In this section, disturbances that are generated inside the loop will be treated.

A valve (or actuator) with friction is known to result in "stick slip" motion and oscillations. Figure 3 shows an example of a control loop which is subjected to severe disturbances which the controller obviously is unable to

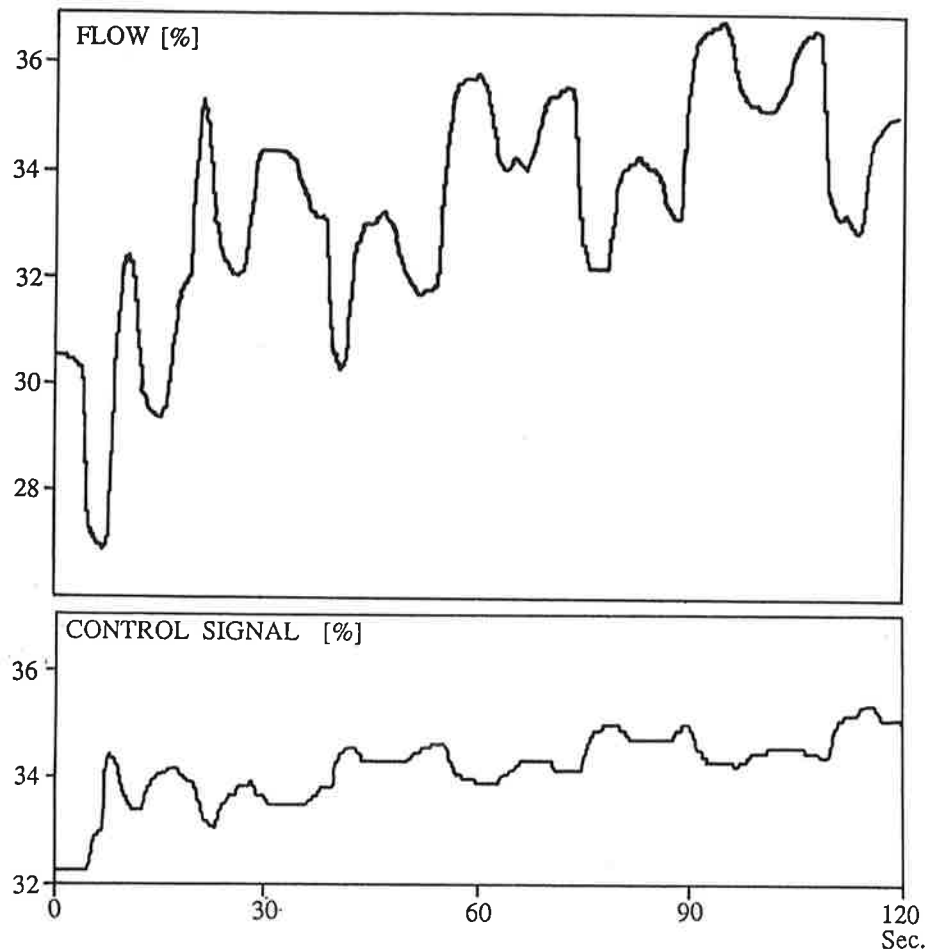


Figure 3. Stick slip motion in a flow control loop

compensate for. The disturbances are generated inside the loop because of too high friction in the valve. The example is a flow control loop in a paper mill. The figure shows the result of a step change in the set point. The controller used was a PI controller with controller parameters

$$K = 0.30$$

$$T_i = 34s$$

Notice that the settling time is very long.

Friction modeled as load disturbances

Friction in valves gives rise to nonlinear phenomena. The natures of these nonlinearities vary, depending on the type of valve and the type of friction (Coulomb, viscous, static). Some valves have a design that gives rise to an unsymmetric friction, so that the friction is much higher when the valve opens than when it closes (or vice versa). The friction will furthermore often result in wear which gives rise to more or less hysteresis in the valve.

Friction has often been treated using non-linear analysis, e.g. describing function techniques. See e.g. Smith (1958). These methods are powerful and give lots of insight to the problem. We have, however, decided not to use nonlinear analysis in this project. The major reason is the great variety of

nonlinearities that may appear and that we don't want to limit ourselves to any specific type of friction or valve.

A common feature of all types of friction is that they force the process input to be constant during certain periods, and to jump at certain time instants. The friction can therefore be modeled as a load disturbance $v(t)$ added to the control signal $u(t)$ according to Figure 1. The disturbance signal $v(t)$ representing the friction takes the following values:

$$v(t) = \begin{cases} 0 & \text{if no friction} \\ u(t_i) - u(t) & \text{if friction and } t_i \leq t < t_{i+1}, i = 1, 2, \dots \end{cases} \quad (2)$$

where $t_i, i = 1, 2, \dots$ are the time instants when the valve jumps to a new position. In Equation (2) it is assumed that when the valve jumps, it jumps to the position corresponding to the actual value of the control signal $u(t)$. This might not always be the case, but this restriction will not be of any importance for the further discussion.

Determination of oscillation frequency

If the valve in the control loop has too high friction, resulting in stick slip motion, it should of course be subjected to maintenance. It is very common that the oscillations caused by the friction are believed to be caused by bad controller tuning, and the controllers are therefore often detuned by decreasing the gain or increasing the integral time. So was the case with the flow loop presented in Figure 3. A retuning of the controller resulted in the following controller parameters:

$$K = 0.19$$

$$T_i = 2s$$

Notice that the integral time was decreased from 34s to 2s! A step response experiment using the new controller settings is shown in Figure 4. The settling time is significantly shorter than in Figure 3. It is also more obvious that the oscillations are really caused by friction, since we have the typical pattern of the measurement signal being close to a square wave and the control signal close to a triangular wave.

The period of oscillation T_{osc} is dependent of the process dynamics, the controller dynamics and the nature of the friction. In Figure 4, we have a situation of almost pure static friction. The period of oscillation T_{osc} can in this case be calculated in the following way.

Assume that we have a situation where the measurement signal is oscillating around the set point because of static friction in the control loop. This means that the friction will force the valve to move only when the control signal has changed an amount Δu since the last valve movement. Assume further that the oscillation is symmetric and that the control error e is constant between the valve changes, i.e. that it forms a square wave. The amplitude $|e|$ is given by the static process gain K_p times the amplitude of the control signal:

$$|e| = \frac{K_p |\Delta u|}{2} \quad (3)$$

The output from the PI controller is

$$u(t) = K \left(e(t) + \frac{1}{T_i} \int e(\tau) d\tau \right)$$

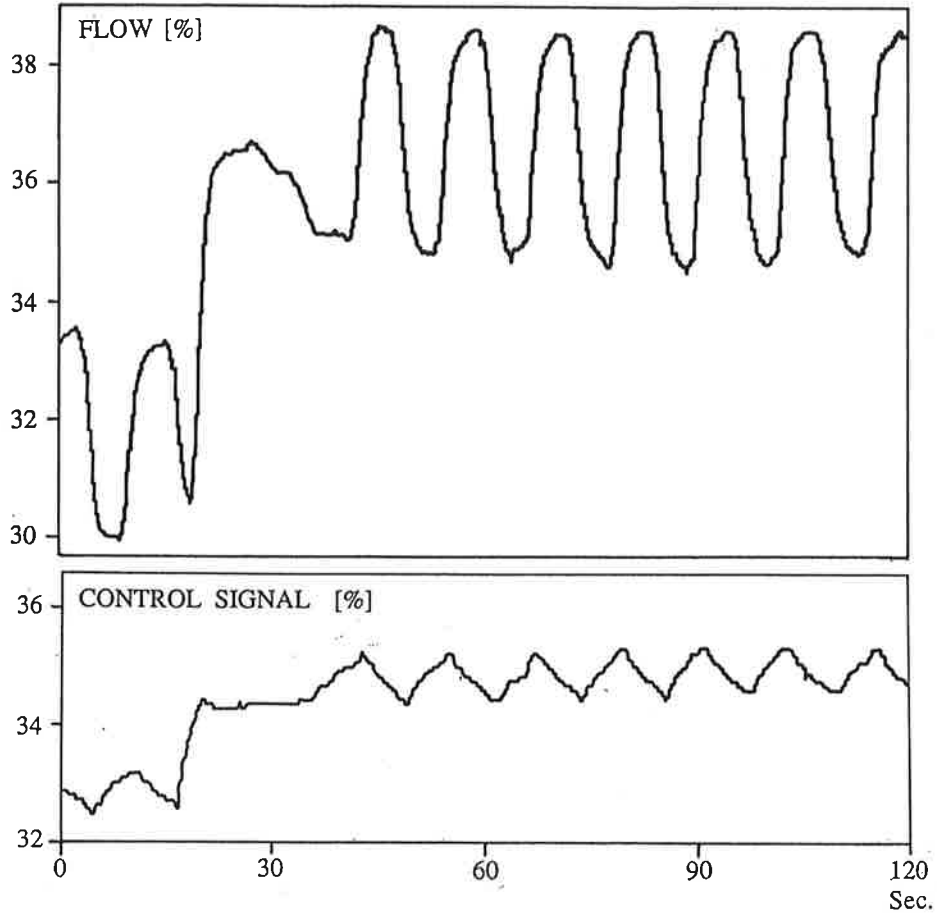


Figure 4. Stick slip motion in a flow control loop - retuned controller

The control error e causes the control signal to change an amount Δu between two consecutive valve jumps. This gives

$$\Delta u = Ke + \frac{K}{T_i} \int_{t_{i-1}}^{t_i} e dr = K \left(1 + \frac{t_i - t_{i-1}}{T_i} \right) e$$

Using Equation (3) and the fact that $T_{osc} = 2(t_i - t_{i-1})$ we get

$$\Delta u = K \left(1 + \frac{T_{osc}}{2T_i} \right) K_p \frac{\Delta u}{2}$$

This gives

$$T_{osc} = 2T_i \left(\frac{2}{K_p K} - 1 \right) \quad (4)$$

Equation (4) shows that the oscillation period is increased when the controller is detuned, i.e. when the gain is decreased or the integral time is increased. The oscillation period will also be longer if the oscillation is nonsymmetric.

The static gain in the flow loop above was calculated to be 3. Together with the controller parameters $K = 0.19$ and $T_i = 2s$ this gives an estimated oscillation period equal to $T_{osc} = 10s$ using Equation (4). In figure 4 the period can be measured to about 12s.

If the controller parameters used in Figure 3 are used in Equation (4), an oscillation period of 83s is obtained. This is clearly not in agreement with

the figure, were a much shorter oscillation period is given. We can therefore conclude, that with these controller settings, other phenomena than pure static friction occurs.

The ultimate period T_u of the process was identified to about $3.5s$, while the oscillation period in Figure 4 is about $10s$. With a detuned controller, Equation (4) shows that the oscillation period may become even longer. Hence, when the disturbances are caused by friction, both low frequency and middle range frequency oscillations are seen in the measurement signal, since the controller is unable to remove these.

To summarize, when the friction in the valves becomes large, stick-slip motion and oscillations occur in the control loop. These oscillations can not be eliminated by the controller. The oscillation frequencies can be either of low or middle range, depending on the controller tuning and the nature of the friction.

4. Detection of oscillations

In the previous sections, we have described different types of disturbances which the controller is unable to compensate for. In Section 2, it was demonstrated that externally generated disturbances with frequencies in the middle range area were particularly troublesome. In the previous section, friction was shown to give rise to serious disturbances in the low or middle range area. Since the controller is unable to compensate for these disturbances, and since they deteriorate the control performance, it is desirable to detect them and make the operator aware of the problem.

A major problem for the detection is that the nature of the disturbances that we want to detect can vary quite a lot. The frequency can be both low or middle range. Furthermore, the disturbances are often far from pure sinusoidals. See e.g. Figure 4, where the friction results in the characteristic square wave form of the measurement signal and triangular wave form of the control signal. The amplitudes can also vary. Of course it is more serious if the amplitude is high, but since we want to detect the disturbances at the source, even disturbances with a small amplitude is of interest for the detection.

The goal for this project is to obtain a detection procedure which is robust and easy to use, since the idea is that it should be possible to connect the procedure to almost every control loop in a process instrumentation. This is only possible if it is free from parameters to be specified by the user. Such a procedure will be presented in this section. First, the problem of detecting isolated load disturbances is treated.

A load disturbance detection procedure

The principle behind this new detection procedure is to study the magnitude of the integrated absolute error (IAE) between successive zero crossings of the control error, i.e.

$$IAE = \int_{t_{i-1}}^{t_i} |e(t)| dt \quad (5)$$

where t_{i-1} and t_i are two consecutive times of zero crossings.

During periods of good control, the magnitude of the control error is small, and the times between the zero crossings are relatively short. (It is

assumed that the controller has integral action, so that the average control error is zero.) This means that during good control, the IAE calculated in Equation (5) becomes small. When a load disturbance occurs, the magnitude of $e(t)$ increases, and a relatively long period without zero crossings occurs. This means that the IAE becomes large.

We can therefore conclude that the IAE , calculated according to Equation (5), can be used to detect load disturbances. When IAE exceeds a certain limit, which we shall denote IAE_{lim} , it is likely that a load disturbance has occurred. An advantage with this approach is that we do not assume any particular behaviour of the load disturbance apart from the fact that it should cause a significant deviation between the measurement signal and the set point.

To complete the detection procedure, a suitable value of IAE_{lim} must be determined. The choice of this limit is a compromise between the demand for a high probability of detection and the demand for a small probability of getting false detections. A low limit means that the detection probability is high, but unfortunately also that the rate of false detections becomes high. A high limit means that only large load disturbances will be detected, but with a smaller probability of false detections.

We will later use the load detection procedure to detect oscillations. Suppose that the control error is a pure sine wave with amplitude a and frequency ω , and that we want this signal to be detected as a sequence of load disturbances. This means that the integral of each half period of the oscillation must be greater than IAE_{lim} . We then get the following upper limit of IAE_{lim}

$$IAE_{lim} \leq \int_0^{\pi/\omega} a \sin(\omega t) dt = \frac{2a}{\omega} \quad (6)$$

The procedure should be able to detect oscillations in the low and middle range area. A requirement is therefore that frequencies up to ultimate frequency ω_u should be detected. A reasonable choice of a is 0.5 %, which means that we accept 1 % peak to peak oscillation. These parameter choices give

$$IAE_{lim} = \frac{1}{\omega_u} \quad (7)$$

The ultimate frequency ω_u is known if e.g. the controller is tuned with a relay autotuner, see Åström and Hägglund (1984), but normally it is unfortunately unknown. If the controller is tuned manually, the only information about the time scale of the process might be the integral time T_i . With a properly tuned PI(D) controller, the integral time is of the same magnitude as the ultimate oscillation period T_u . If ω_u is unknown, it will therefore be replaced by $\omega_i = 2\pi/T_i$ in Equation (7).

The load detection procedure can therefore be summarized as follows:

1. Choose a suitable acceptable amplitude a , e.g. $a = 0.5\%$.
2. Calculate IAE_{lim} as $2a/\omega_u$ if ω_u is available, otherwise as $2a/\omega_i$.
3. Monitor the IAE , where the integration is restarted every time the control error changes sign.
4. If the IAE exceeds IAE_{lim} , conclude that a load disturbance has occurred.

In Figure 4, the IAE is between 5 and 10 for the stable oscillation at the end of the recording, while $1/\omega_u \approx 0.56$ and $1/\omega_i \approx 0.32$. In this case, each half period would therefore be detected by the load disturbance detection procedure above.

Oscillation detection

Using the load detection procedure, we will now proceed and derive the oscillation detection procedure. The underlying idea is to conclude that an oscillation is present if the frequency of load disturbance detections becomes high. The behaviour of the control performance is monitored over a certain period of time, here called the supervision time T_{sup} . If the number of detected load disturbances exceeds a certain limit, which we denote n_{lim} , during this time, we will conclude that an oscillation is present.

What is then a suitable supervision time? A lower limit is given by the fact that we want an oscillation with frequency ω_u to be detected. If n_{lim} detections, where every half period is detected as a load disturbance, is to be obtained during the supervision time, it is required that

$$T_{sup} \geq n_{lim} \frac{T_u}{2} \quad (8)$$

where T_u is the ultimate oscillation period. In this project, we have chosen $n_{lim} = 10$. In this case, Equation (8) gives the lower limit of T_{sup} equal to $5T_u$. The oscillations that we want to detect may, however, have a significantly longer time period than T_u . To be able to detect these oscillations, we have chosen a supervision time which is 10 times longer, i.e.

$$T_{sup} = 50 T_u$$

If the ultimate period T_u is unknown, we relate the supervision time to the controller integral time instead:

$$T_{sup} = 50 T_i$$

Now, the algorithm could have been completed: "If at least n_{lim} load disturbances have been detected during the last T_{sup} seconds, conclude that an oscillation is present". This procedure is, however, quite ineffective to implement, since it requires that every load detection must be given a time label. It is easier to make an exponential weighting of the detections in the following way. At every sampling instant, the following procedure is called.

```

load := if a load is detected then 1 else 0;
x := γ x + load;
if x ≥ nlim then conclude that an oscillation is present;

```

The parameter γ is related to the supervision time T_{sup} as

$$\frac{1}{1 - \gamma} = T_{sup}/h$$

where h is the sampling period of the detection algorithm. This gives the following value of γ :

$$\gamma = 1 - \frac{h}{T_{sup}}$$

```

INITIALIZATION
a = 0.5 [%]
n_lim = 10
if the_ultimate_frequency is available then
begin
    iae_lim = 2 * a / omega_u;
    t_sup = 5 * n_lim * t_u;
end else
begin
    iae_lim = 2 * a / omega_i;
    t_sup = 5 * n_lim * t_i;
end;
gamma = 1 - h / t_sup;

LOAD DETECTION
if sign(e) = sign(e_old) then
begin
    iae = iae + abs(e) * h;
    load = 0;
end else
begin
    if iae ≥ iae_lim then load = 1 else load = 0;
    iae = abs(e) * h;
end;

OSCILLATION DETECTION
x = gamma * x + load;
if x ≥ n_lim then
begin
    oscillation = true;
    x = 0;
end;

```

Figure 5. The oscillation detection procedure

The oscillation detection procedure is summarized in Figure 5.

5. Diagnosis

When it is detected that a certain control loop is oscillating, it remains to find out why, i.e. to make a diagnosis. Figure 6, shows a systematic way to obtain the correct diagnosis.

The first problem is to determine if the oscillations are generated outside the control loop, or if they are generated inside the loop. This can be done by disconnecting the feedback by switching the controller to manual mode. If

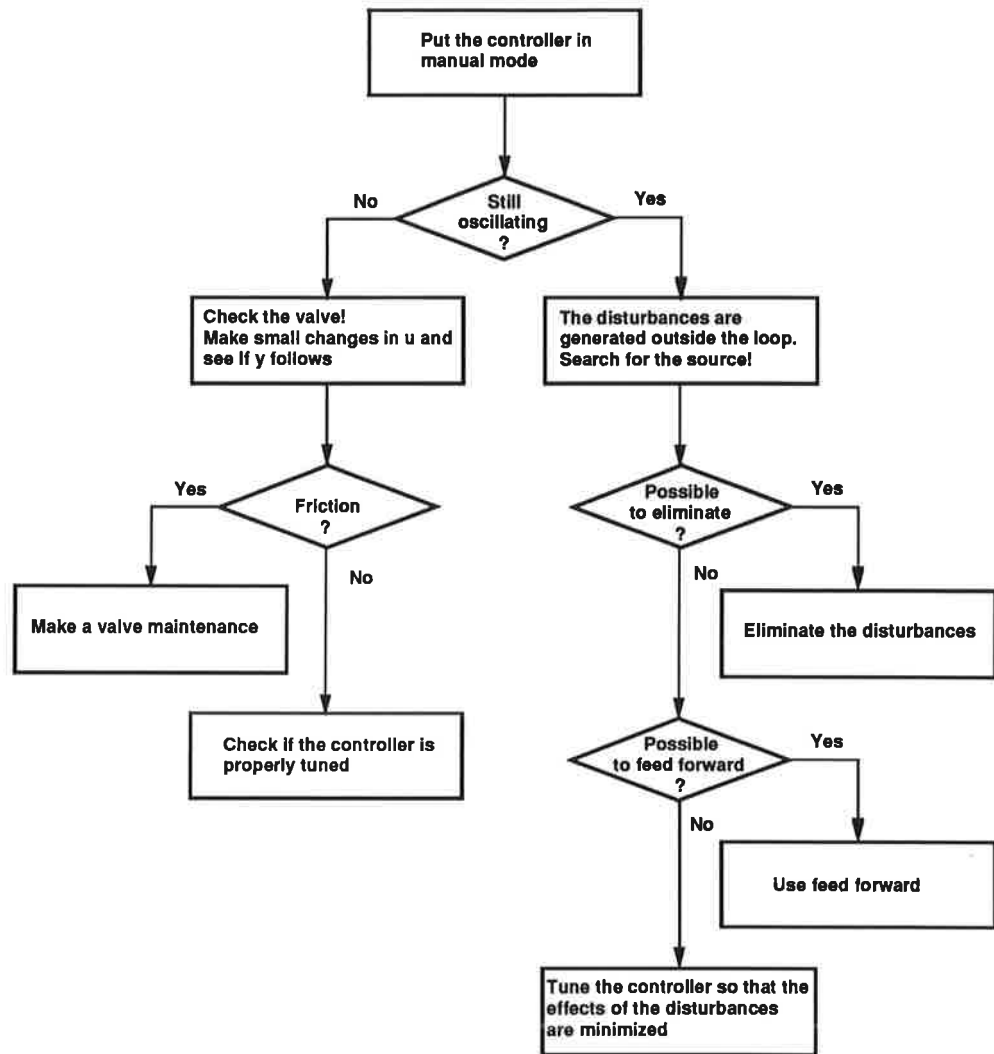


Figure 6. Oscillation diagnosis

the oscillation is still present, the disturbances must be generated outside the loop, otherwise they were generated inside the loop.

If the disturbances are generated inside the loop, the reason can be either friction in the valve or a badly tuned controller. Whether friction is present or not can be determined by making small changes in the control signal u and checking if the measurement signal y follows. If friction is causing the oscillations, the solution to the problem is to make a valve maintenance.

If the disturbances are generated outside the control loop, one should of course try to find the source of the disturbances and try to eliminate them. This is not always possible, even if the source is found. One can then try to feed the disturbances forward to the controller, and in this way reduce their effect on the actual control loop.

6. Analysis

In this section, we will investigate which disturbances that will be detected as oscillations by the proposed oscillation detection procedure.

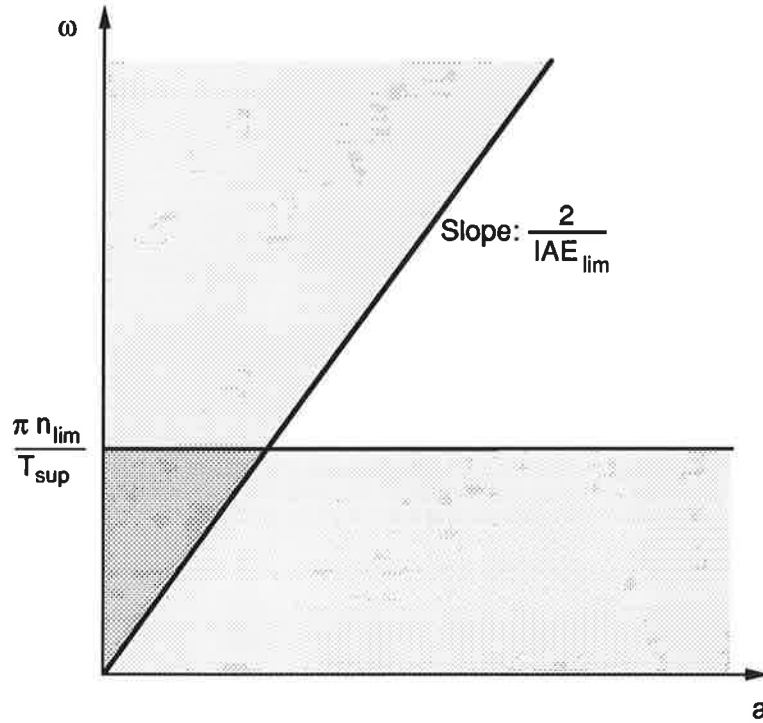


Figure 7. Requirements for detection of a pure sine wave with amplitude a and frequency ω . Detection is made for pairs of a and ω outside the shaded areas.

Suppose first that the disturbances causes the control error to be a pure sine wave with amplitude a and frequency ω . There are two requirements that must be fulfilled if this signal shall be detected as an oscillation. The first one comes from the load detection procedure. Each half period is detected as a load disturbance if and only if the IAE , where the integration is restarted every time the control error e changes sign, is greater than IAE_{lim} , i.e. if

$$IAE = \int_0^{\pi/\omega} a \sin(\omega t) dt = \frac{2a}{\omega} \geq IAE_{lim} \quad (9)$$

This requirement is fulfilled if the amplitude is sufficiently high and the frequency is sufficiently low.

The second requirement is only imposed on the frequency, not on the amplitude. The fact that at least n_{lim} half periods must occur within the supervision time T_{sup} gives a lower bound on the frequency according to

$$\frac{T_{sup}}{\pi/\omega} \geq n_{lim} \quad (10)$$

The two requirements (9) and (10) are illustrated graphically in Figure 7. The two requirements can be united as

$$\frac{\pi n_{lim}}{T_{sup}} \leq \omega \leq \frac{2a}{IAE_{lim}} \quad (11)$$

In previous sections, the following values of n_{lim} , IAE_{lim} and T_{sup} have been

suggested:

$$\begin{aligned}n_{lim} &= 10 \\ IAE_{lim} &= \frac{1}{\omega_u} \\ T_{sup} &= 50T_u = \frac{100\pi}{\omega_u}\end{aligned}$$

With these values, Relation (11) can be simplified to

$$\frac{\omega_u}{10} \leq \omega \leq 2a\omega_u \quad (12)$$

where a is given in [%]. A sine wave with amplitude $a = 0.5\%$ will thus e.g. be detected if the frequency is between $\omega_u/10$ and ω_u .

In the analysis above, it has been assumed that the oscillation is a pure sine wave. In practice, the shape of the measurement signal is often more like a square wave than a pure sine wave. Since a square wave has a larger magnitude of IAE than a sine wave with the same amplitude, these signals are easier to detect with the presented detection procedure than the sine waves.

7. Conclusions

Most control problems are solved using feedback, which is an approach with many advantages. However, this approach causes also some problems. Disturbances in the middle range frequency area can not be handled efficiently - they may even be amplified because of the feedback. These disturbances can be introduced to the control loop from an external source, but they can also be generated inside the loop by friction in the valves or actuators.

In process control, the many control loops are often coupled more or less strong to each other. This means, that if one control loop starts to oscillate, it is likely that this oscillation will be spread to other surrounding control loops. This is the reason why so many control loops in the process industry are in fact oscillating.

The oscillation problem is especially troublesome when adaptive control is used, since most adaptive controllers will be detuned by oscillating disturbances with middle range frequencies.

In this paper, it is suggested to detect this control situation automatically. A robust detection procedure is presented, which is based on the idea of monitoring the integrated absolute value of the control error (IAE) between successive zero crossings of the control error. The method does not assume any particular shape of the oscillations - only that the measurement signal should deviate significantly from the set point sufficiently many times during a certain supervision time.

The detection procedure is simple and robust. It does not have any parameter to be tuned by the operator. It can therefore be connected to all controllers in a process instrumentation. The operator will then get alarms from the controllers whenever the control loops starts to oscillate. A diagnosis procedure is also presented in the paper, which guides the operator in the problem of finding and eliminating the oscillations.

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9. References

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